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COMPARISON OF THREE THEORIES OF WATER-JET PROPULSION

John H. Garrett

U. S. Navy Marine Engineering Laboratory
Annapolis, Maryland

February 1967

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Comparison of Three Theories
of Supercavitating Propellers

Assignment 55 501
Technical Memorandum 484/66
February 1967

By
Carl H. Gerhart

U. S. NAVY

MARINE ENGINEERING LABORATORY

Annapolis, Md.

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Comparison of Three Theories
of Water-Jet Propulsion

Assignment 55 501
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ABSTRACT

The three different approaches to derivation of formulae expressing the relations among speed, thrust, power, and efficiency of water-jet propulsion systems, as developed by Lockheed California Company; Virgil Johnson of Hydronautics, Incorporated; and Joseph Levy of Aerojet-General Corporation, are summarized and compared. Certain modifications and simplifications are incorporated, and terminology is modified as necessary to facilitate comparison.

The Lockheed system, which provides a method for including the weight and drag of the propulsion system in the optimization procedure, appears to be the more useful.

The problems of compromising the performance of the propulsion system at cruising speed in order to provide reasonable hump performance are briefly discussed

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COMPARISON OF THREE THEORIES OF WATER-JET PROPULSION

1.0 INTRODUCTION

The U. S. Navy Marine Engineering Laboratory is assisting the David Taylor Model Basin in the machinery aspects of the surface effects ship and hydrofoil programs. Both of these programs may utilize water jets as the main propulsion method. In order to establish a base from which to initiate machinery system layouts as well as to identify areas requiring further development, a review of some of the principles of water-jet propulsion was undertaken. This is a summary of previous papers on the subject together with suggestions for further work. A list of the nomenclature used for this study is contained in Appendix A.

1.1 Considerations. Development of the preliminary design of a water-jet system involves the following steps as a minimum:

- Devising a rationale for optimizing the V_j/V_o ratio. The important factor is the method of incorporating various losses chargeable against the propulsion system, since these are primary considerations in determining the optimum jet velocity.

- Consideration of the compromises required to provide adequate performance at the hump condition, while retaining high efficiency at cruise.

- Determination of the relation between performance parameters developed under the two above items and pump design.

- Optimization of the system by iteration procedures.

1.2 Comparisons. The three studies,[1,2,3] provide three different methods for optimizing the V_j/V_o ratio. These are summarized and compared modifying the approach and the nomenclature of the originals where necessary to facilitate comparison.

2.0 WATER-JET PROPULSION FUNDAMENTALS

2.1 Ideal. The same basic approach to developing the theory of water-jet propulsion is adopted in all reviews of the subject. Considering first an ideal system, that is, one in which there are no internal energy losses, the thrust produced by the jet is equal to the increase in momentum of the water:

$$T = \rho Q (V_j - V_o). \qquad \text{..... (1)}$$

[1]Superscripts refer to similarly numbered entries in Appendix B.

The useful work done on the vessel is equal to the product of the thrust and the velocity of the vessel:

$$\text{Useful work} = T\,V_o. \qquad \ldots\ldots (2)$$

The power output of the pump is equal to the rate of increase of kinetic energy of the water passing through the system:

$$P_{pump} = \frac{\rho Q}{2}\,(V_j^{\,2} - V_o^{\,2}), \qquad \ldots\ldots (3)$$

or, expressed in terms of head produced by the pump:

$$P_{pump} = \rho Q g H_{ideal} \qquad \ldots\ldots (4)$$

The ideal propulsive efficiency is equal to the ratio of the useful work done on the ship to the power supplied by the pump:

$$\eta_{pi} = \frac{T V_o}{P} \qquad \ldots\ldots (5)$$

$$= \frac{\rho Q\,(V_j - V_o)\,V_o}{\frac{\rho Q}{2}\,(V_j^{\,2} - V_o^{\,2})} \qquad \ldots\ldots (5a)$$

$$= \frac{2}{1 + \frac{V_j}{V_o}}. \qquad \ldots\ldots (6)$$

Equating Equations (3) and (4) shows that, as would be expected, the potential energy of the water, as represented by the pump head, is ideally equal to the kinetic energy of the water in the jet, less the kinetic energy of the intake water:

$$\frac{\rho Q}{2}\,(V_j^{\,2} - V_o^{\,2}) = \rho Q g H_{ideal}$$

or

$$H_{ideal} = \frac{V_j^{\,2} - V_o^{\,2}}{2g}.$$

The shaft power input is equal to the power output of the pump (Equation (3)), divided by the pump efficiency:

$$P = \frac{\rho Q}{2\eta_{pu}} (V_j^2 - V_o^2), \qquad \text{..... (7)}$$

or, using Equation (4),

$$P = \frac{\rho Q g H_{ideal}}{\eta_{pu}}. \qquad \text{..... (7a)}$$

2.2 Actual. In an actual system, the power requirement is increased by the amount required to overcome the frictional losses of the water in the inlet, ducts, and nozzle. It is convenient to express these losses in the form of the additional head required to offset the losses. As the losses are likely to be roughly proportional to the square of the velocity of the water through the system, the loss is equal to a constant times some velocity head.

$$H_L = k \frac{V_x^2}{2g}.$$

The choice of V_x varies among the three studies being reviewed. Johnson uses the ship velocity, V_o; Levy[2] uses the jet velocity, V_j; and Lockheed[3] uses the velocity of the water in the inlet duct, V_i. The value of K varies, depending on which velocity is selected as the measure of the loss.

In any case, the equations for the ideal system are modified to represent the actual system by adding to the head supplied by the pump the additional head required to overcome the loss.

Thus,

$$H = H_{ideal} + H_L$$

$$= \frac{V_j^2 - V_o^2}{2g} + k \frac{V_x^2}{2g} . \qquad \text{..... (8)}$$

If V_j is selected as the measure of internal loss, Equation (8) becomes

$$H = \frac{(1+k) V_j^2 - V_o^2}{2g} \qquad \text{..... (9)}$$

If V_o is selected,

$$H = \frac{V_j^2 - (1 - k) V_o^2}{2g} \qquad \text{..... (10)}$$

If V_i is selected,

$$H = \frac{V_j^2 + k V_i^2 - V_o^2}{2g} \qquad \text{..... (11)}$$

If the difference of elevation between the outside water level and the nozzle exit level is substantial, the energy required to elevate the water may be included in the equation as an additional pump input, so that

$$H = \frac{V_j^2}{2g} + k \frac{V_x^2}{2g} + h - \frac{V_o^2}{2g} . \qquad \text{..... (12)}$$

It should be noted that, in all of these equations, the "head" is a measure of energy with the dimension foot-pounds per pound of water and is not simply a dimension, feet.

Substituting the value of H from Equation (12) into Equation (4) and dividing by the pump efficiency to obtain an expression for shaft power input,

$$P = \frac{\rho Q g}{\eta_{pu}} \left(\frac{V_j^2}{2g} + k \frac{V_x^2}{2g} + h - \frac{V_o^2}{2g} \right)$$

$$= \frac{\rho Q}{2\eta_{pu}} (V_j^2 + k V_x^2 + h - V_o^2). \qquad \text{..... (13)}$$

Substituting this value for P in Equations (5) and (5a),

$$\frac{\eta_{pi}}{\eta_{pu}} = \frac{2 (V_j - V_o) V_o}{V_j^2 + k V_x^2 + h - V_o^2} . \qquad \text{..... (14)}$$

Dividing the numerator and denominator of Equation (14) by V_o^2, an expression with the jet velocity ratio V_j/V_o as the variable is formed, Equation (14) becomes

$$\frac{\eta_{pi}}{\eta_{pu}} = \frac{\left(\frac{V_j}{V_o} - 1 \right)}{\frac{V_j^2}{V_o^2} + k \frac{V_x^2}{V_o^2} + \frac{h}{V_o^2} - 1} \qquad \text{..... (15)}$$

A plot of this equation for various values of k provides a series of curves with shape similar to those of Figure 1. The optimum V_j/V_o ratio can either be found graphically from the plot or numerically by differentiating Equation (15).

Another expression for efficiency using H instead of $V^2/2g$ is secured from Equation (7a).

$$\frac{\eta_{pi}}{\eta_{pu}} = \frac{\rho Q (V_j - V_o) V_o}{\rho Q g H}$$

$$= \frac{(V_j - V_o) V_o}{g H} . \qquad \ldots\ldots (16)$$

It should be recognized that, in an actual system, the summation of losses contributed by each of the components over a range of operating conditions cannot be precisely evaluated by any such expression as k $(V_x^2/2g)$. To analyze the losses of a specific design, it is necessary to calculate the loss to be expected in each element under an appropriate range of parameters and then to optimize the system by iteration. Computer programming is almost mandatory to accomplish this with any precision within a reasonable allocation of engineering manpower.

3.0 SUMMARIES

With this as a foundation, it is in order to proceed to a review of the differences in the detailed approach taken by the respective authors of the references:

3.1 "Water Jet Propulsion for High Speed Hydrofoil Craft." Johnson's[1] basic assumption is that the summation of internal energy losses (except nozzle loss) is related to V_o; that is,

$$H_L = k \frac{V_o^2}{2g} .$$

Another factor, K, includes both the elevation and the loss heads. Thus,

$$K = k + \frac{2gh}{V_o^2} .$$

For convenience in developing the equations, Johnson uses the factor, H*, defined as the ratio of head produced by the pump to the dynamic head,

