A Water-Turbine Driven Propeller for High Performance Ship Propulsion

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The feasibilty of a water-turbine driven propeller is investigated for high performance craft. The proposed system uses a gas turbine driven pump installed in the craft. The high pressure output from this pump is used to spin a ducted marine propeller by means of an impulse turbine attached to a ring around the propeller's tips. This system appears to be attractive for crafts operating at speeds up to 50 knots. It also appears to offer an appreciable reduction in propulsive machinery weight as well as elimination of mechanical transmissions and reduction gears. Propulsive efficiencies of the order of 43 to 46% are attainable. A point design for a 120 ton, 40 knot hydrofoil is presented.

Nomenclature

A_0	= equivalent orifice cross-section area for piping system
$C_D = \dfrac{D}{[T_T/(\rho U^2)]^{1/2}}$	= propeller diameter coefficient
$C_{p\min} = \dfrac{P_{\min} - P_\infty}{1/2 \rho U^2}$	= minimum pressure coefficient on propeller blades
$C_T = \dfrac{T}{1/2 \rho U^2 \pi D^2/4}$	= thrust coefficient
D	= propeller diameter
g	= acceleration due to gravity
H	= water head
H_D	= frictional head loss
H_p	= pump total head
H_{sv}	= net positive suction head
H_v	= vapor pressure head
$J = U/n_{pr}D$	= advance ratio
K_D	= friction constant
K_Q	= torque constant
K_T	= thrust constant
M	= torque
N	= rotative velocity in rpm
N_s	= pump specific speed
N_{sv}	= suction specific speed
n	= rotative velocity in rps
P	= power
p	= pressure
Q	= volume flow rate
T	= thrust applied to ship by the "screw + nozzle" combination
T_J	= thrust applied to ship by jet
$T_T = T + T_J$	= total thrust applied to ship
U	= ship velocity
u	= turbine blade peripherical velocity
V_J	= jet velocity
V_*	= ideal attainable nozzle exit velocity
$\bar{V}_* = V_*/U$	= velocity ratio
$\mu = T/T_T$	= propeller + nozzle thrust to total thrust ratio
$\tau = T_p/T$	= propeller thrust to propeller + nozzle thrust ratio
η	= propulsive efficiency
η_{pm}	= pump efficiency
η_{pr}	= propeller efficiency
η_t	= turbine efficiency
ρ	= mass density of fluid

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Introduction

AMONG the many possible forms of propulsion machinery for high performance craft applications, the marine propeller and the water jet are the most accepted ones. Since these craft are naturally weight critical, gas turbines are usually used as prime movers.

When a marine propeller is used, a mechanical drive is necessary to transmit the power from the prime mover (located inside the hull) to the propeller (usually well below the hull). A reduction in rotational speed is also required in this drive. The rotative velocity of an efficient gas turbine is of the order of 6,000 to 10,000 rpm, depending on its size and particular design, while for most hydrofoil ship applications the optimum speeds of subcavitating propellers are below 1,000 rpm. A great deal of effort has been put into the development of light and reliable mechanical transmissions. Still, transmission failure is one of the main problems in geared-propeller-driven hydrofoil craft.[1] In most cases, a compromise between efficiency and weight is made, resulting in a gas turbine rotating at approximately 1,500 rpm. Consequently, a reduction in efficiency is unavoidable, 55 to 60% can be thought as an average propulsive efficiency.

Often a water-jet system is selected instead of a propeller, since it eliminates the need for an inherently unreliable mechanical transmission. A well-designed water-jet propulsion system offers propulsive efficiencies of the order of 45 to 50%.[1,2] The weight of such a system is higher than that of an equivalent geared-propeller system, mainly because of the large quantity of water which is carried onboard.

The propulsion system proposed here is based on a marine propeller driven by a "hydraulic transmission" in which the required power is transmitted by a relatively small amount of water at high pressure. This system is most similar to aircraft turboprop systems. The main advantage of the water-jet system, the elimination of the mechanical transmission, is retained along with a light ducting system and a favorable pump specific speed. Further, it will be shown that the off-design performance for this system compares very favorably with either water-jet or geared propellers. The chief disadvantage of the system is its relatively low propulsive efficiency which is somewhat lower than or, at best, competitive with that of the water-jet system.

Water-Jet Propulsion

Since the turboprop system here has much in common with the water-jet system, a short review of the principles of the latter serves as a good starting point. The thrust T_J obtained from a jet propulsion system may be expressed as $T_J = \rho Q(V_J - U)$, where ρ, Q, U, and V_J denote the mass density of the fluid, the flow rate, ship velocity, and jet velocity relative to the ship, respectively. A fundamental design problem is to determine Q and V_J separately, so as to maximize the overall efficiency of the system. Energy losses occur in the piping system (including both the inlet or scoop and the jet nozzle), in the pump and in the direct loss of kinetic energy in the jet itself. The pump losses are usually expressed in terms of a pump efficiency η_{pm} and the jet kinetic energy loss is $\frac{1}{2}\rho Q(V_J - U)^2$. However, the losses in the piping system depend on a number of factors related to its size and shape, as well as the character of the flow within. If this flow is turbulent (as it is for all reasonable prototypes), then the losses can be approximated as a head loss H_D given by

$$H_D = Q^2/2gA_0^2 \tag{1}$$

where g is the gravitational constant; and A_0 is the cross-sectional area of a discharge-coefficient = 1 orifice which has the same head loss as the piping configuration. Lack of recovery of the inlet velocity head, $U^2/2g$, can be included in H_D.

