Some Experimental Results of Tests of a Low-Speed, Waterjet Propulsion System

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Experimental data for static and dynamic performance of a 16½-in.-diam prototype waterjet propulsion system are presented. This system is designed for propulsion of a hydrofoil craft to operate in displacement mode at 9 knots, 175 hp, and be direct-driven at 2500 rpm. These specifications dictate low head and high throughflow pump characteristics and corresponding specific speeds and suction specific speeds exceeding 26,000 and 21,000, respectively. The single-stage axial-flow pump designed to perform this task required particular attention to correlation of pump parameters with cavitation criteria to override some of the limitations of current axial-flow pump technology. Performance results of the prototype unit indicate that useful thrust is not difficult to attain, but that the flow-choking and head-reducing influence of cavitation phenomena rapidly decreases over-all system efficiency. Evidently more sophisticated hydrodynamic theory to rationalize the balance between blading losses and blading total static pressure rise is required to maintain better efficiency levels. Preliminary observations and performance data presented here indicate some of the problems associated with the design of waterjet propulsion systems in which achievement of useful propulsive efficiency is dependent on the ability of the propulsor to operate with high axial velocities.

Nomenclature

A_j = jet nozzle area, ft^2
A_x = rotor inlet flow area, ft^2
C_* = dimensionless flow velocity, Eq. (5)
C_{x_1} = rotor inlet axial velocity, fps
F_0 = thrust, lb
g = gravitational constant, ft/sec^2
H_0'' = $(V_j^2 - V_0^2)/2g$ = system total head rise, ft
H_0 = $(P_{j0} - P_{10})/\rho$ = pump total head rise, ft
hp = horsepower
K_1 = cavitation parameter, Eq. (4)
NPSH = $(P_{10} - P_v)/\rho$ = net positive suction head, ft
N_s = rpm (gal/min)$^{1/2}/H_0^{3/4}$ = pump specific speed
P_1 = rotor inlet static pressure, psf
P_{10} = rotor inlet total pressure, psf
P_{j0} = pump exit total pressure, psf
P_v = vapor pressure, psf
Q = flow rate, ft^3/sec
S = rpm(gal/min)$^{1/2}$/NPSH$^{3/4}$ = pump suction specific speed
U_1 = rotor inlet peripheral speed, fps
U_* = dimensionless wheel speed, Eq. (6)
V_j = jet velocity, fps
V_k = craft speed, knots
V_0 = craft velocity, fps
W_1 = rotor inlet relative velocity, fps
$\dot{w}$ = flow rate, lb/sec
η_p' = $2V_0/(V_j + V_0)$ = ideal propulsive efficiency
$\eta_0\eta_1$ = $\dot{w}H_0''/550$ hp = system efficiency
η_p = $\eta_p'\eta_0\eta_1$ = net propulsive efficiency
ρ = specific weight, lb/ft^3
σ_1 = $[2g(P_1 - P_v)]/\rho C x_1^2$ = cavitation sigma

Introduction

THE successful demonstration of waterjet applications that can operate at sufficient system efficiency to obtain thrust and net propulsive efficiencies competitive with the marine propeller will offer practical and technical advantages not inherent to the simple propeller system. Propellers designed to absorb full power and deliver maximum thrust at starting or low craft velocities are ordinarily slow in rotation and large in diameter. With proper advancement in the hydrodynamics of pump performance, single-stage and multistage waterjet propulsors displaying small diameter and high peripheral speed characteristics can be designed to accomplish the same tasks. Smaller diameters and higher rotation for equivalent power inputs will greatly improve many gear-reduction operational and weight problems.

The installation of a waterjet system flush or semiflush with hull bottoms provides low draft characteristics advantageous to logistic considerations. Preliminary evidence also suggests that the pump jet in operation produces less underwater noise than a comparable propeller.[1] It is the intent of this paper to report the design and preliminary results of a waterjet propulsion system specifically designed to operate in this spectrum of ambitious objectives, and secondly, to indicate some of the more persistent obstacles that require additional research and development.

Design Considerations

The objective task for this propulsion system was specified succinctly. The unit had to be as small as possible, rotate at 2500 rpm, and deliver 2500 lb of dynamic thrust (F_0) at 9 knots (V_k) with 175 input shaft horsepower.