$$H^* = \frac{2gH}{V_o^2} .$$

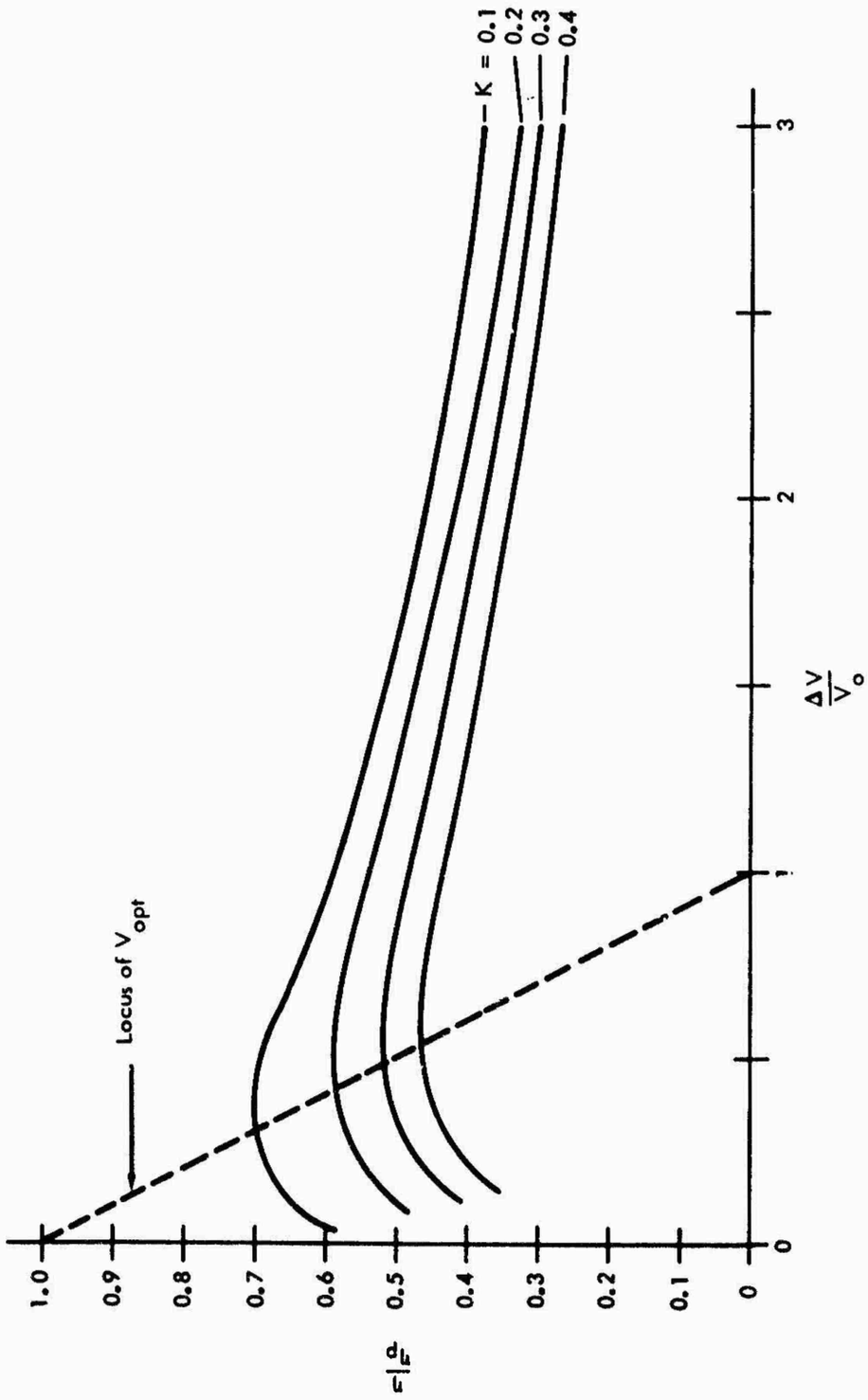

Figure 1
Relation Between Propulsive Efficiency and Velocity Ratio for Various Loss Factors

If Equation (12) is solved for V_j, the following expression can be obtained:

$$V_j = V_o \left(1 - k + \frac{2gH}{V_o^2} - \frac{2gh}{V_o^2}\right)^{\frac{1}{2}} \qquad \text{..... (17)}$$

Substituting Equation (17) in (16):

$$\frac{\eta_{pi}}{\eta_{pu}} = 2 \left[\left(1 - k + \frac{2gh}{V_o^2} - \frac{2gH}{V_o^2}\right)^{\frac{1}{2}} - 1\right] \frac{V_o^2}{2gH} .$$

Substituting K and H* (defined above), this equation simplifies to

$$\frac{\eta_{pi}}{\eta_{pu}} = \frac{2\left[(1 - K + H^*)^{\frac{1}{2}} - 1\right]}{H^*} . \qquad \text{..... (18)}$$

This is plotted in Figure 2 for the range of H* from zero to five and for several values of K. Values of H* beyond five are generally not of interest (except perhaps at or below hump speed, where V_o is relatively low). By differentiating Equation (18), the following expression for $H^*_{optimum}$ results:

$$H^*_{opt} = 2K + 2\sqrt{K} . \qquad \text{..... (19)}$$

The locus of H^*_{opt} is also shown in Figure 2.

Equation (19) implies that

$$H_{opt} = \frac{V_o^2}{g} (K + \sqrt{K}) . \qquad \text{..... (20)}$$

A plot of H_{opt} against V_o for various values of K is shown in Figure 3.

An expression for $V_{j\,opt}$ in terms of V_o and K can be derived from Equation (20),

Since

$$\frac{V_{j\,opt}^2}{2g} = H_{opt} - H_L - h + \frac{V_o^2}{2g} ,$$

$$\frac{V_{j\,opt}^2}{2g} = H_{opt} - K\frac{V_o^2}{2g} + \frac{V_o^2}{2g}$$

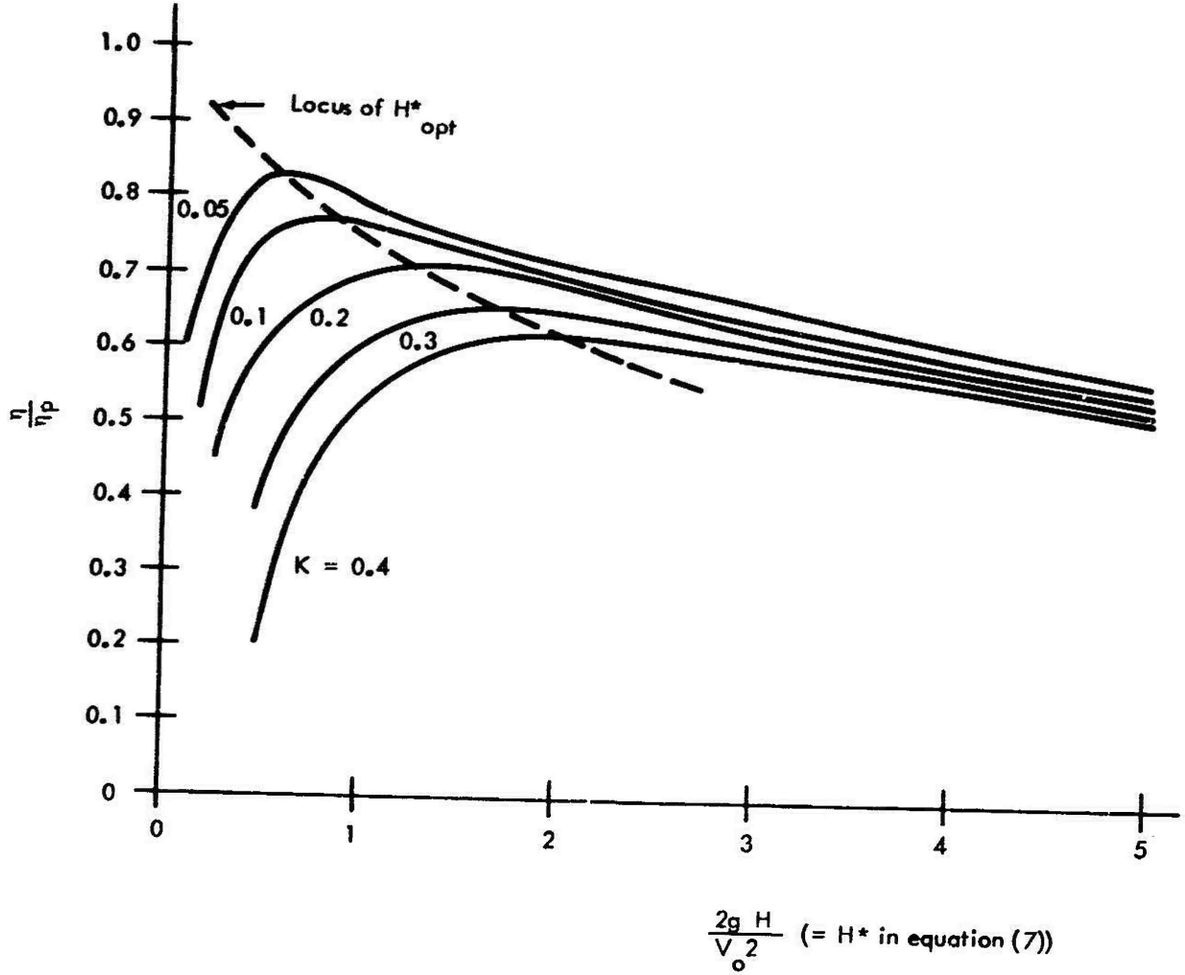

Figure 2
Relation Between Propulsive Efficiency and Pump/Velocity Head Ratio for a Range of Loss Factors (Johnson Method)

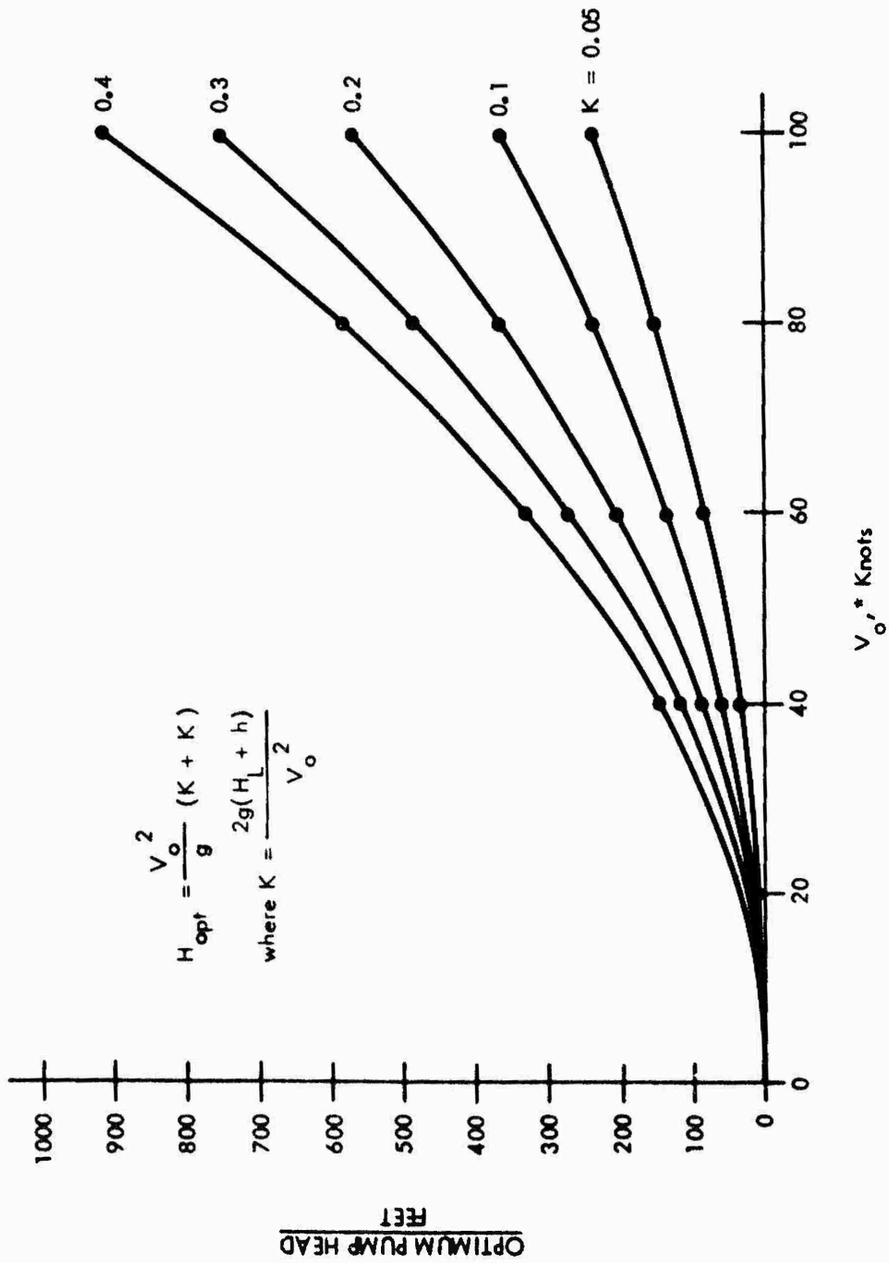

Figure 3
Relation Between Optimum Pump Head and Ship Speed
for a Range of Internal Energy Loss Factors (Johnson Method)

Substituting the value of H_{opt} from Equation (19),

$$\frac{V_j^2\ opt}{2g} = \frac{V_o^2}{g}\ (K + \sqrt{K}\ ) - \frac{V_o^2}{2g} K + \frac{V_o^2}{2g}.$$

This reduces to

$$\frac{V_j\ opt}{V_o} = \sqrt{K + 2\sqrt{K} + 1} \qquad \ldots\ldots (21)$$

Figure 4 is a plot of Equation (21) for various values of K.