The efficiency of the propulsion system, η is the ratio of the useful power $T_J U$ divided by the power input

$$P_{in} = [UT_J + \tfrac{1}{2}\rho Q(V_J - U)^2 + Q\rho g H_D]/\eta_{pm} \tag{2}$$

After elimination of Q, we can express η as

$$\eta = 2\eta_{pm}/[1 + \bar{V}_J + K_D/(\bar{V}_J - 1)^3] \tag{3}$$

where

$\bar{V}_J$ $= V_J/U$, the nondimensional jet velocity.

K_D $= [T_J/(\rho U^2 A_0)]^2$ a nondimensional constant relating the required thrust to the internal losses in the piping system.

Figure 1 shows the efficiency [Eq. (3)] and flow rate of a water-jet system vs the velocity ratio with K_D as a parameter and constant pump efficiency. Note that the dimensionless friction constant K_D does not depend on the velocity ratio $\bar{V}_J$. It is defined by the geometry and size of the pipes and the desired thrust and speed.

It can be seen from Fig. 1 that a compromise between efficiency and weight has to be made. A light system (large K_D's) yields a low propulsive efficiency. A highly efficient system, on the other hand, is necessarily large and heavy. Existing and proposed water-jet system offer propulsive efficiencies of the order of 45 to 50%.[1] This is obtained at velocity ratios of the order of 2 to 3, resulting in a relatively high flow rate and a low pump head. Consequently, extremely high pump specific speeds are encountered. This dictates the application of one or more of the following: 1) relatively slow gas turbines; 2) Reduction gears; 3) pumps arranged in parallel; and ; and 4) mixed flow or axial flow pumps.

All of these tend to increase weight and size and to reduce efficiency. The relatively high flow rates lead to difficult pump cavitation problems, especially at the take-off condition where only a small ram pressure is available.

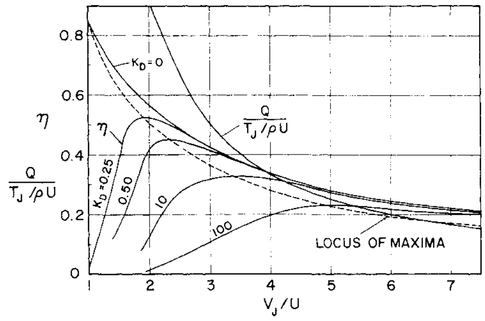

Fig. 1 Propulsive efficiency and flow rate of a water-jet system as a function of jet velocity (pump efficiency = 85%).

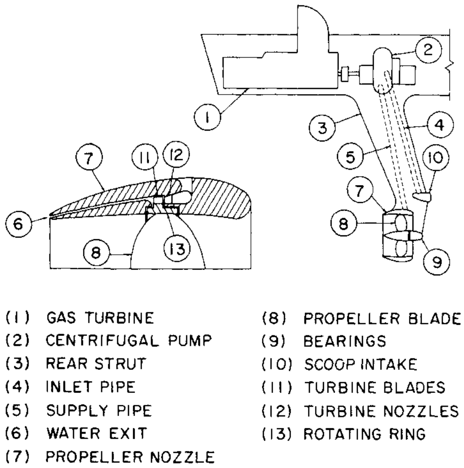

Fig. 2 Schematic arrangement of a turboprop system.

Turboprop Propulsion System

Introduction

The system is schematically shown in Figs. 2 and 3 in an arrangement appropriate for a hydrofoil craft. It consists of a pump, suction, and outlet pipes, and an impulse water turbine incorporated in the nozzle of a ducted propeller. The turbine blades are attached to the exterior of a ring, the interior of which is connected to the propeller blade tips. A water stream is drawn into the ship through a scoop. The water pressure is then raised with a conventional pump, *directly* driven by the prime mover (gas turbine). Water is then ducted to impulse turbine nozzles which are located inside the propeller duct along its periphery. Passing through the turbine nozzles, the pressure energy is converted into kinetic energy. Turbine blades attached to the outer surface of the propeller ring are driven by the water jet. The ring is fitted in the propeller nozzle such that the inner surface of the ring completes the interior surface of the nozzle.

The water turbine may be either single or two stage, depending on the particular design. When two stages are employed, an additional disk of stationary blades is necessary. The pressure across the turbine is assumed to be designed to be kept constant and equal to that existing at the propeller blade tips. Most of the thrust (up to 90%) is obtained from the propeller while the rest is contributed by the backward discharge of water from the turbine. The propeller shaft is used for support and thrust delivery, while the torque is transmitted to the propeller around its periphery.