A net propulsive efficiency, then, is simultaneously specified:

$$\eta_p = F_0 V_k / 325 \text{ hp} = 0.395 \tag{1}$$

Examination of the basic requirements to meet these specifications indicates that the system necessitates a pump jet of low head rise and very high throughflow. Subsequently, with the specified rpm, the pump specific speed and suction specific speed characteristics are substantially higher than encountered in ordinary practice. There is no question that this machine poses a challenging task in axial-flow pump design.

Experimentation and research in waterjet propulsion emphasizes cavitation as the major influence on the specific

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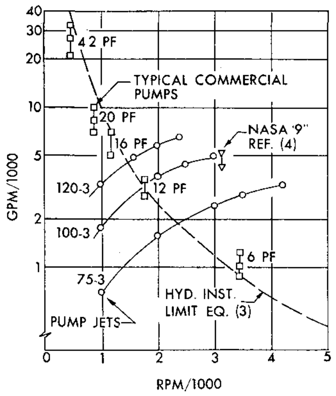


Fig. 1 Typical pump speed-flow characteristics, axial flow pumps.

mass flow characteristics of waterjet pumps.[2] That propulsion is most effective when mass flows are high and pressure increases are low is axiomatic. The efficient attainment of high mass flows per unit inlet area, however, is sorely impeded by cavitation phenomena, particularly when the inlet pressure field is ambient or lower. There is no simple theory that will describe these phenomena and permit the design of high-speed, high-flow pumps to guarantee their efficient operation in the cavitation regime. In lieu of this, empirical rules and limitations have been formulated, and are commonly accepted by the general field of pump technology.

Typical of these recommended limits is Fig. B5-35 of the handbook of standards of the Hydraulic Institute (HDI).[3] This chart can be expressed approximately by

$$H_z = 30 - (N_s^{4/3}H_0/0.166 \times 10^6) \tag{2}$$

For zero lift ($H_z = 0$) then,

$$N_s^{4/3}H_0 = 5\cdot 10^6$$

or in terms of flow (gal/min) and speed (rpm)

$$\text{rpm} = 106{,}000/(\text{gal/min})^{1/2} \tag{3}$$

Equation (3) is shown in Fig. 1. Flow capacities for various diameter pumps are shown. The data for the commercial pumps have been taken from a standard pumps catalog, and show the general compliance with HDI for "propeller" single-stage pumps ranging in diameter from 6 to 42 in. The flow capacities of three production-model, axial-flow, waterjet pumps operating at zero lift (static condition), and the NASA axial-flow pump[4] operating at an NPSH of 33 ft, are included in this figure.

It is evident, then, that it is indeed possible to increase the flow capacity of axial-flow pumps considerably over the recommended limits of standard practice. That is a necessary characteristic for propulsion applications; the question remaining is "How can this be accomplished without excessive compromise in over-all efficiency?" The approach to this problem and the design philosophy used for this machine, in general, reflect the current technology of the so-called cavitation parameter,

$$K_1 = [2g(P_1 - P_v)]/\rho W_1^2 \tag{4}$$

but with some modification deemed appropriate to advancement of the state-of-the-art.

A dimensionless number is defined for the velocity of the throughflow, corrected to the inlet total pressure field, as

$$C_*^2 = \rho C x_1^2/[2g(P_{10} - P_v)] \tag{5}$$

and the corresponding wheel speed number as

$$U_*^2 = \rho U_1^2[2g(P_{10} - P_v)] \tag{6}$$

Then, it can be shown that the cavitation parameter (K_1) is, simply

$$K_1 = (1 - C_*^2)/(U_*^2 + C_*^2) \tag{7}$$

and applies to each blade cascade segment of the rotor.

Equation (7) is plotted in Fig. 2. The traverse data from the NASA axial-flow stage[4] and a 78° helical inducer[5] plotted on this figure were used to indicate quantitatively the useful regime of operation for our pump jet design. The cavitation parameter and its effects, presented in the manner of Fig. 2, is taken to mean that the "corrected wheel speed" is the primary determinant for the efficient operation of a rotating cascade at low (K_1) values. The experimental data shown here indicate reasonable correlation on that basis.

The corrected throughflow velocity (C_*) is easily shown to be related to the classical cavitation sigma (σ_1) as

$$C_*^2 = \rho C x_1^2/[2g(P_{10} - P_v)] = 1/(1 + \sigma_1)$$

and, therefore, will have an absolute limiting value of unity for total cavitation ($K_1 = 0$).