Johnson suggests that the minimum practical value of K for hydrofoil vessels will fall in the range 0.15 to 0.34, depending upon the size and speed.

3.2 "The Design of Water-Jet Propulsion Systems for Hydrofoil Craft." Levy[2] makes the basic assumption that the summation of internal energy losses through the system bears a fixed relationship to the energy represented by the jet velocity, $V_j^2/2g$. He adopts somewhat different terminology, in that he uses a term, k (here called r to avoid confusion with the loss factor k), such that

$$r = \frac{\Delta V}{V_o} = \frac{V_j - V_o}{V_o} = \frac{V_j}{V_o} - 1$$

From Equation (3), neglecting internal energy losses,

$$P = \frac{\rho Q}{2\eta_{pu}}\ (V_j^2 - V_o^2)$$

Since

$$V_j^2 = (V_o + \Delta V)^2$$

$$= V_o^2\ (1 + r)^2$$

$$P = \frac{\rho Q V_o^2}{2\eta_{pu}} \left[(1+r)^2 - 1\right] \qquad \ldots\ldots (22)$$

$$= \frac{\rho Q V_o^2 r}{2\eta_{pu}}\ (r + 2). \qquad \ldots\ldots (23)$$

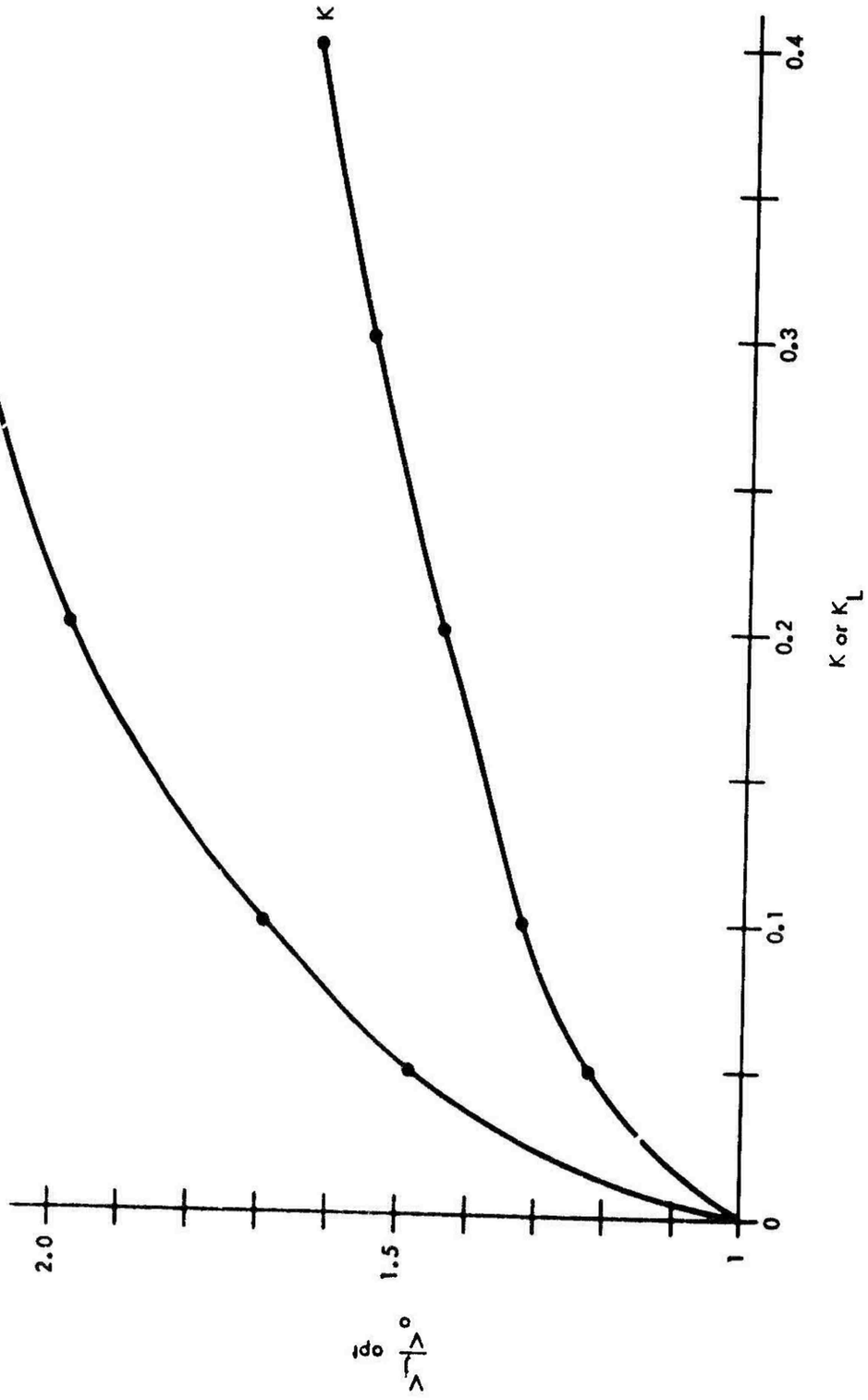

Figure 4
Optimum V_j/V_o Ratio in Relation to Loss Factor

The ideal propulsion efficiency, $T V_o/P$, is then

$$\frac{\eta_{pi}}{\eta_{pu}} = \frac{\rho Q V_o^2 r}{\frac{\rho Q V_o^2 r (r+2)}{2}} \qquad \text{..... (24)}$$

$$= \frac{2}{2+r}. \qquad \text{..... (25)}$$

In order to take account of internal energy losses in the intake, ducts, and nozzle, a head loss coefficient K_L, is adopted such that

$$H_L = K_L \frac{V_j^2}{2g}$$

$$= K_L \frac{V_o^2}{2g} (1+r)^2.$$

Taking account of the internal energy losses, the power input, Equation (22) is modified as follows:

$$P = \frac{\rho Q V_o^2}{2\eta_{pu}} \left[(1+r)^2 (1+K_L) - 1\right].$$

Substituting this value for P in Equation (24),

$$\frac{\eta_{pi}}{\eta_{pu}} = \frac{2r}{K_L + 2(1+K_L)r + (1+K_L)r^2} \qquad \text{..... (26)}$$

The optimum velocity ratio found by differentiating Equation (26) is

$$r_{opt} = \left(\frac{K_L}{1+K_L}\right)^{\frac{1}{2}}. \qquad \text{..... (27)}$$

Equation (26) is plotted in Figure 1 for a range of K_L from 0.1 to 0.4.

By substituting $V_{j_{opt}} - V_o/V_o$ for r_{opt} in Equation (27) and noting that

$$H = (1 + K_L) \frac{V_j^2}{2g} - \frac{V_o^2}{2g}, \qquad \text{..... (28)}$$

an expression for optimum pump head in terms of V_o and K_L can be derived as follows:

$$\frac{V_{j\,opt} - V_o}{V_o} = \left(\frac{K_L}{1 + K_L}\right)^{\frac{1}{2}}$$

$$V_{j\,opt} = V_o \left[\left(\frac{K_L}{1 + K_L}\right)^{\frac{1}{2}} + 1\right]. \qquad \text{..... (29)}$$

Substituting this expression for $V_{j\,opt}$ into Equation (28) results in the desired relationship

$$H_{opt} = \frac{V_o^2}{2g} \left[(1 + K_L)\left(\left[\frac{K_L}{1 + K_L}\right]^{\frac{1}{2}} + 1\right)^2 - 1\right]. \qquad \text{..... (30)}$$

A plot of Equation (30) for several values of K_L is shown in Figure 5. A plot of optimum V_j/V_o ratio as a function of K_L (Equation (29)) is shown in Figure 4.

3.3 "Waterjet Propulsion System Study." In Appendix A of Volume 5 of the Lockheed study[3] a method is derived for establishing the optimum jet velocity for a water-jet propulsion system for a hydrofoil craft. The method depends upon knowledge of the various drag and loss parameters associated with the propulsion system and use of the concept of a basic ship, which is the vessel which would be required to carry the desired payload and fuel if the weight of the necessary propulsion system were zero. The propulsion system is then charged with the additional drag resulting from the growth in size of the vessel required to carry the propulsion plant, as well as additional drag due to propulsion appurtenances, such as underwater struts or scoops.

The overall efficiency of the propulsion plant is defined as the ratio of useful work done on the basic ship to the shaft power output of the engine.

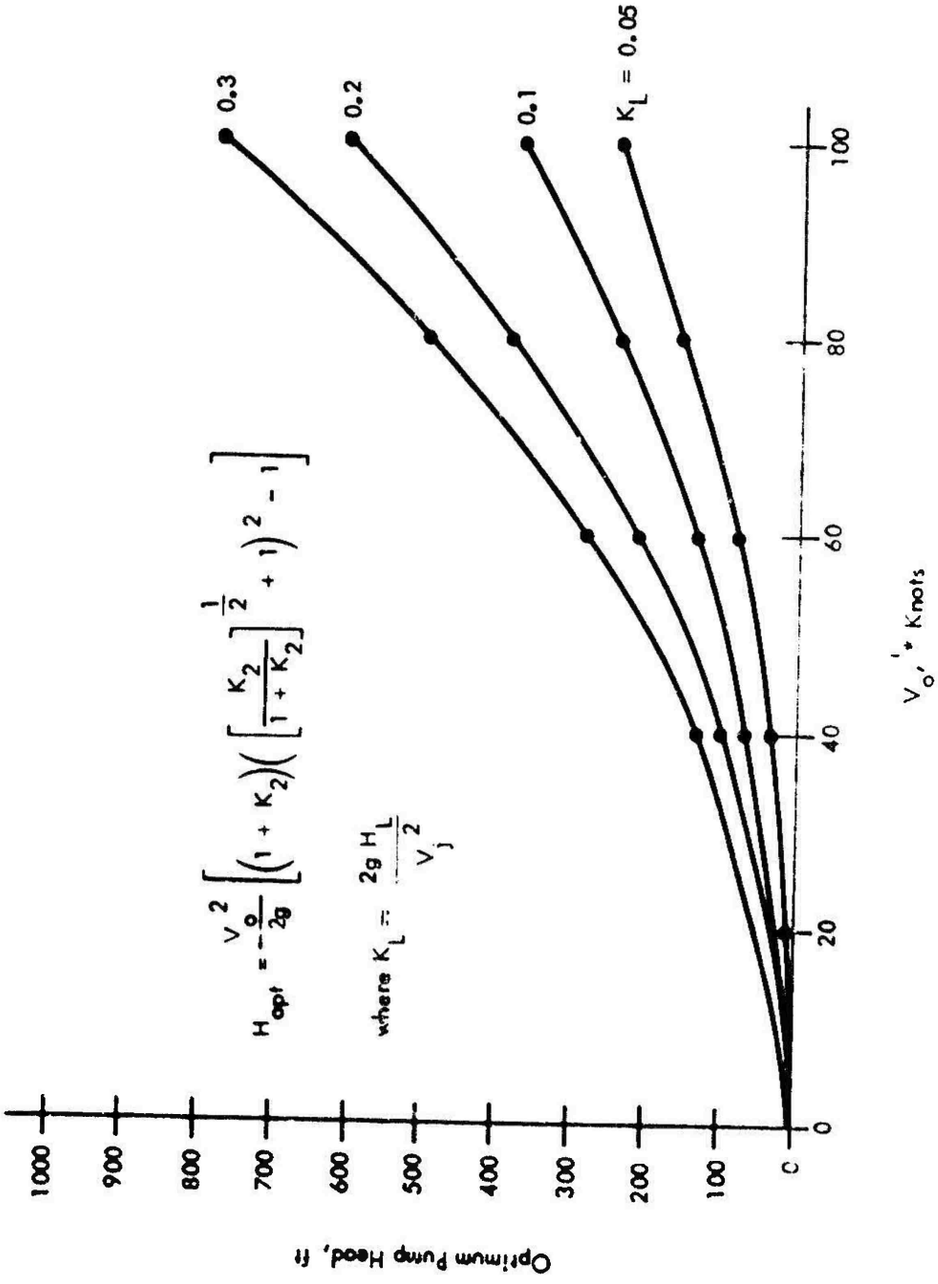

Figure 5
Relation Between Optimum Pump Head and Ship Speed
for a Range of Internal Energy Loss Factors (Levy Methal)