Thrust, Power, and Efficiency

The thrust applied to the ship T_T is the sum of the thrust applied by the propeller and nozzle system and that contributed by the jet discharge. Thus $T_T = T + T_J$. To compute these components, we will first analyze the flow in the pump-turbine loop. The head of water available at the turbine is given by

$$H_T = U^2/2g + H_p - H_D \tag{4}$$

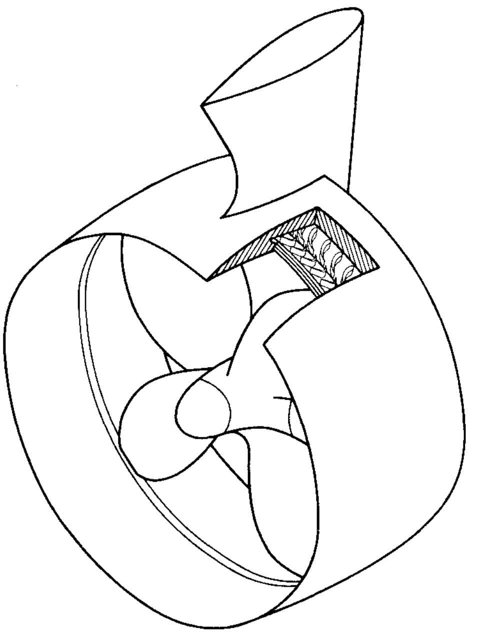

Fig. 3 Perspective view of propeller and turbine arrangement.

where H_p is the pump head, and H_D is the head loss in the piping, as before. If we let $V_* = [2gH_T]^{1/2}$, the ideal turbine inlet nozzle velocity, the useful power of the propeller and nozzle system is

$$P = \tfrac{1}{2}\rho Q V_*^2 \eta_t \eta_{pr} \tag{5}$$

where η_t is the turbine efficiency, and η_{pr} is the propeller efficiency for the nozzle and propeller system.

If we assume that the exhaust stream is discharged backward at V_J, the same velocity as it leaves from the last stage of the turbine (and at ambient pressure), then the thrust applied by this jet is $T_J = \rho Q(V_J - U)$, as before, and the total useful power may now be given as

$$P_T = T_T U = \rho Q[\tfrac{1}{2} V_*^2 \eta_t \eta_{pr} + U(V_J - U)] \tag{6}$$

The power input to the propulsion system is

$$P_{in} = [\tfrac{1}{2}\rho Q(V_*^2 - U^2) + Q\rho g H_D]/\eta_{pm} \tag{7}$$

The head loss H_D in Eq. (7) depends upon the flow rate Q as shown in Eq. (1). If we solve Eq. (6) for Q and insert this expression into Eq. (1), we obtain

$$H_D = (U^2/2g)\{K_D/[\tfrac{1}{2}\bar{V}_*^2 \eta_t \eta_{pr} + (\bar{V}_* \theta_J - 1)]^2\} \tag{8}$$

where

$$\bar{V}_* = V_*/\mathrm{U}$$

$$\theta_J = V_J/V_*$$

Combining Eqs. (6–8) we can determine the efficiency of the whole system, $\eta = P_T/P_{in}$, as

$$\eta = 2\eta_{pm} f/[\bar{V}_*^2 - 1 + K_D/f^2] \tag{9}$$

Table 1 Component design constants

Pump efficiency	$\eta_{Pm} = 0.85$
Propeller efficiency	$\eta_{Pm} = 0.65$
Turbine efficiency	$\eta_t = 0.75$
Exit velocity-ideal turbine inlet velocity ratio	$\theta_J = 0.40$

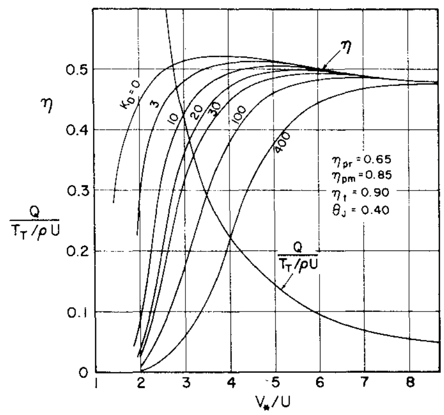

Fig. 4 Propulsive efficiency and flow rate of a turboprop system.

where

$$f = \tfrac{1}{2}\bar{V}_*^2 \eta_t \eta_{pr} + (\bar{V}_* \theta_J - 1)$$

Note that the water-jet system is a particular case of this one. In the special case where $\theta_J = 1$, $(V_J = V_*)$, no energy is delivered to the turbine and the whole thrust is obtained from the jet only. The propulsive efficiency and flow rate have been computed for various values of $\bar{V}_*$ and several K_D's with the parameters in Table 1 constant.

The results are shown in Fig. 4. As in the water-jet case, the parameter K_D is independent of the velocity ratio $\bar{V}_*$. The line $K_D = 0$ represents the efficiency of a system with no frictional losses. It reaches a maximum of $\eta = 52\%$ at $V_*/U = 3.8$. At higher velocity ratios it decreases relatively slowly and asymptotically approaches a constant value of $\eta = 41.8\%$.

When a real system is considered, frictional losses do occur, i.e., $K_D > 0$. In this case, the efficiency drops drastically at small values of $\bar{V}_*$, but approaches that of the frictionless system as the velocity ratio increases. It can be seen from Fig. 3 that for each particular value of K_D, there is a unique value of $\bar{V}_*$ at which the efficiency is maximum; the locus of maxima lies very close to the line corresponding to $K_D = 0$. In other words, for a given ducting system, regardless of its length and diameter, one can always choose a certain velocity ratio, above which the propulsive efficiency is higher than 41.8%, given the previous values of efficiency. The flow rate vs the velocity ratio is also shown.