The theoretical minimum limit curve shown in Fig. 2 suggests that when (C_*) approaches zero, NPSH approaches infinity and, therefore, suction specific speed approaches zero! Evidently, it is not suction specific speed that determines effective operation in cavitation environment, but (K_1) and (U_*). This would explain why some pumps cavitate badly at low suction specific speeds and some at high values. Pursuit of this concept on a theoretical basis can become quite complex, and is not in the intent of this paper.

The rotor cascade efficiency traverse data corresponding to the data included in Fig. 2 from Refs. 4 and 5 are shown in Fig. 3. The efficiency data for the 78° helical inducer show considerable scatter; nonetheless it is evident that blade-element losses are inversely proportional to the cavitation parameter (K_1), and increase rapidly from hub to tip as (U_*) increases.

The simple solution to the efficiency defects of cavitation at given flows (C_*) is to reduce corrected wheel speed (U_*). When the majority of the basic task parameters are specified, however, reduction in rotational speed may not be possible since wheel speed, in the main, is the theoretical source of pump work. Equitable combination of work requirements,

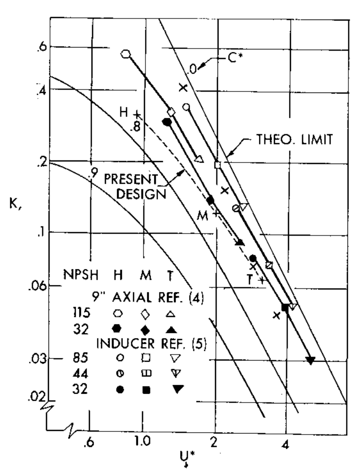


Fig. 2 Correlation of cavitation parameter K_1 with C_* and U_*.

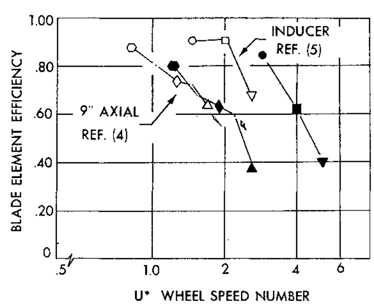


Fig. 3 Typical effect of cavitation intensity on rotor blade-element efficiency.

mass flow, cavitation criteria, and wheel speed must be accomplished to obtain the most useful compromise.

Design Approach

The preliminary task criteria for our design were obtained using the specified net propulsive efficiency ($\eta_p = 0.395$) and an assumed value of system efficiency ($\eta_0\eta_1$) of about 75%. The ideal propulsive efficiency is a function only of jet velocity ratio (V_j/V_0) and is readily obtained by dividing (η_p) by ($\eta_0\eta_1$).[6] The corresponding jet velocity (V_j) and desired thrust (F_0) obtain the throughflow (Q) and the system head rise (H_0''). The maximum NPSH is established by the design boat velocity (V_0).

From Fig. 2, a minimum value of cavitation parameter ($K_1 = 0.065$) stipulated for the rotor tip section concludes the determination of rotor diameter. This prologue, with some minor considerations, established the design point for this single-stage, axial-flow pump, as shown in Table 1. The design vector diagram values of (C_*) and (U_*) for the hub, mean, and tip rotor blade sections are indicated on Fig. 2.

A blade-element design approach quite similar to aerodynamic, axial-flow, compressor design was used to compute vector diagrams at five radial sections. With the exception that diffusion factor is not considered as the blade loading limit, this design concept parallels that of Ref. 4, and falls in the category of an inlet-type stage in which cavitation can occur, and in which the effects of loading and cavitation on the performance of the blade row are interrelated.

The rotor was bladed with double circular-arc elements using conventional rules to establish deviation and solidity. In order to expedite the fabrication and assembly of the unit for prototype testing, rotor blade patterns and molds were handmade in house, and stainless-steel blades were rough-cast individually at a local foundry. After nominal polishing, these were welded to slotted rotor hubs. Two handmade rotors were obtained, one corresponding to the design solidity with five blades, and another with seven blades. Photographs of the two impellers are shown in Figs. 4a and b. A cross section of the waterjet propulsion system is shown in Fig. 5. The stator comprises nine vanes cast integrally with the aft housing.