$$\eta_o = \frac{T_b V_o}{P}. \qquad \text{..... (31)}$$

The power required to propel the complete ship is TV. The difference between $T V_o$ and $T_b V_o$ is the power required to propel the propulsion system weight and to overcome the additional drag due to propulsion system appurtenances.

The overall efficiency can be divided into three elements, η_{pi}, η_{pu}, and η_s, such that

$$\eta_o = \eta_{pi} \eta_{pu} \eta_s, \qquad \text{..... (32)}$$

where η_{pi} is the ideal propulsive efficiency as defined by Equations (5), (5a), and (6); η_{pu} is the pump efficiency; and η_s is the "system" efficiency, the definition of which will appear below.

The internal energy balance of the water-jet system requires that the net increase in energy of the water in passing through the system is equal to the energy added by the pump less the summation of internal energy losses and the difference in elevation between the mean water level and the nozzle.

$$\frac{Q\rho}{2} (V_j^2 - V_o^2) = Q\rho g (H - H_L - h),$$

or

$$V_j^2 - V_o^2 = 2g (H - H_L - h). \qquad \text{..... (33)}$$

The external energy balance requires that the net rate of increase in energy of the water, multiplied by the propulsive efficiency, is equal to the work done on the ship.

$$\eta_{pi} \frac{Q\rho}{2} (V_j^2 - V_o^2) = T_b V_o + T_d V_o. \qquad \text{..... (34)}$$

Since, from Equation (34),

$$T_b V_o = \eta_{pi} \frac{Q\rho}{2} (V_j^2 - V_o^2) - T_d V_o,$$

Equation (31) can be transformed to

$$\eta_o = \frac{\eta_{pi} \frac{Q\rho}{2} (V_j^2 - V_o^2) - T_d V_o}{P}.$$

Since, by definition,

$$P = \frac{Q\rho g H}{\eta_{pu}} ,$$

$$\eta_o = \frac{\eta_{pu}\,\eta_{pi}\,\frac{Q\rho}{2}\,(V_j^{\,2} - V_o^{\,2}) - \eta_{pu}\,T_d\,V_o}{Q\rho g H}$$

$$= \eta_{pu}\,\eta_{pi}\,\frac{V_j^{\,2} - V_o^{\,2}}{2gH} - \frac{\eta_{pu}\,T_d\,V_o}{Q\rho g H}$$

Substituting Equation (33)

$$\eta_o = \eta_{pu}\,\eta_{pi}\left(\frac{H - H_L - h}{H}\right) - \frac{\eta_{pu}\,T_d\,V_o}{Q\rho g H} .$$

This can be written as

$$\eta_o = \eta_{pu}\,\eta_{pi}\left(1 - \frac{H_L + h}{H} - \frac{T_d\,V_o}{\eta_{pi}\,Q\rho g H}\right). \qquad \text{..... (35)}$$

The term between the parentheses involves the various internal and external energy losses attributable to the water-jet propulsion system and can therefore appropriately be called "system" efficiency, η_s.

From Equation (33),

$$H = \frac{V_j^{\,2} - V_o^{\,2}}{2g} + H_L + h.$$

Substituting this value for H in the expression for η_s in Equation (35),

$$\eta_s = \frac{\frac{V_j^{\,2} - V_o^{\,2}}{2g} - \frac{H_d}{\eta_{pi}}}{\frac{V_j^{\,2} - V_o^{\,2}}{2g} + H_L + h} . \qquad \text{..... (36)}$$

In its report, Lockheed marshalls support for the assumption that the difference between the power required to propel the actual ship and the basic ship (that is, the power required to offset the added weight and drag of the propulsion system) bears nearly a linear relationship to the weight of water passing through the system. It is also reasonable to suppose that it would be proportional to the dynamic energy of the water due to the ship speed. Thus,

$$T_d V_o = k_2 \rho g Q \frac{V_o^2}{2g} .$$

Then

$$H_d = \frac{T_d V_o}{\rho g Q} = k_1 \frac{V_o^2}{2g} .$$

$V_j^2 - V_o^2$ in Equation (36) may be written

$$V_o^2 \left[\left(\frac{V_j}{V_o} \right)^2 - 1 \right] .$$

It may be assumed that the summation of internal energy losses is proportional to the velocity head in the inlet duct,

$$H_L = k_2 \frac{V_i^2}{2g} .$$

Making these three substitutions, Equation (36) becomes

$$\eta_s = \frac{\frac{V_o^2}{2g} \left[\left(\frac{V_j}{V_o} \right)^2 - 1 - \frac{k_1}{\eta_{pi}} \right]}{\frac{V_o^2}{2g} \left[\left(\frac{V_j}{V_o} \right)^2 - 1 \right] + k_2 \frac{V_i^2}{2g} + h}$$

Dividing by $V_o^2/2g$

$$\eta_s = \frac{\left(\frac{V_j}{V_o}\right)^2 - 1 - \frac{k_1}{\eta_{pi}}}{\left(\frac{V_j}{V_o}\right)^2 - 1 + k_L\left(\frac{V_j}{V_o}\right)^2 + \frac{2g\,h}{V_o^2}}$$

Substituting the value of η_{pi} from Equation (6),

$$\eta_s = \frac{\left(\frac{V_j}{V_o}\right)^2 - 1 - \frac{k_1\left(1 + \frac{V_j}{V_o}\right)}{2}}{\left(\frac{V_j}{V_o}\right)^2 - 1 + k_2\left(\frac{V_j}{V_o}\right)^2 + \frac{2g\,h}{V_o}} \quad \ldots\ldots (37)$$

From Equation (32)

$$\eta_o = \eta_{pi}\ \eta_{pu}\ \eta_s,$$

and from Equation (6)

$$\eta_{pi} = \frac{2}{1 + \frac{V_j}{V_o}}.$$

Substituting the value of η_{pi} in Equation (6) in Equation (32),

$$\eta_o = \frac{2\,\eta_{pu}}{1 + \frac{V_j}{V_o}}\,\eta_s. \quad \ldots\ldots (38)$$

Substituting the value of η_s from Equation (37) in (38),

$$\eta_o = 2\,\eta_{pu} \frac{\left(\frac{V_j}{V_o}\right) - 1 - \frac{k_1}{2}}{\left(\frac{V_j}{V_o}\right)^2 - 1 + k_2\left(\frac{V_i}{V_o}\right)^2 + \frac{2gh}{V_o^2}} \qquad \ldots\ldots (39)$$

By differentiating the variable, V_j/V_o (treating all other values as constants), setting the result equal to zero, and solving for V_j/V_o, the optimum value providing the highest overall efficiency can be determined. This operation is simplified if the constant terms in the numerator and denominator of Equation (39) are collected as follows:

$$C_1 = 1 + \frac{k_1}{2}$$

$$C_2 = k_2\left(\frac{V_i}{V_o}\right)^2 + \frac{2gh}{V_o^2} - 1.$$

Then

$$\eta_o = 2\,\eta_{pu} \frac{\frac{V_j}{V_o} - C_1}{\left(\frac{V_j}{V_o}\right)^2 + C_2}\,. \qquad \ldots\ldots (40)$$

Differentiating and solving for V_j/V_o opt,

$$\frac{V_j}{V_o}_{\text{opt}} = C_1 + \sqrt{C_1^2 + C_2}\,. \qquad \ldots\ldots (41)$$

Substituting this value of V_j/V_o opt in Equation (40),

$$\eta_o\left(\text{opt. } \frac{V_j}{V_o}\right) = 2\,\eta_{pu} \frac{C_1 + \sqrt{C_1^2 + C_2} - C_1}{\left(C_1 + \sqrt{C_1^2 + C_2}\right)^2 + C_2}\,.$$

Expanding the denominator, this becomes

$$\eta_o\left(\text{opt}\,\frac{V_j}{V_o}\right) = 2\,\eta_{pu}\,\frac{\sqrt{C_1^{\,2} + C_2}}{2\,C_1\sqrt{C_1^{\,2} + C_2} + 2\,C_1^{\,2} + 2\,C_2} \qquad \ldots\ldots (42)$$

Factoring the denominator,

$$\eta_o\left(\text{opt}\,\frac{V_j}{V_o}\right) = \frac{\eta_{pu}\,\sqrt{C_1^{\,2} + C_2}}{\sqrt{C_1^{\,2} + C_2}\,\left(C_1 + \sqrt{C_1^{\,2} + C_2}\right)}$$

$$= \frac{\eta_{pu}}{C_1 + \sqrt{C_1^{\,2} + C_2}}$$

$$= \frac{\eta_{pu}}{\left(\frac{V_j}{V_o}\right)_{opt}} \,. \qquad \ldots\ldots (43)$$

Recalling that $\eta_o = \eta_{pu}\,\eta_{pi}\,\eta_s$, and substituting the value of η_o (opt V_j/V_o) from Equation (43) and the value of η_{pi} from Equation (6),

$$\frac{\eta_{pu}}{\left(\frac{V_j}{V_o}\right)_{opt}} = \frac{2\,\eta_{pu}\,\eta_s}{1 + \left(\frac{V_j}{V_o}\right)_{opt}} \,.$$

Solving for $\left(\frac{V_j}{V_o}\right)_{opt}$ $\qquad \ldots\ldots (44)$

$$\frac{V_j}{V_o}\left(\text{opt}\right) = \frac{1}{2\,\eta_s - 1}.$$

There are four principal variables involved in the determination of the overall efficiency and optimum V_j/V_o ratio. These are the propulsion system drag coefficient (k_1), the internal energy loss coefficient (k_2), the inlet velocity ratio (V_i/V_o), and the elevation of the nozzle above the mean water level (h). An additional factor, the boundary layer, is not included and should be investigated to see how it might be taken into consideration.