In view of the foregoing, it is interesting to reexamine Fig. 1. One sees that the locus of maximum efficiencies of a water-jet system is well below the efficiency of a frictionless system. This means that whatever the velocity ratio $\bar{V}_J$ is, the efficiency of a real water jet system is less than $2\eta_{pm}/(1 + \bar{V}_J)$, which is itself a decreasing function that vanishes at large values of $\bar{V}_J$. Figure 4 shows that efficiencies of the order of 45 to 50% might be obtained from a relatively small weight turboprop system. This is based, however, on assumptions concerning the efficiencies of the various components.

It should be noted that the total thrust T_T is provided partly by the nozzle exhaust and partly by the thrust of propeller-nozzle combination T. If we denote by μ the ratio of these two thrusts, $\mu = T/T_T$, then the foregoing analysis results in

$$\mu = 1/[1 + 2(\theta_J \bar{V}_* - 1)/(\bar{V}_*^2 N_t N_{pr})] \tag{10}$$

System Design

To validate the various efficiencies used in the overall analysis derived above, it is necessary to consider in detail the design of each of the components. In particular, the design of the propeller-nozzle-turbine unit is the most critical component, since the turbine and the propeller must rotate at the same speed.

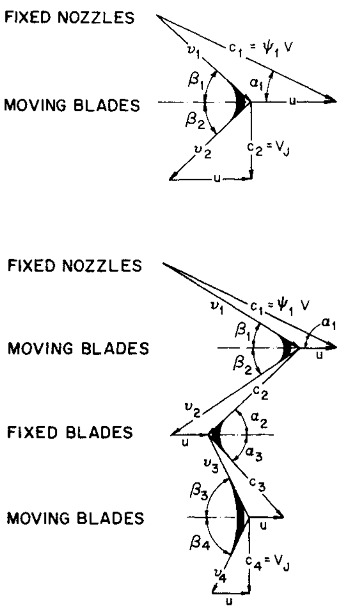

Fig. 5 Blade diagrams for one and two stage turbines: a) Single stage machine; b) Two stage machine.

Let us first consider the turbine. The type of turbine selected here is an impulse turbine where the pressure of the incoming flow is reduced to the ambient pressure in a set of fixed nozzles. The energy of the resulting high speed stream is extracted by a moving cascade of blades (a single state impulse machine) or a series of moving and fixed blades (a multiple-state or Curtis machine.). The velocity diagrams for one and two stage machines are shown in Fig. 5.

The water suffers viscous losses as it passes through each set of blades in the turbine. For the purposes of this study, we will assume that the actual inlet nozzle velocity is $0.97 V_\bullet$ (i.e. a 3% loss). We will also assume that in the other sets of blades a loss of 2% of the velocity relative to those blades occurs. We will define $\theta_u = u/V_\bullet$, the ratio of the peripheral velocity, $u = \pi n D$, of the turbine to the ideal inlet nozzle velocity, and, as before, $\theta_J = V_J / V_\bullet$. These quantities for one and two stage machines are given, along with the computed turbine efficiency η_T in Table 2. The values of θ_u were selected so that the maximum efficiency was obtained. Note also that a mechanical and windage loss of 3% for a single stage machine and 5% for a two stage machine were assumed.[3]

The propeller-nozzle combination is a very complicated hydrodynamic system, a detailed analysis of which[7] would not be appropriate to present in this feasibility study. Fortunately, there exists a wealth of literature covering model tests of propeller-nozzle systems. In particular, the results obtained at the Netherlands Ship Model Basin (NSMB) are the most extensive and are reported by van Manen,[8,9] and by Oosterveld.[5,10]

These results are presented in a standard format, in which the measured quantities are presented as functions of the advance ratio $J = U/nD$, where n is the propeller revolutions per second, and D is the propeller diameter. For each propeller type, characterized by number of blades and blade area ratio, the following quantities are plotted for several pitch-diameter ratios: $K_T = T/pn^2D^4$, the thrust coefficient; $K_Q = M/pn^2D^5$, the torque coefficient; $\eta_p = JK_T/(2\pi K_Q)$, the open water efficiency; and τ, the ratio of the propeller thrust to the total propeller + nozzle thrust. Since these results were obtained using reasonably large propeller models, we can expect that predicted performance based on these results is achievable.

Table 2 Optimum impulse turbine velocity and efficiency factors

Type	θ_u	θ_J	N_t
Single stage	0.435	0.401	0.75
Two stage	0.212	0.382	0.70

Selection of a propeller-nozzle system compatible with the rim mounted turbine is necessarily an iterative procedure. The best approach appears to be to select the ideal turbine inlet velocity ratio $\bar{V}_\bullet$, as an independent parameter. A short outline process follows. For each turbine type (one-stage or two-stage) select values of θ_u and θ_J from Table 2. The advance ratio J is then given by $J = \pi(\theta_u \bar{V}_\bullet)$ and the pitch-diameter ratio corresponding to the optimum efficiency can be selected, along with the corresponding values of K_T and K_Q. At this point, the propeller is selected in a nondimensional sense. In order to determine the exact diameter and rpm, we must determine the proportion of the total thrust produced by the propeller-nozzle combination and that by the turbine exhaust.