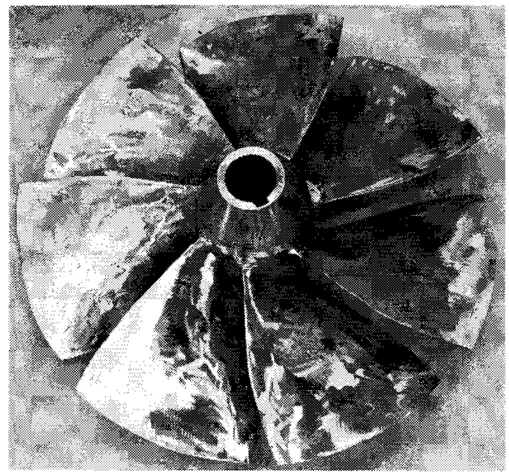
The inlet is a conventional flush-type to fit at the bottom of the hull. The tail section is fitted with a vectorable conduit for steering, and a flow-reversing bucket for reverse motion.

Experimental Results

The performance of the prototype pump was established from measurements of static thrust (Bollard pull), torque, rpm, and craft velocity. Nominal total and static pressure instrumentation was applied for a gross indication of 1) the total pressure losses in the intake, 2) the rotor inlet pressure field (or NPSH), and 3) the pump exit total pressure. The measured static thrust and horsepower absorbed are shown in Fig. 6 plotted against the theoretical prediction as functions of unit rpm. The static performance shown is for both

Table 1 Design point criteria for 16.5-in., single-stage, axial-flow pump

Total head rise (H_0), ft	25
Flow (Q), ft^3/sec	39.4
rpm	2,250
hp	137.5
Jet nozzle area (A_j), ft^2	0.973
Number of blades (Z)	5
NPSH, ft	32
Hub-tip ratio	0.212
Rotor tip diameter, in.	16.5
Pump specific speed (N_s)	26,800
Pump suction specific speed (S)	21,900



a) 7-bladed rotor

b) 5- and 7-bladed rotors

Fig. 4 Handmade $16\frac{1}{2}$-in. bladed rotor.

the 5-bladed and the 7-bladed rotors. The measured data do not indicate any material difference and the effect of solidity, in this case, is inconclusive.

The theoretical prediction is obtained from a general equation of performance for jet propulsion,

$$\frac{\eta_0\eta_1 2g550\ \text{hp}}{\rho A_j} = V_j^3\left(1 - \frac{V_0^2}{V_j^2}\right) \tag{8}$$

in which the design value of system efficiency ($\eta_0\eta_1$) is considered to remain constant with horsepower input, and the inverse velocity ratio (V_0/V_j) is zero.

The experimental data indicate that below 2250 rpm the unit is developing somewhat higher static thrust than predicted, but with higher horsepower absorption. Consequently, the over-all system efficiency is lower than predicted in the design. The unit, however, does operate at the specified 2500 rpm without obvious difficulty. The static thrust curve has "looped" somewhat at 2500 rpm, indicating that cavitation effects are setting in. These effects are not continuously audible and, after approximately 20 hr of testing, inspection reveals very slight physical markings of significance on the aluminum housing, and none on the blades and vanes.

The flow (Q) and system total head rise (H_0'') of the pump as obtained from the measured static thrust and jet nozzle area are shown in Fig. 7. It is interesting to note that under static test conditions the flow characteristics indicate "choking" as the static thrust loops over, and are reflected by the drop in system total head rise from the predicted values.

Only two total pressure pickups were used at the inlet face of the rotor to indicate the total pressure drop across the intake and the NPSH. At full rpm a maximum of 3 in. of Hg total pressure drop was measured. The corresponding values of suction specific speed calculated from these pressure data and flow and rpm are included in Fig. 7. It is