The effect of variation of these four variables over a range is shown in Figures 6 through 9. The main difference between these curves and those resulting from plotting the equations of Johnson and Levy, Figures 1 and 2 is due to incorporation of the concept of the basic ship and charging against the propulsion system the power required to overcome the added drag due to the propulsion system. This has the effect of both lowering and flattening the tops of the curves and moving the optimum V_j/V_o ratio to higher values.

Both Johnson and Levy indicate in their papers that it may be desirable to raise the V_j/V_o ratio above the theoretical optimum in order to save weight. The Lockheed approach permits exact numerical calculation of the new optimum, provided the propulsion system drag can be calculated.

Figure 10 is a plot of optimum pump head at various cruising speeds, using several combinations of loss factors, k_1 and k_2. This figure is based on use of optimum V_j/V_o ratio for each speed and loss factor from equation (41) and the following relationship:

$$H = \frac{V_j^2}{2g} + H_L + h - \frac{V_o^2}{2g},$$

with

$$H_L = k_2 \frac{V_i^2}{2g}.$$

4.0 WATER-JET PUMPS

The limiting factor on pump design will in all probability be cavitation. A commonly used measure of cavitation tendency is suction specific speed, S, which is defined as

$$S = \frac{\eta \sqrt{Q}}{(H_{sv})^{\frac{3}{4}}}$$

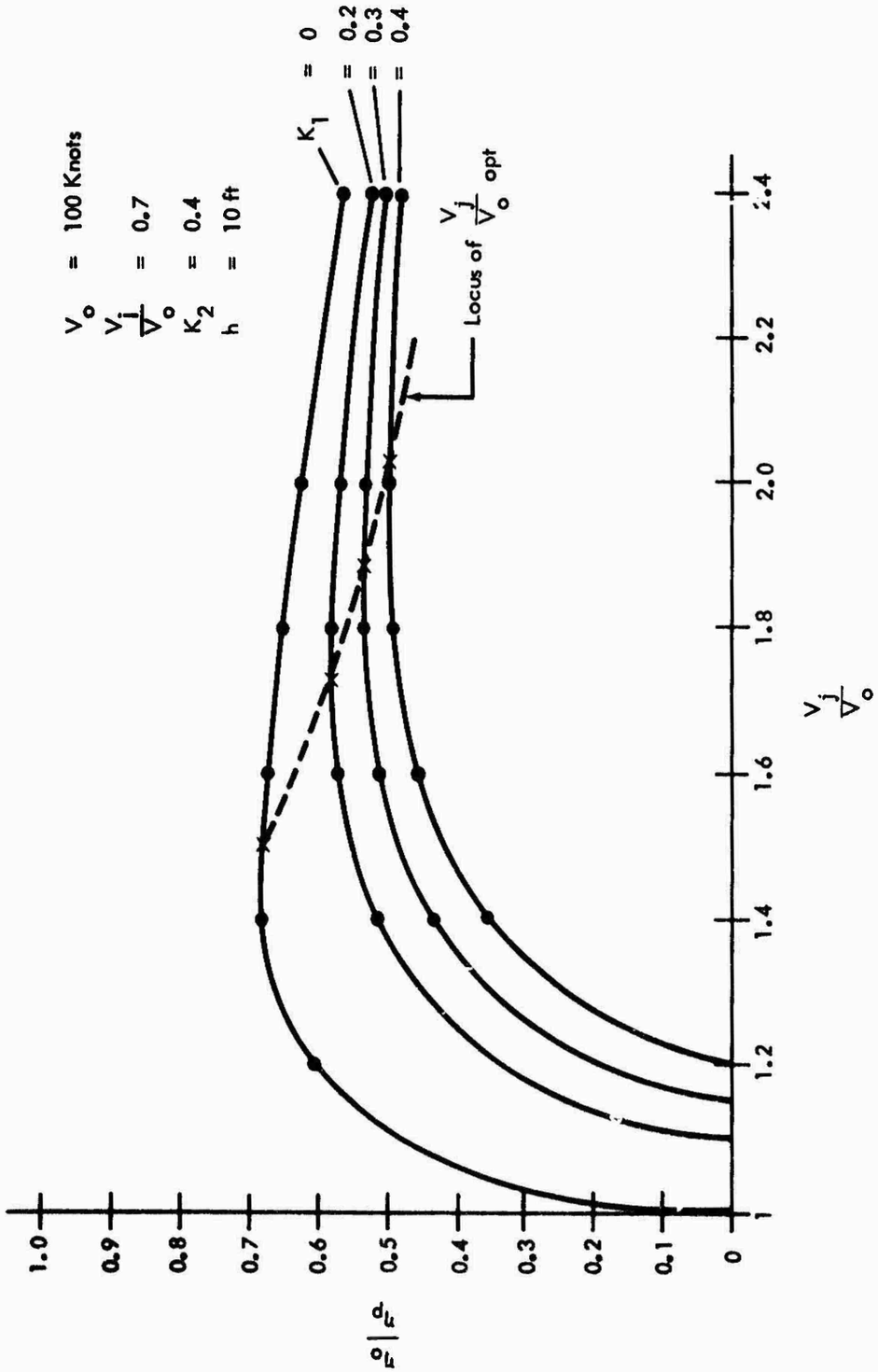

Figure 6
Relation Between Propulsion Efficiency and Propulsion System Drag (Lockheed Method)

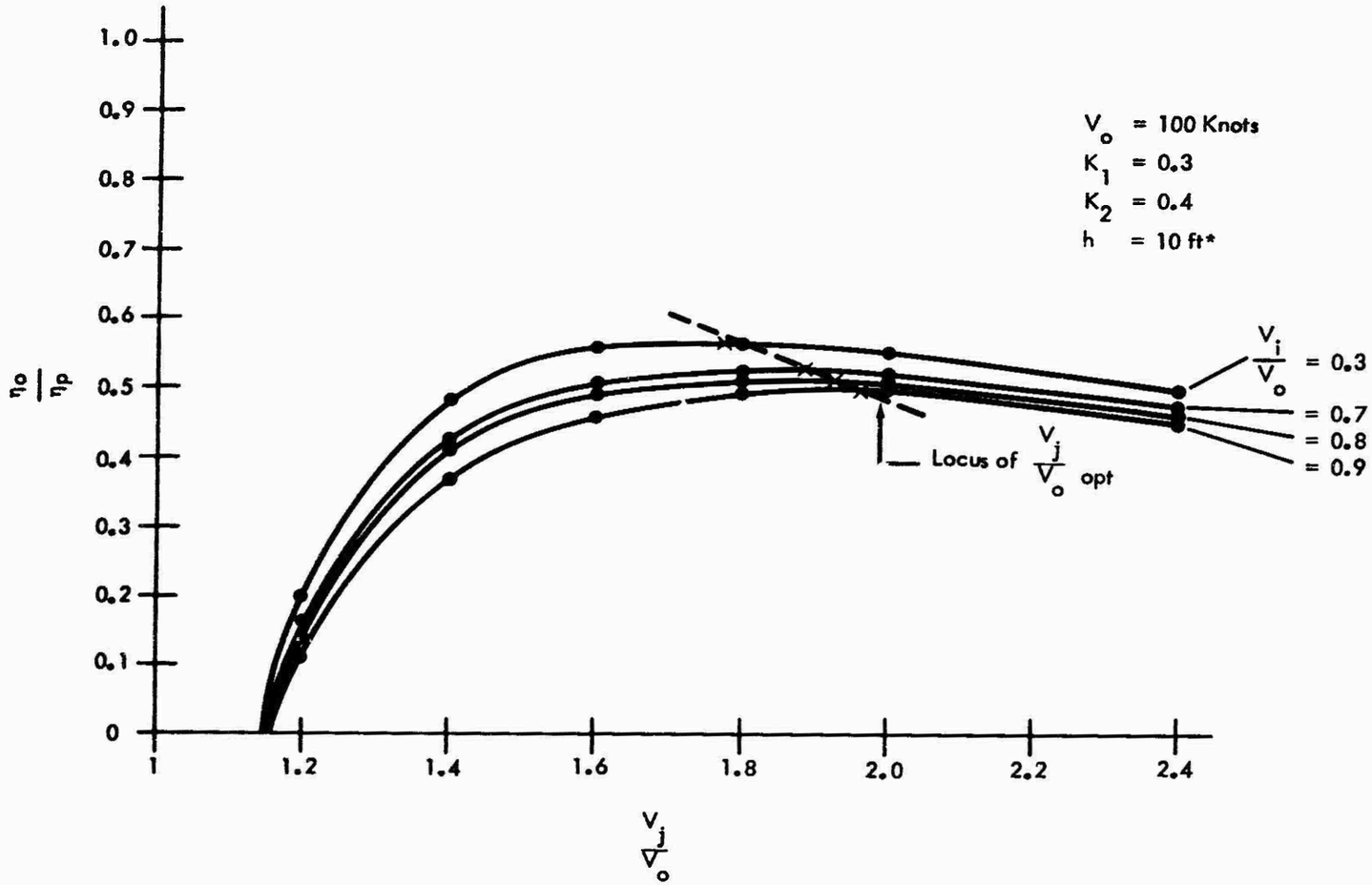

Figure 7
Effect of inlet Velocity Ratio on Overall Efficiency (Lockheed Method)