First, we should note that the system here is different in one essential aspect from those tested. That is, the propeller has a rim drive turbine which because of its contact with the surrounding fluid will require more torque than the corresponding hub driven propeller. If we let the area of the rim exposed to the external flow be A_r, then the additional torque created by the rim drag will be

$$\Delta M = \rho u^2 A_r D C_f / 4 \tag{11}$$

where $u = \pi n D$ is the peripheral velocity of the rim relative to the flow and C_f is the viscous drag coefficient. The increased nondimensional torque coefficient is then

$$\Delta K_Q = \Delta M / \rho n^2 D^5 = \pi^2 C_f (A_r / D^2) / 4 \tag{12}$$

Thus the efficiency of the rim driven propeller-nozzle is lower than the open water efficiency of the same system and is given by

$$\eta_{pr} = K_T / [2\pi(K_Q + \Delta K_Q)] \tag{13}$$

With this efficiency and the turbine parameters already selected, the thrust ratio μ can be determined from Eq. (10). The propeller diameter and rps are then given by

$$D = [T/(\rho K_T U^2)]^{1/2}/J$$

$$n = U/JD \tag{14}$$

The second factor which must now be considered is cavitation. The propeller characteristics selected in the above process are appropriate for a noncavitating propeller. We have presumed here that the ship is a high performance one in which a subcavitating propeller is to be used. A significant problem will certainly be minimization of cavitation in this situation. A potential solution is to use a decelerating nozzle, as proposed by van Manen[4] and reported by Oosteweld.[10] Such a nozzle slows down the fluid flowing into the propeller and thereby increases the static pressure at the propeller blade. The result is a delay of cavitation. For the purposes of this study we will assume that a decelerating nozzle is used. Oosterveld gives the following empirical criteria for determining the pressure coefficient $C_{p\min}$ on the propellers in such a nozzle

$$\begin{aligned} C_{p\min} &= (\mathrm{P}_{\min} - \mathrm{P}_\infty) / \tfrac{1}{2}\rho U_2 \\ &\approx -\pi(2.4 + 0.6Z)\tau C_T / [8(BAR + 0.2)] \\ &\quad + 1 - \{1 + (1-\tau)[1 + (1 + C_T)^{1/2}]/2\tau\}^2 \end{aligned} \tag{15}$$

where

Z = is the number of blades
C_T $= T/(\tfrac{1}{2}\rho U^2 A_p) = 8K_T/(\pi J^2)$, the thrust coefficient
A_p $= \pi D^2/4$, the propeller disk area
BAR $= A_E/A_p$, the propeller expanded area ratio

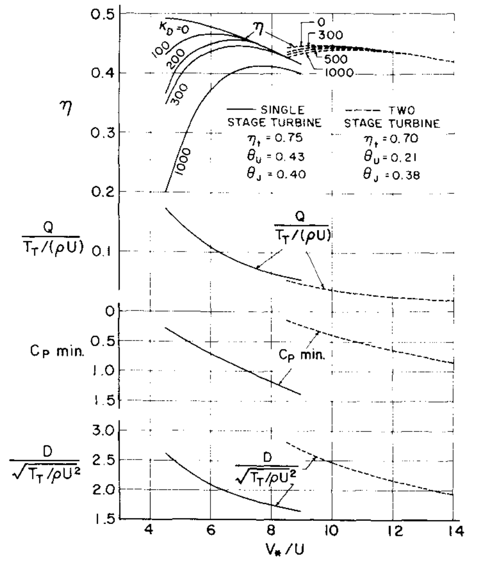

Fig. 6 Propulsive efficiency, nondimensional flow rate, minimum pressure on blades and propeller diameter for turboprop system.

For a particular design situation, we can determine the pressure of the fluid in the neighborhood of the hub (the usual reference point). To avoid cavitation, we require that the pressures created by the flow are always sensibly higher than vapor pressure. The criterion proposed by O'Brien[11] is that

$$1.2C_{p\min} \geq (p_v - p_\infty)/(\tfrac{1}{2}\rho U^2) \qquad (16)$$

As a result, the propeller characteristics determined in Eq. (14) must now be examined with regards to its cavitation performance, Eqs. (15) and (16). If it does not meet this requirement, then a propeller with a larger blade area ratio or different number of blades must be selected.

Design of the pump is simpler for this system than the previously described turbine-propeller-nozzle system since we are concerned only with the flow rate and pressure rise developed. If in addition, the rpm of the prime mover is also known, then we can form the specific speed, N_s, of the pump, given by

$$N_s = N_{pm} Q^{1/2} / H_p^{3/4} \qquad (17)$$

This number of dimensional and is usually calculated where N_{pm}, Q and H_p are in rpm, gpm, and feet, respectively. H_p can be determined from Eq. (4) and the assumed V_*, and Q can be determined from Eq. (6) and the required total thrust T_T. A typical design situation leads to specific speeds between 1000 and 3000, a range in which high efficiency centrifugal pumps can be designed. We will assume here that a pump efficiency $\eta_{pm} = 85\%$ is a typical, achievable efficiency at the design point for this kind of pump.

Using these considerations, Fig. 6 shows the performance of the total system. The test results of Oosterveld[10] were used for the $Kd-5-100$ propeller in nozzle number 31. For the turbine ring drag A_r/d^2 was chosen to be 0.03 and $C_f = 0.006$ in Eq. (12). It can be seen that a single stage turbine is best for a velocity ratio $\overline{V}_*$ between 4.5 and 9 whereas the two stage turbine is suitable for values of $\overline{V}_*$ between 7.3 and 15. The general behavior of the efficiency curves is similar to that shown in Fig. 4. The differences are due to the variation of η_{pr} with $\overline{V}_*$.