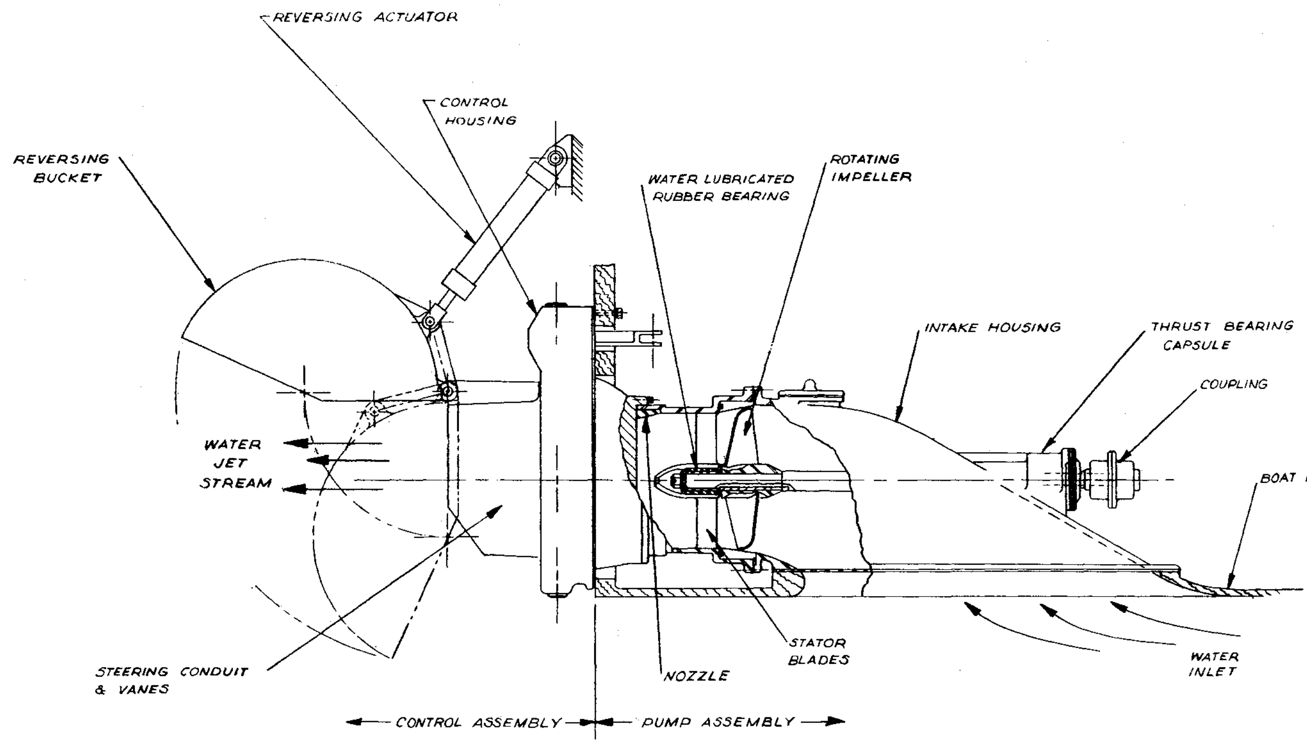


Fig. 5 Cross-sectional view of 16½-in. waterjet.

quite evident that this unit operates in a regime considerably beyond conventional limitations, and still provides useful performance.

Waterjet propulsion pumps operate on a constant-throttle line with varying rpm. A qualitative indication of the blading performance can then be obtained from the measured performance vs rpm, calculated against a representative blade-element diameter (tip diameter in this case).

Accordingly, the calculated values of cavitation parameter (K_1), corrected flow (C_*), and tip speed (U_{*t}) for the static performance are shown in Fig. 2 by the (X) symbols. The minimum NPSH is 29 ft. This technique shows fairly good agreement with the design concept insofar as Eq. (7) is concerned. However, it must be reasoned that because system efficiency was below expectations, more adequate means of relating (K_1), the work criteria at each radial segment, and the subsequent losses must be established.

Measurements of the tailpipe total pressures were taken both in the static test phase and in the dynamic test phase. This instrumentation, however, because of its location and sparsity, does not represent mass average values of jet velocity.

In order to establish a reasonable reference on which calculations could be based both to substantiate the measured static performance and calculate the dynamic performance, the following technique was used. Figure 8 compares the total pressure at the jet nozzle calculated from the static performance, and measured from the static and dynamic operation vs horsepower absorbed. The difference between the calculated pressures and the measured pressures in the static test phase is taken as the correction to the measured pressures in the dynamic phase. It can be noted that under dynamic operation the tailpipe total pressure is higher than in static operation. This simply means that the pump is recovering some of the dynamic pressure of the craft velocity.