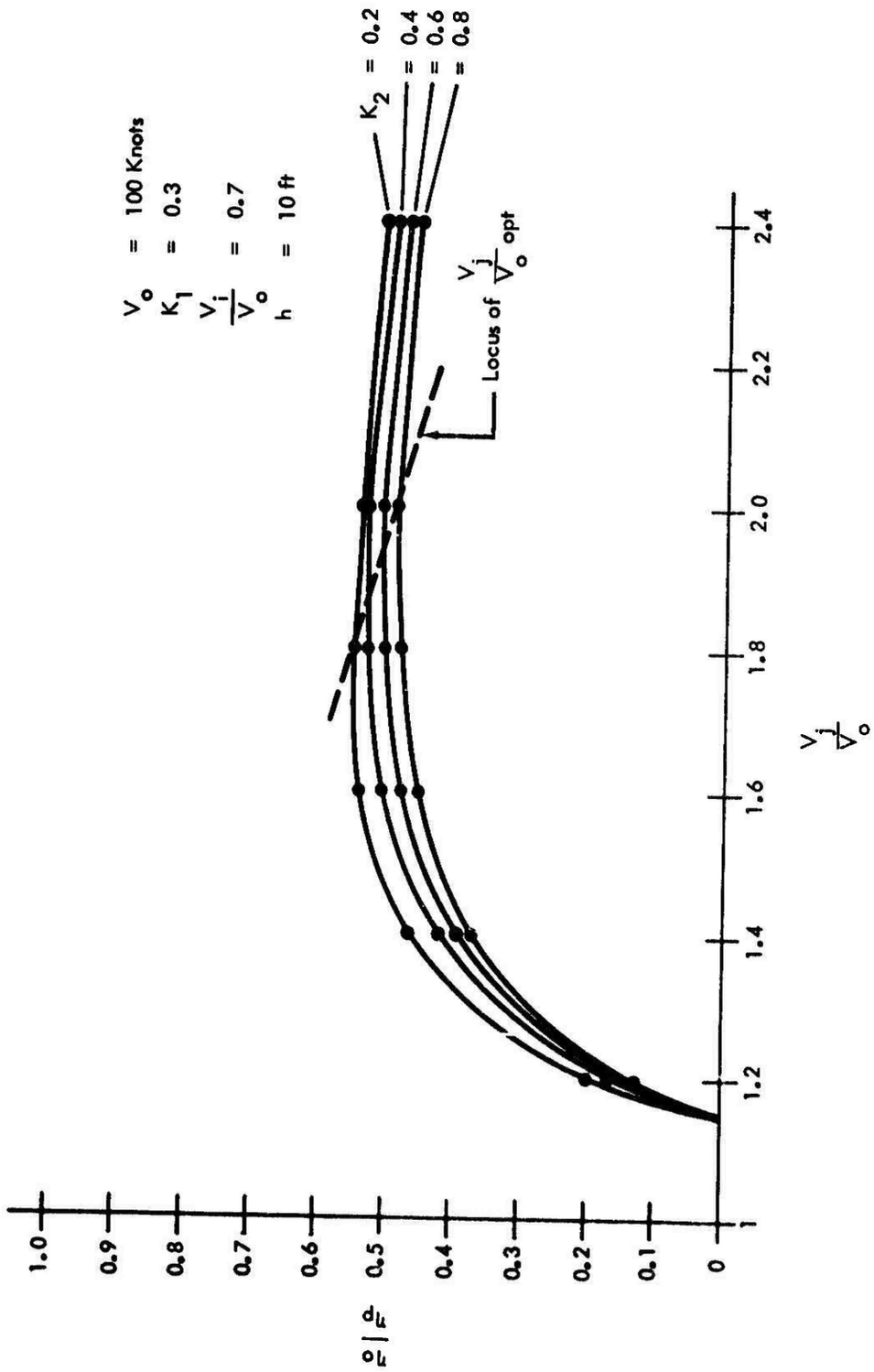

Figure 8
Effect of Internal Loss Factor or Overall Efficiency (Lockheed Method)

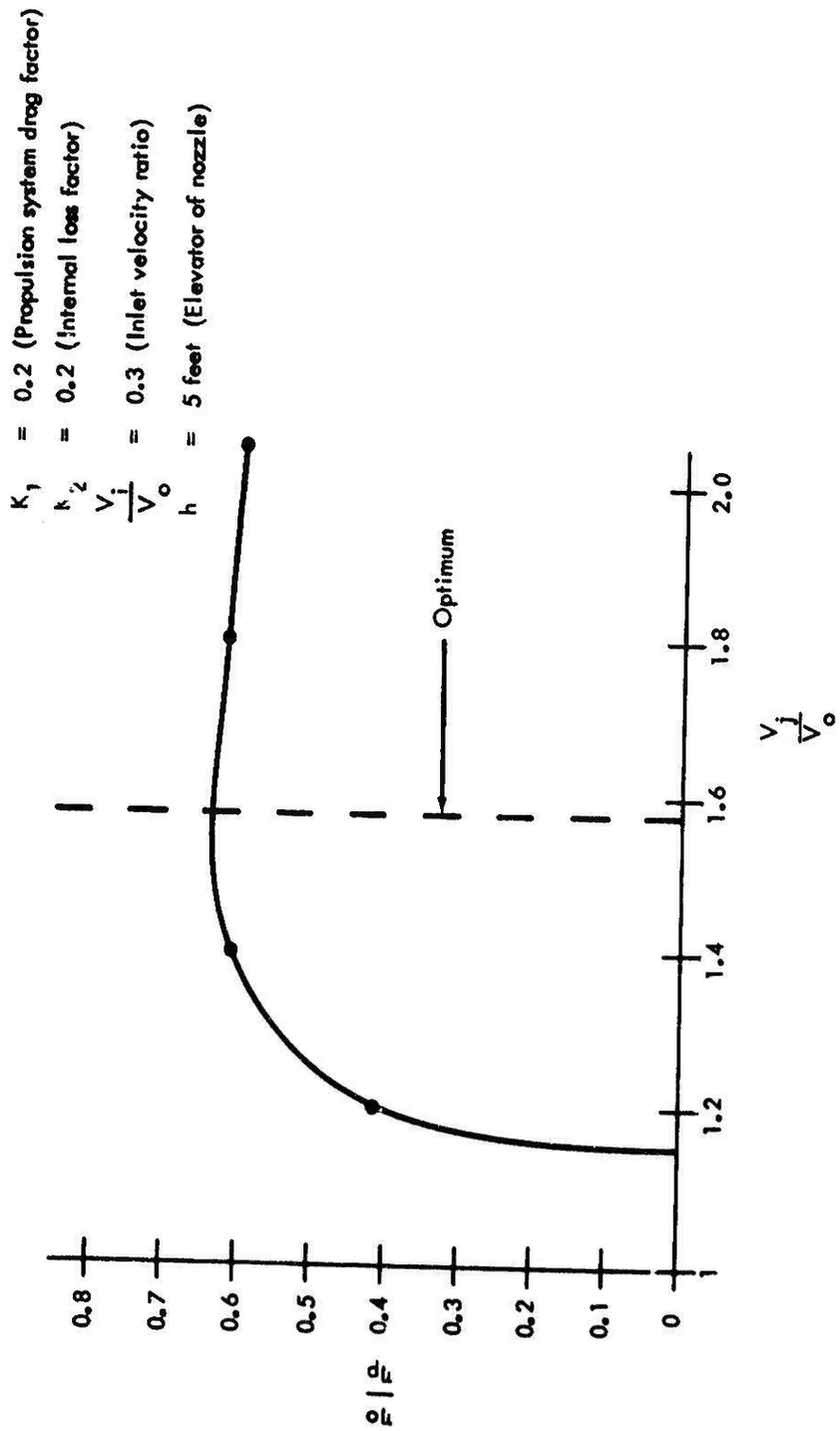

Figure 9
Efficiency of Optimum System at Various V_j/V_o Ratios

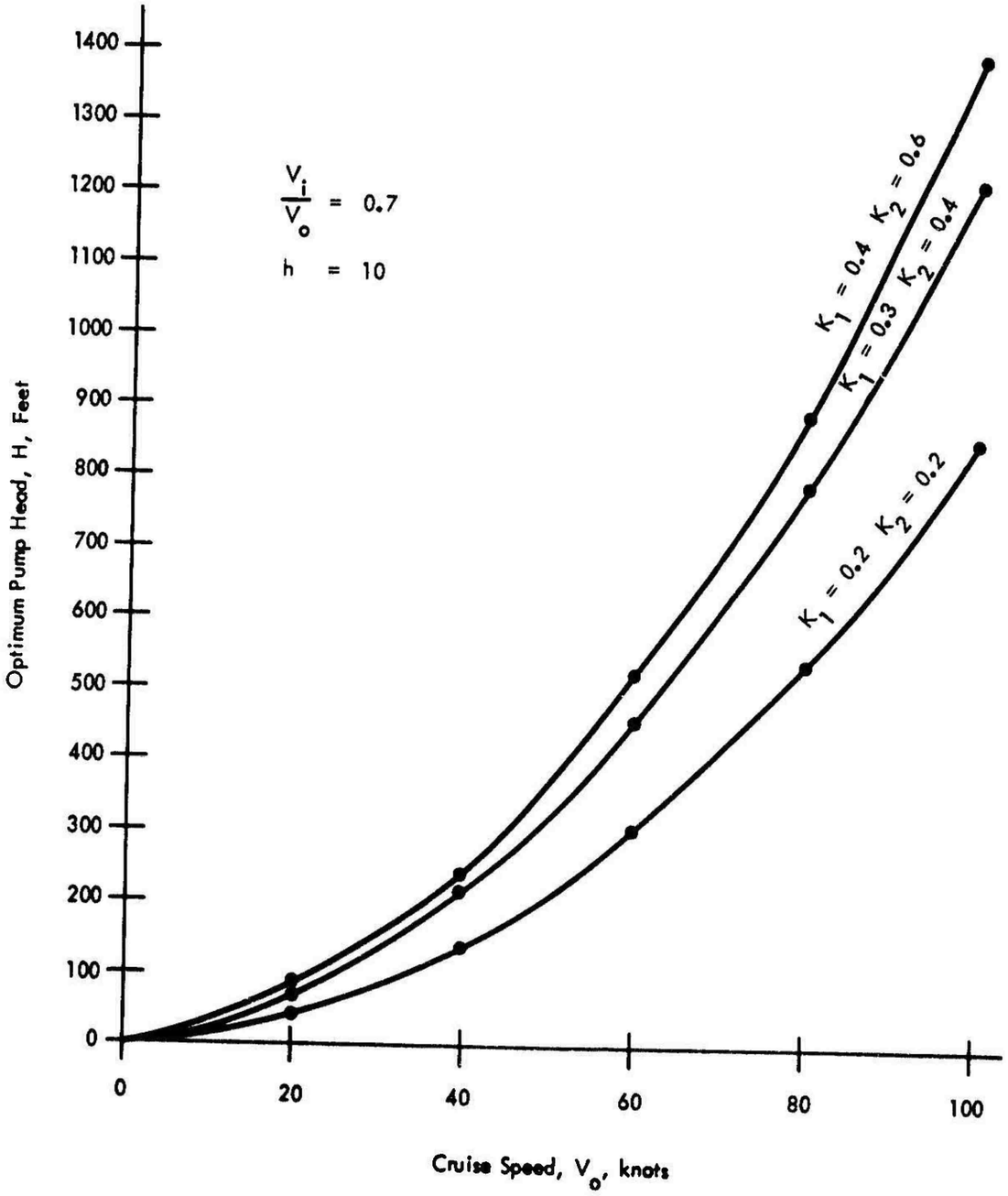

Figure 10
Relation Between Optimum Pump Head and
Ship Speed for a Range of Loss Factors (Lockheed Method)

where

n = Impeller revolution per minute.

Q = Discharge, gallons per minute.

$H_s v$ = Net positive sunction head.

As applied to conventional double-suction centrifugal pumps, a value of S of 12,000 is found to be about the upper limit of cavitation-free operation. Thus,

$$\frac{n\sqrt{Q}}{(H_{sv})^{\frac{3}{4}}} = 12{,}000$$

or

$$n\sqrt{Q} = 12{,}000\,(H_{sv})^{\frac{3}{4}}. \qquad \ldots\ldots (45)$$

Using the conventional definition of pump specific speed,

$$N_s = \frac{N\sqrt{Q}}{H^{\frac{3}{4}}},$$

and substituting the cavitation-limited value of $n\sqrt{Q}$ from Equation (45), the limiting value of N_s is

$$N_s = 12{,}000\left(\frac{H_{sv}}{H}\right)^{\frac{3}{4}}. \qquad \ldots\ldots (46)$$

4.1 As applied to hydrofoils, and probably also to surface effect ships, it may be assumed that the maximum thrust of a water-jet system will be required at the hump transition, when the ship speed is third or half the value of V_0 at cruising speed. In order to provide this thrust, head and flow rate must be maintained at about the same level at hump and cruising speeds. Since the net positive suction head will be lower at hump speed than at cruising speed, pump cavitation tendency will be greatest at hump speed.