Design Example

It is illustrative to determine the characteristics of a real turboprop system. Consider, for example, a 120 ton, 40 knot hydrofoil ship. The required thrust at cruise is estimated to be

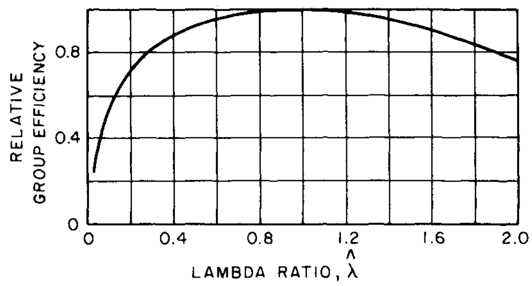

Fig. 7 Variation of group velocity with lambda ratio.

28,500 lb, and two screws are assumed. If the propeller shaft is to be submerged 5 ft below sea level then Eq. (16) becomes

$$C_{p\min} \geq -0.44$$

If the maximum allowed propeller diameter is 4 ft, the corresponding nondimensional propeller diameter

$$C_D = D/[T_T/\rho U^2]^{1/2} \leq 3.2$$

These limits restrict the velocity ratio to the intervals 4.5 to 5.1 (single state turbine) or 8.5 to 10.6 (two stage turbine). Of the maximum allowable diameter were 3 ft only, the corresponding C_D is 2.4. In this case only a single stage turbine can be used at velocity ratios between 5.0 to 5.1. The results for a two-stage turbine arrangement are given in Table 3.

For comparison, the internal ducting loss parameter $K_D = 500$ corresponds to the equivalent of a 4¾ in. internal diameter pipe approximately 30 ft long. An allowance has been made for the necessary bends in this pipe.

Off Design Performance

The performance of the propulsion system at off-design conditions and especially at take-off is a crucial factor in high performance craft. A typical hydrofoil takes off at about half of its design speed where its drag is about the same or perhaps even somewhat more than that at design speed. To accelerate the ship past this condition of very high drag in a reasonable time the propulsion system must have a thrust capability with considerable margin at take-off. Since propulsion systems are usually designed for optimum performance at the design speed, the efficiency at such an off-design condition may be lower, and the power required may be large. The loading on the propeller is much higher and cavitation problems are probable at take-off. Only a small fraction of the design ram pressure is available, and the pump may cavitate as well. Clearly, all of these factors must be addressed if a real design is to be performed.

Calculation of the system performance at an off-design condition requires a knowledge of the off-design characteristics of each of the components, such as the propeller-nozzle, the turbine and the pump. In the previous design example, the test results of Oosterveld were used, and these included a full range of operating conditions.

The take-off performance of typical turbines is well known and may be estimated by the "lambda ratio", $\hat{\lambda}$, which is the ratio of $\lambda = \theta_u^2$ at the off-design condition to that at the optimum efficiency point (the design point). Figure 7 reproduced from Ref. 12 shows the influence of $\hat{\lambda}$ on $\hat{\eta}_t$,‡ the ratio of the off-design efficiency to the optimum efficiency.

No such standard information appears to be available for pumps since the off-design performance depends very much on the actual configuration of the pump (centrifugal, Francis-

‡In this development we will adopt the convention that a circumflex (ˆ) over a variable will imply the ratio of the variable at the off-design condition to that at the design point.

Table 3 System characteristics for a 14,250# thrust unit for a design speed of 40 knots

Selected velocity ratio	$V_\cdot/U$	$=9$
Selected friction constant	K_D	$=500$
Propulsive efficiency	η	$=44\%$
Advance ratio	J	$=1.646$
Pitch diameter ratio	P/D	$=1.752$
Propeller diameter	D	$=3.38$ ft
Propeller rpm	N_{pr}	$=730$ rpm
Flow rate	Q	$=5.08$ cfs $=2,280$ gpm
Total pump head	H_p	$=5,750$ ft
Selected pump rpm	N_{pm}	$=6,000$ rpm
Pump specific speed	N_s	$=1,225$ (4 stages are assumed)

type, propeller, etc.). Figure 8 shows a typical set of characteristics for a centifugal pump like that selected for the design example. It relates the variation of head, flow rate and efficiency to the rpm of the pump.

It appears best to start the off-design analysis at the pump. If we consider the water path which passes through the pump, we see that the losses are the viscous head loss in the ducting and the kinetic energy loss represented by the fluid escaping through the turbine inlet nozzle. These losses must be balanced by the head developed by the pump and that recovered from the freestream. Referring to Eq. (4), the relative head required $\hat{H}_p$ can be expressed

$$\hat{H}_p = \{\hat{V}_\cdot^2 H' + H_D'\}\hat{Q}^2 - H'\hat{U}^2 \tag{18}$$

where the following quantities are to be evaluated at the original design point

$$H' = U^2/(2gH_p)$$

$$H_D' = H_D/H_p$$

This relation between $\hat{H}_p$ and $\hat{Q}$ for the design example is also shown on Fig. 8 for $\hat{U}=0$ and $\hat{U}=1.0$. The small difference between these two curves is due to the fact that for this system the static head, $U^2/2g$, is very small compared to the head developed by the pump. We will assume that the pump rpm is externally controllable. For any given pump speed and ship speed, the values of $\hat{H}_p$, $\hat{Q}$, and $\hat{\eta}_{pm}$ can be determined from Fig. 8 for the design example or from an equivalent figure for other specific designs.