The corrected tailpipe pressure is used to calculate jet velocities (V_j). The change in momentum is the difference between this jet velocity and the velocity obtained from the boat speed over the measured time course. Effective jet stream area is assumed to equal the geometric jet nozzle area to obtain mass flow.

A dynamic thrust and net propulsive efficiency can then be calculated at each measured horsepower or rpm setting.

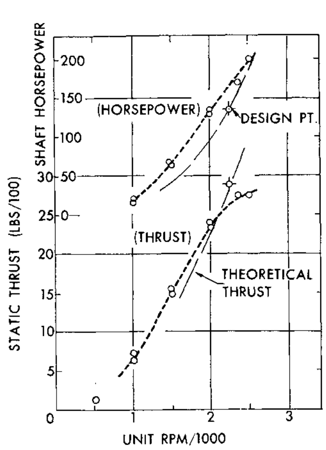


Fig. 6 Static performance of prototype 16½-in. waterjet propulsion unit.

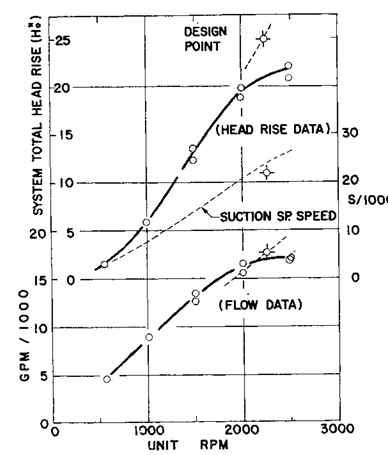


Fig. 7 Flow and head rise characteristics, 16½-in. prototype waterjet unit.

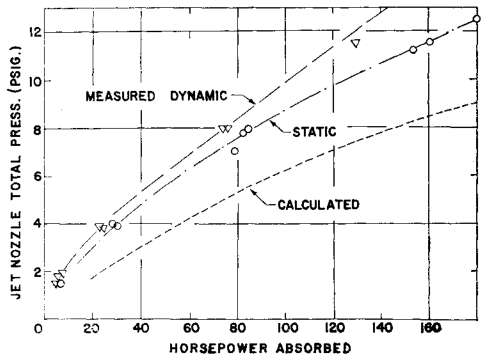


Fig. 8 Jet nozzle total pressure, 16½-in. waterjet.

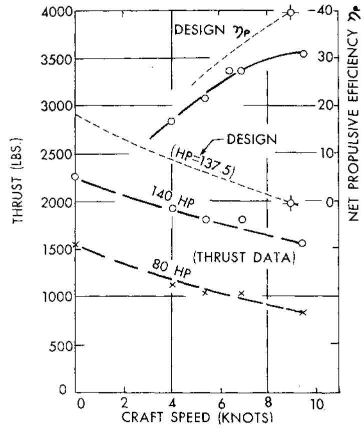


Fig. 9 Dynamic performance, 16½-in. waterjet system.

Dynamic performance is shown in Fig. 9 for two constant rpm-hp settings. The reduction in over-all system efficiency ($\eta_0\eta_1$) from design is reflected in the loss of approximately 400 lb of dynamic thrust at 9 knots. It can be speculated from these data that an increase in ($\eta_0\eta_1$) of only 0.089 points will make up this loss. It is not illogical to presume reassurance from the performance of this prototype unit. Its handmade blading is certainly not representative of modern production quality, and the results obtained are an incentive to the advancement of pump jet application to general marine propulsion.

Conclusions

Current experience shows that useful propulsive thrust can be achieved by single-stage, high-suction-specific-speed, axial-flow pumps with some compromise in system efficiency. It is to be expected that system efficiency of single-stage propulsors operating at high peripheral speeds and high specific mass flows will improve as more fundamental relationships are established between the design work of each radial blade element and its losses under its particular degree of cavitation environment. For propulsion tasks in the high-speed regime, it appears feasible that system efficiency can be increased with multistaging techniques that permit each additional stage to operate more advantageously as the total and static pressures increase across the propulsor.

Preblading of the inducer type, commonly used with pumps where inlet cavitation environment appears unavoidable, has unusually high solidity. The experimental results of increasing the solidity by a ratio of 7:5 in this prototype axial-flow stage do not indicate the effect of solidity. There is need for the demonstration of any results that will vitalize the role of waterjet propulsion in the future of advanced marine systems. It is hoped that the experimental results reported here will contribute to that need.

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