4.2 The net positive suction head (H_{SV}) is the sum of the atmospheric pressure head plus the ram pressure head (reduced by internal friction and diffusion pressure losses), less the water vapor pressure head and the elevation of the pump above the water level outside the hull. If hump speed is half V_O, $H_S{}^V$ (using the approach of reference 1) will be approximately

$$33 + (1 - k)\left(\frac{1}{4}\right)\left(\frac{V_o^{\,2}}{2g}\right) - h.$$

Substituting this value in Equation (46),

$$N_s = 12000 \left[\frac{33 + (1-k)\frac{1}{4}\left(\frac{V_o^{\,2}}{2g}\right) - h}{H}\right]^{\frac{3}{4}}$$

Remembering that $K = k + 2gh/V_o^{\,2}$,

$$N_s \equiv 12000 \left[\frac{33 + \frac{V_o^{\,2}}{8g}(1 - K)}{H}\right]^{\frac{3}{4}} \qquad \ldots\ldots (47)$$

It will be noted that the internal energy loss calculated by this equation is a fourth that at cruising speed. Since the head and flow are to be approximately the same as at cruise, it seems reasonable to suppose that the internal energy losses would remain about the same. Accordingly, the present reviewer suggests that Equation (47) be modified to

$$N_s \equiv 12000 \left[\frac{33 + \left(\frac{1}{4} - K\right)\frac{V_o^{\,2}}{2g}}{H}\right]^{\frac{3}{4}} . \qquad \ldots\ldots (48)$$

This would, of course, lead to a somewhat lower cavitation-limited value of N_S than Equation (47). In Figure 11, Equation (48) is plotted to show the relation between N_S and V_O for various K factors. For these curves, the optimum head is taken from Figure 3 for each V_o and K.

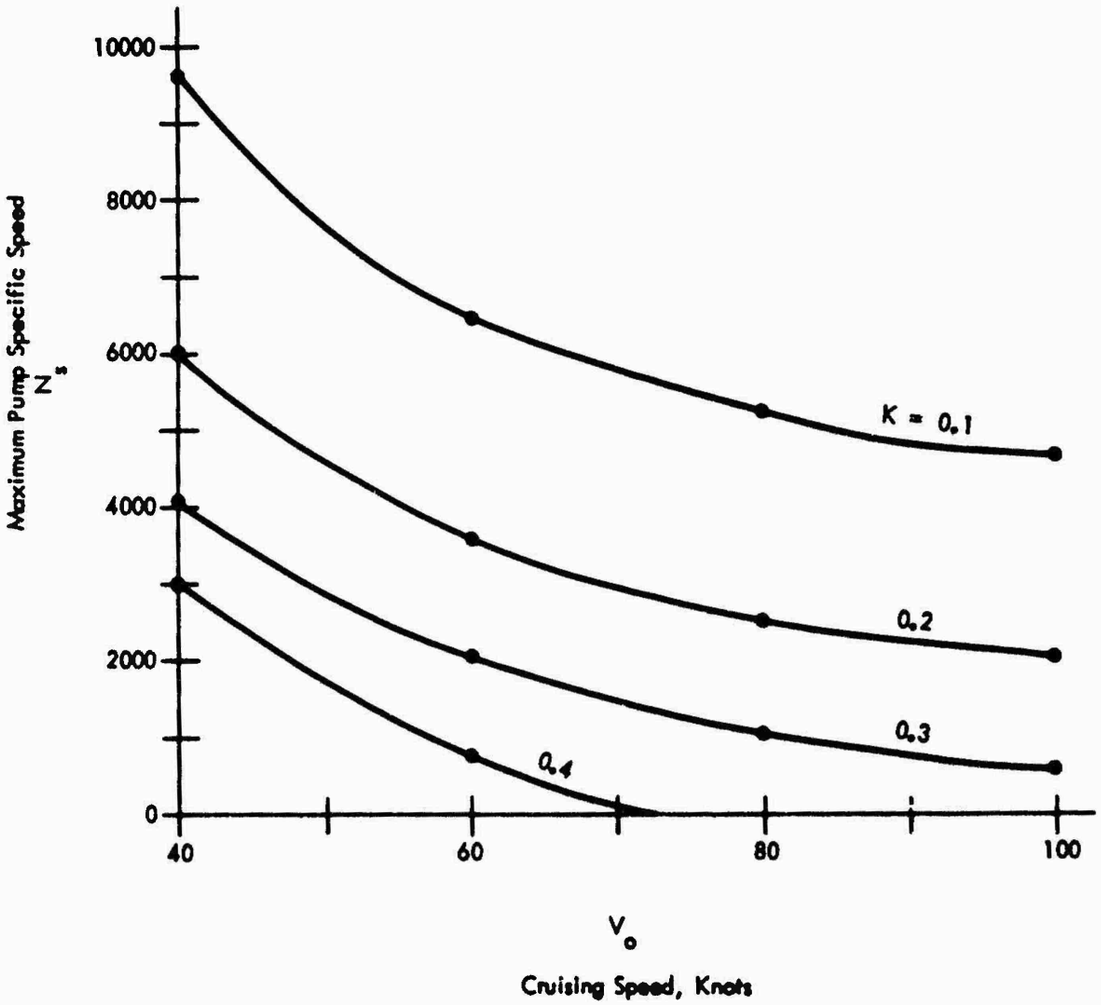

Figure 11
Maximum Pump Specific Speed as Limited by Cavitation

According to Stepanoff data,[4] based on analysis of a large number of centrifugal pumps, large-flow centrifugal pumps have achieved maximum efficiency when designed for specific speeds in the range 2500 - 3000. The curves of Figure 11 show that if the criteria assumed are correct (same head and flow at hump speed as at cruising speed and maximum suction specific speed limited to 12,000) for a ship designed to cruise at 100 knots, the loss factor must be below 0.2 if highest efficiency pumps are to be used.

4.3 Since the Lockheed approach leads to somewhat higher (but probably more nearly correct) values of optimum pump head than the methods of Johnson and Levy, cavitation limits will be more severe, particularly during hump transition. It may be of interest to look at a conventional pump system, optimized for the cruise condition, and then consider the operating conditions for the hump transition. At cruise speed,

$$H_{sv} = 33 + \frac{V_o^2}{2g} - k_2 \frac{V_i^2}{2g} - h. \qquad \ldots\ldots (49)$$

If h is taken as 10 feet and V_i/V_o as 0.7, Equation (49) becomes

$$H_{sv} = 23 + (1 - 0.7\,k_2) \frac{V_o^2}{2g}. \qquad \ldots\ldots (50)$$

Using Equation (46),

$$\text{Maximum } N_s = \frac{12000 \left[23 + \left(1 - 0.7\,k_2\right) \frac{V_o^2}{2g} \right]^{\frac{3}{4}}}{H^{\frac{3}{4}}}. \qquad \ldots\ldots (51)$$

Figure 12 is a plot of maximum Ns for low, medium, and high loss factors for various cruising speeds. For these curves, the optimum pump head for each loss factor and speed is taken from Figure 10. This shows pump specific speeds at 100 knots as being limited to 3000 - 7000, depending on system losses. These values are, however, based on the conventional practice with double-suction centrifugal pumps of limiting suction specific speed to 12,000. In an axial-flow pump, particularly if special designs involving inducers or the equivalent are employed, this value would be greatly increased.

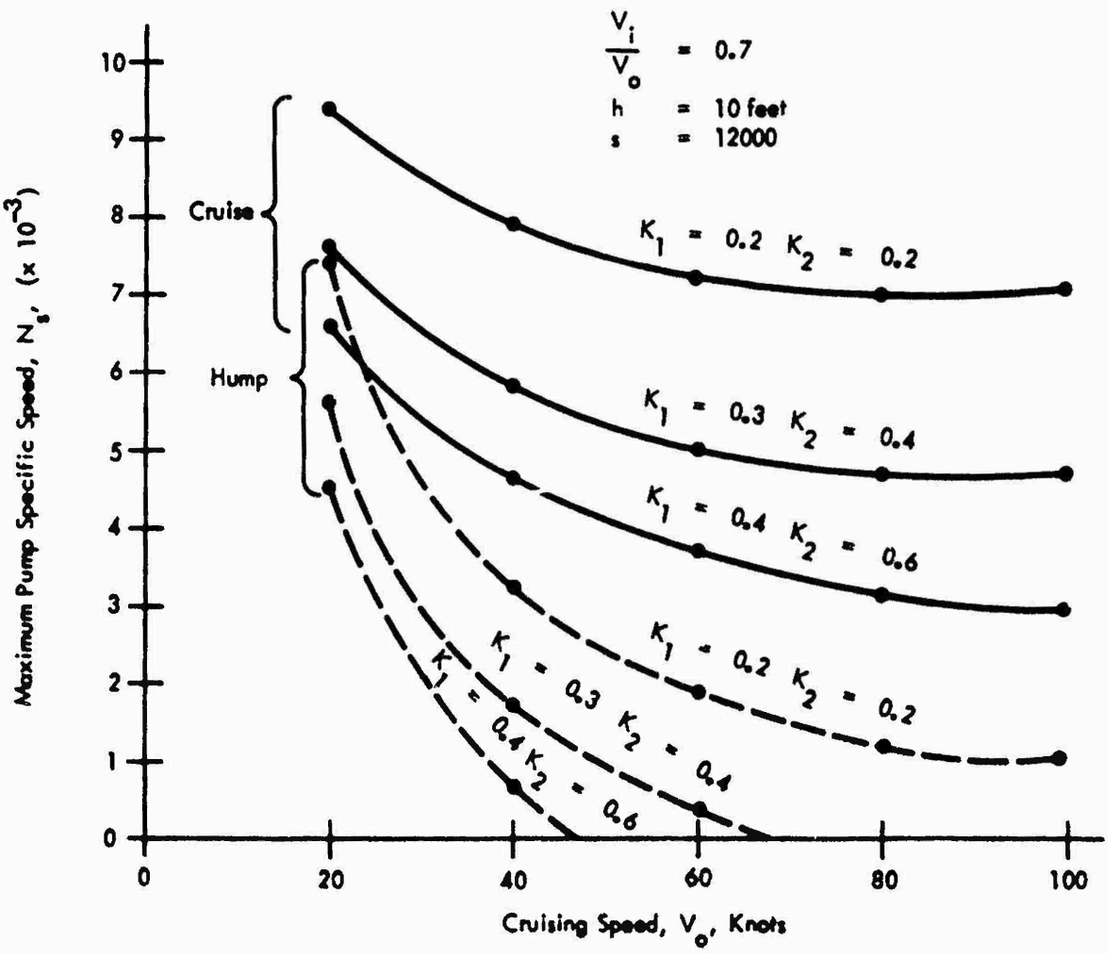

Figure 12
Cavitation Limit on Maximum Specific Speed at Cruising Speed

The real cavitation problem, however, lies in traversing the hump region. This is because the thrust requirements here are nearly as high as, or possibly even higher than, at cruise speed, but the net positive suction head available, being nearly proportional to the square of the speed, is very much lower. Figure 12 also shows curves of maximum specific speed at the hump transition assuming that hump speed is 40 percent of V_0 and using the same cavitation criterion of 12,000 suction specific speed. For these calculations, the ship thrust-speed characteristics of Figure 13 are used, which show hump thrust and flow conditions essentially the same as at cruise speed.