The rotative speed of the propeller-turbine combination results from a balance between the input from the turbine and the power expended by the propeller. Solution for this rotative speed is an iterative process. The following relations can be easily determined

$$\hat{\lambda} = (\hat{n}\hat{Q})^2 \tag{19a}$$

$$\hat{J} = \hat{U}/\hat{n} \tag{19b}$$

If an $\hat{n}$ is chosen then the first of these relations (together with the previously determined $\hat{Q}$ determines the lambda ratio. Figure 7 then yields the relative turbine efficiency $\hat{n}_t$ and the power delivered to the propeller by the turbine can now be determined. The second relation yields $\hat{J}$. From the propeller characteristics (together with the pitch-diameter ratio), new values of K_T and K_Q can be determined.

With this new value of K_Q and the modification for the ring drag Eq. (12), the power absorbed by the propeller can be determined. If the turbine delivers more power than the propeller is absorbing, a larger $\hat{n}$ must be selected; and, conversely, if the turbine delivers less power than the propeller absorbs, a smaller $\hat{n}$ must be selected. The iteration appears to converge rapidly. When convergence is obtained the propeller + nozzle thrust is readily determined from the corresponding K_T. The jet exhaust thrust is determined from

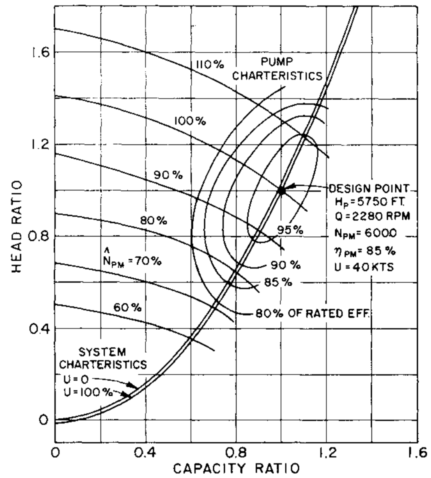

Fig. 8 Typical centrifugal pump characteristics.

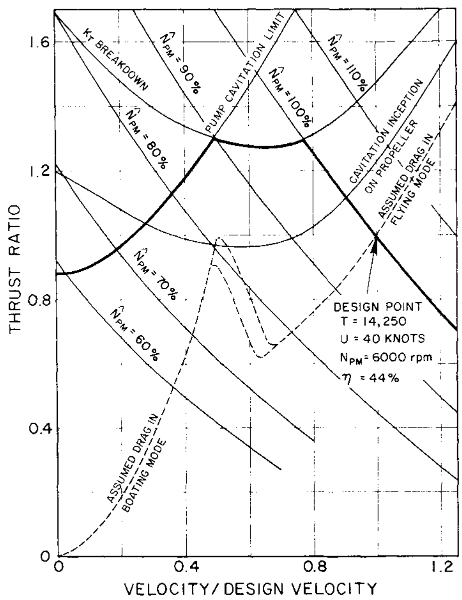

Fig. 9 Thrust vs speed—predicted performance for a 120 ton, 40 knot hydrofoil craft.

Q, θ_J, and $\overline{V}_\cdot$. The results shown in Fig. 9 were derived for the design example using this procedure, for various $\hat{N}_{\rm pm}$ values and various $\hat{U}$ values. As mentioned earlier, cavitation might occur on both propeller and pump. Suction specific speed, N_{sv} may be used as pump cavitation criterion.

$$N_{sv} = N_{pm}Q^{1/2}/H_{sv}^{3/4} \tag{20}$$

where H_{sv} denotes net positive suction head at pump inlet in feet fluid; and Q and N_{pm} are in gpm and rpm, respectively. Cavitation risk exists when N_{sv} exceeds a certain value, which depends on the particular pump used. Typically it is of the order of 10,000 to 14,000 for most water pumps. At any ship velocity there is a certain resulting N_{sv} which may not be exceeded without causing cavitation of the pump. A curve corresponding to $N_{sv_{\rm max}} = 11,200$ is shown in Fig. 9.

The minimum pressure coefficient on propeller blades and its value as cavitation inception also may be determined. The thrust at which cavitation inception on propeller blades occurs may, therefore, be determined for any ship velocity. This is also shown in Fig. 9 for the design example propeller. Cavitation inception, however, is too conservative a limit for short durations such as the take-off mode which usually takes one minute or less. The "K_T break-down" limit seems more adequate for such cases. For any given operating condition the cavitation number corresponding to thrust drop off is approximately 20% less than that for inception. Thus it seems reasonable to consider

$$C_{p\min} \geq 1.2(p_v - p_\infty)/\tfrac{1}{2}\rho U^2 \tag{21}$$

and adequate cavitation limit when operating at the hump condition, and this "K_T break-down" curve is also shown on Fig. 9.

Weight Comparison

It is of interest to compare the weight and the efficiency of the design example to those of a corresponding water-jet system, providing the same thrust at the design speed. In general it is very difficult to determine the weights of the individual components of either such systems, since for the most part, the components are not off-the-shelf items. If we restrict our view to the case where the water jet and the propjet have the same efficiency, then it is reasonable to assume that the identical prime mover can be used for each. Differences in weight therefore occur due to the following elements: 1) reduction gears; 2) pump and propeller; 3) ducting materials; and 4) the water in the ducting. The turboprop uses a pump which is, by comparison with the water-jet, run at a high rpm but may be somewhat heavier due to the higher internal pressures. The higher pump rpm eliminates the gear reduction system, but this, in turn, is balanced by the need for an external propeller. It appears that these competing qualities lead to machinery combinations of the same weight for either system.

Major differences do occur in the last two elements. The example turboprop system has an overall efficiency of 44% at the design point. If the same pump efficiency is used, the equivalent water-jet system requires a duct loss parameter, $K_{DJ}=1$ (see Fig. 1). For the turboprop system we found $K_D=500$. A typical water-jet system will require about one-half the ducting length that a turboprop system would, since the jet can be exhausted above the water. If the well-known Darcy-Weisbach formula for the ducting head loss is used (see Eq. (4), p. 678, [12]) then the weight of water in the water-jet ducting W_J is related to that in the turboprop ducting W by

$$(W_J/W) = (L_J/L)^{1.4}(K_D/K_{DJ})^{.4} \tag{22}$$

for ducts of equal roughness and where L and L_J are the equivalent lengths of the corresponding ducts, including effects for bends, etc. Using the values of K_D and K_{DJ} previously mentioned and assuming that $L_J=L/2$, we have $W_J/W=4.55$.

In other words, the equivalent water-jet system has a much greater amount of water in the ducts than the turboprop system. Further, since the water-jet ducting has a significantly larger inner diameter, the weight of the ducting is also much larger for this system. Preliminary estimates indicate that the example turboprop system has approximately 1000# per unit less weight in the ducting and water board than that of a comparable water-jet system.

Conclusions

It has been shown that propulsive efficiencies of the order of 43 to 46% are obtainable from a small weight turboprop system. These calculations were based on realistic evaluations of the pump, propeller, and turbine performance using experimental data and manufacturers specifications where possible. Based on this study the turboprop system appears to be an alternate to the water-jet system. Although no more efficient, the turboprop system possesses a number of distinct advantages over an equivalent water-jet system. These can be summarized as follows: 1) The turboprop requires less water weight on board; 2) The turboprop efficiency is virtually independent of the recovery of static head within the ducting unlike the water-jet (see Fig. 8); 3) The turboprop produces a much higher thrust at take-off than a fixed-inlet water-jet system; and 4) The pump for a turboprop system operates at a higher rpm and requires no reduction gearing. Balancing these advantages are the fact that the turboprop system is somewhat more complex than the water-jet system and, of course the system is not yet proven.

It is possible that with careful design improvement efficiencies approaching 50% can be achieved with the turboprop system. These might be:

1) A different nozzle-propeller combination can be used for this system. The combination selected for this study (a flow decelerating nozzle and a Kaplan propeller) was chosen primarily because the performance data were available. Nozzles and propellers more amenable to the turboprop may lead to higher efficiencies.

2) It might be possible to support the propeller by means of a ring bearing mounted within the propeller nozzle rather than to use a hub and struts. In addition to the removal of the drag of the hub and struts, this would provide a nearly uniform flow to the propeller, further reducing the probability of intermittent cavitation.

3) The water jet exhaust from the turbine can be deflected to change the circulation around the propeller nozzle. In this way a fixed nozzle can be made to either accelerating or decelerating as required for cavitation prevention or for increased propeller efficiency.

References

[1]Myers, G.R., "Observations and Comments on Hydrofoils," paper presented at the Spring Meeting of SNAME, May 1965.

[2]Kim, H. C., "Hydrodynamics Aspects of Internal Pump-Jet Propulsion," *Marine Technology,* Vol. 3, No. 1, Jan. 1966, pp. 80-98.

[3]Nechleba, M., "Hydraulic Turbines," translated from the Czech edition, Constable & Co., London, 1957.

[4]Van Manen, J. D., "Analysis of Ducted Propeller Design," *Transactions of SNAME,* Vol. 74, 1966, pp. 522-562.

[5]Oosterveld, M. W. C., "Wake Adapted Ducted Propeller," Publication No. 345, Netherlands Ship Model Basin.

[6]Chen, C. F., "Shrouded Supercavitating Propellers," *Fourth Symposium of Naval Hydrodynamics, Office of Naval Research, ACR-92*, pp. 339-356.

[7]Morgan, W. B., "Theory of the Annular Airfoil and Ducted Propeller," *Fourth Symposium of Naval Hydrodynamics, Office of Naval Research ACR-92*, pp. 151-197.

[8]Van Manen, J. D., "Recent Research of Propellers in Nozzles," *International Shipbuilding Progress*, Vol. 4, No. 36, Aug. 1957, pp. 395-424.

[9]Van Manen, J. D., "The Design of Screw Propellers in Nozzles," *International Shipbuilding Progress*, Vol. 6, No. 55, Mar. 1959, pp. 95-113.

[10]Oosterveld, M. W. C., "Model Tests with Decelerating Nozzles," NSMB Publication No. 309, *International Shipbuilding Progress,* Vol. 15, No. 6, June 1968, pp. 206-220.

[11]O'Brien, J., "The Design of Marine Screw Propellers," Hutchinson & Co. Ltd. (publishers), London, 1969.

[12]*Marine Engineering*, edited by R. L. Harrington, *SNAME*, 1971.