At hump speed,

$$H_{sv} = 33 + \frac{(f\,V_o)^2}{2g} - H_L - h,$$

where f is the fraction of cruising speed (V_o) represented by hump speed, and

$$H_L = K_L \frac{V_i^2}{2g}.$$

The resulting cavitation limits for pumps optimized for different loss ratios and cruising speeds are shown as dashed lines in Figure 12. It will be observed that, if the configuration of the inlet, ducts, and nozzles remains the same at hump speed as at cruise, the flow rate required to provide the necessary thrust cannot be secured except with very efficient inlets and ducts, or the net positive suction head otherwise falls to zero due to inlet losses. Therefore, it appears likely that either the inlet will have to be enlarged, or the nozzle constricted, or both at the hump condition, in order to increase the net positive suction head on the pump.

4.4 The widely reproduced chart of Stepanoff (Figure 5.1 in reference 4, page 76) indicates, on the basis of analysis of a large number of pump design, optimization of efficiency at a specific speed around 3000 for very large flow pumps. The optimum conventional impeller form for such a pump would be a so-called "Francis" type, where the flow is more nearly radial than axial. From Figure 12 it appears that as a "first cut" pump for a large surface effect ship, a specific speed of about 3000 would be appropriate, with suitable inlet and nozzle variation to provide a reasonable positive suction head at hump speed. It would undoubtedly be necessary to operate the pump in a cavitating condition during transit of the hump if the hump is in fact as pronounced as shown by Chaplin and Ford.[5]

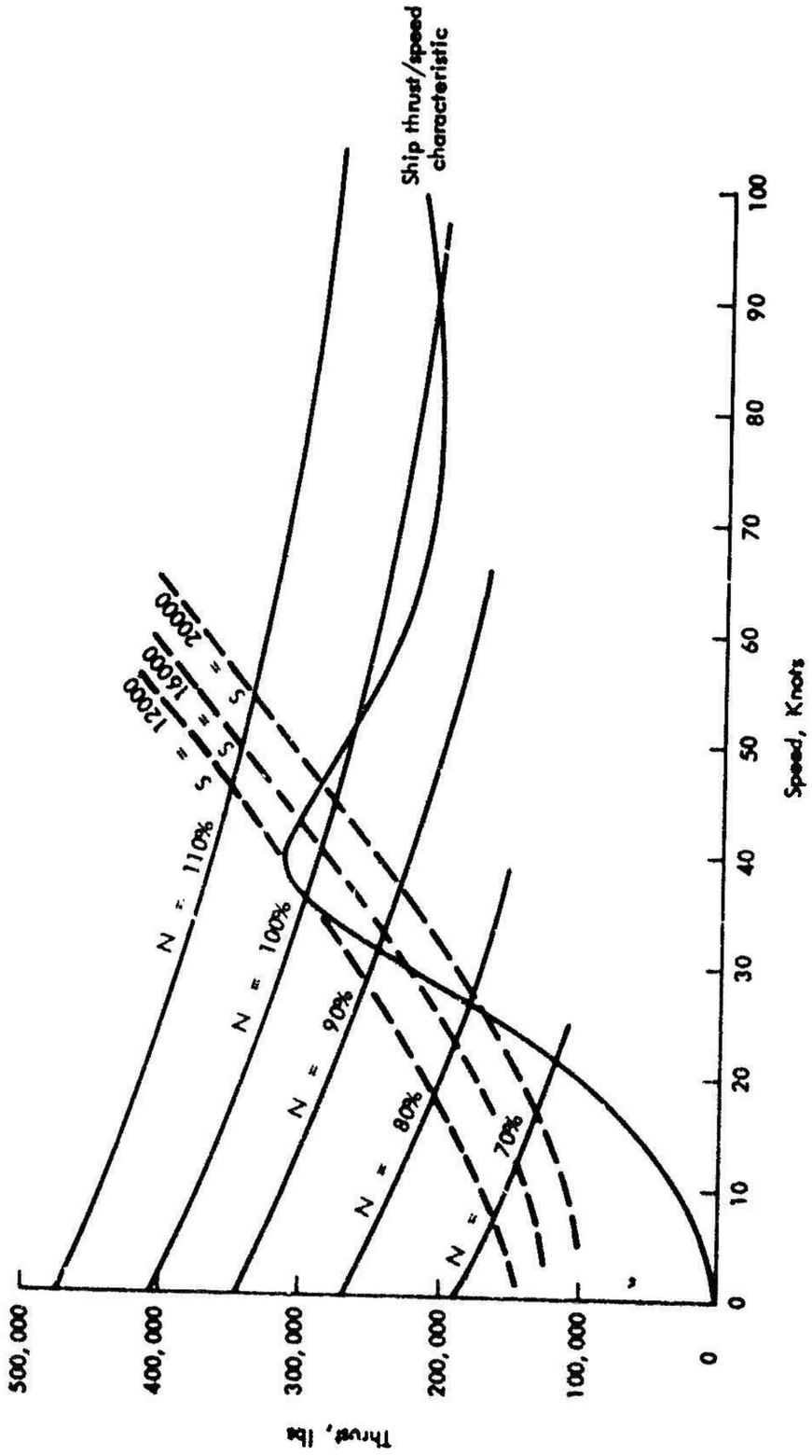

Figure 13
Thrust Provided at Various Pump Speeds Ship Drag-Speed Relation

5.0 PERFORMANCE MAPS

Levy[2] shows two types of performance maps which are useful in matching pump characteristics to vessel characteristics. In one, the coordinates are head and capacity ratios; in the other, the coordinates are thrust and speed.

For the system head-capacity ratio chart, it is noted that Q is related to V_j in the following manner:

$$Q = V_j A_j,$$

or

$$V_j = \frac{Q}{A_j}, \qquad \text{..... (52)}$$

Where A_j is the cross-sectional area of the jet at the maximum velocity section.

From Equation (28), the head required to be supplied by the pump is equal to the jet velocity head plus the internal energy losses, less the velocity head resulting from the forward motion of the ship.

$$H = (1 + k_L) \frac{V_j^2}{2g} - \frac{V_o^2}{2g}.$$

Substituting the value of V_j from Equation (52),

$$H = \left(\frac{1 + k_L}{A_j}\right) \frac{Q^2}{2g} - \frac{V_o^2}{2g}.$$

Rearranging,

$$Q = \left(\frac{2gH + V_o^2}{\frac{1 + k_L}{A_j}}\right)^{\frac{1}{2}}.$$

The ratio of flow at two different pump heads is then

$$\frac{Q_2}{Q_1} = \left(\frac{2gH_2 + V_o^2}{2gH_1 + V_o^2}\right)^{\frac{1}{2}} . \qquad \text{..... (53)}$$

If H_1 and Q_1 are the design point head and flow rate, the Q_2/Q_1 ratio for any other pump head can be calculated from Equation (53) for any speed. A series of curves similar to those shown in Figure 14 result.

Typical pump characteristics can be superimposed on the same coordinates, as shown in Figure 14. Then, if the thrust required for any selected vessel speed is known, and the pump speed required to produce this thrust is also known, the operating point in Figure 14 for any vessel speed can be found.

The pump speed required to produce a certain thrust can be computed as follows, if the pump characteristics, as shown in Figure 14, are known:

From Equation (28),

$$H = (1 + k_L)\frac{V_j^2}{2g} - \frac{V_o^2}{2g} .$$

Rearranging,

$$V_j = \left(\frac{2g\,H + V_o^2}{1 + k_L}\right)^{\frac{1}{2}} . \qquad \text{..... (54)}$$

From Equation (1),

$$T = \rho Q\,(V_j - V_o).$$

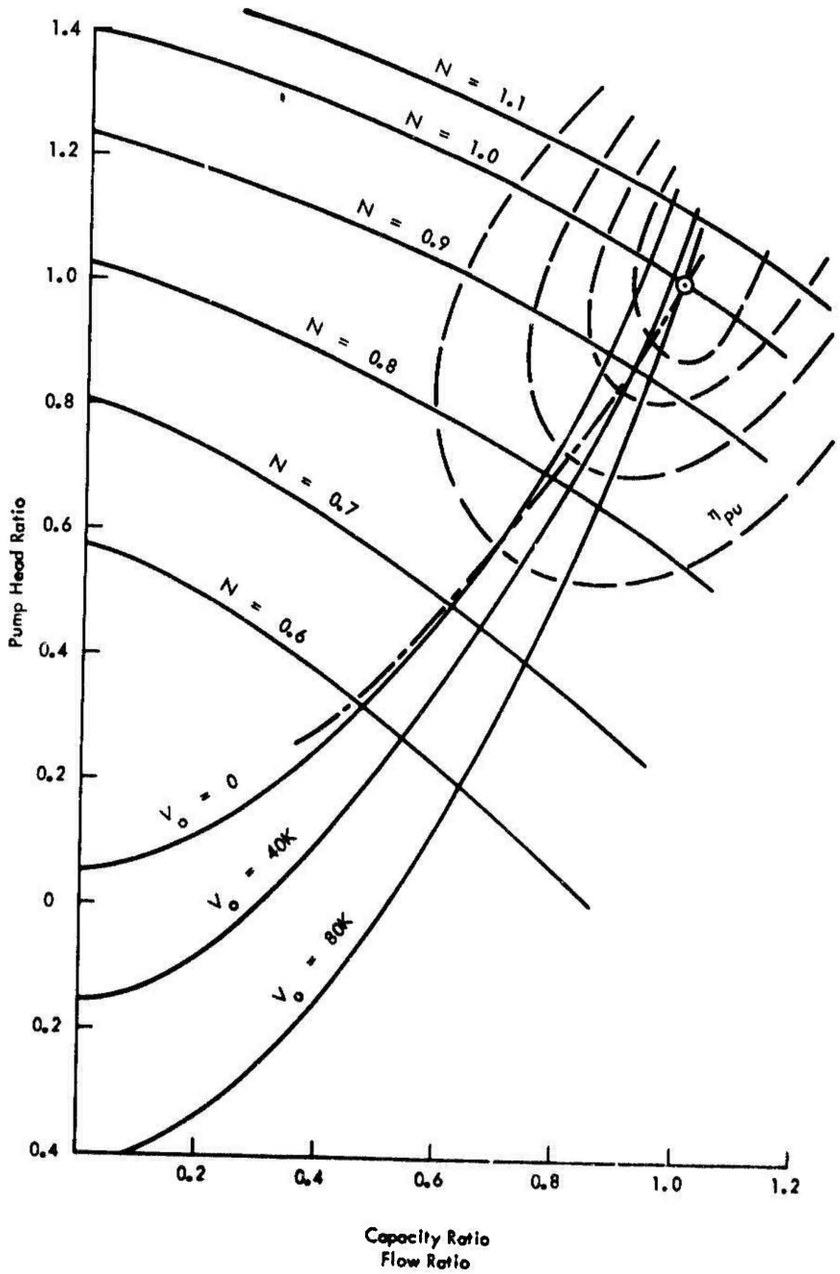

Figure 14
Relation Between Pump and System Flow Characteristics

Substituting the value of V_j from Equation (54) into (1),

$$T = \rho Q \left[\left(\frac{2gH + V_o^2}{1 + k_L} \right)^{\frac{1}{2}} - V_o \right]. \qquad \ldots\ldots (55)$$

Since Figure 14 provides Q and H for any selected pump speed, all of the values necessary to compute thrust from Equation (55) are known.

Figure 13 shows the sort of curves which would result for a typical installation.

A curve of vessel drag in relation to speed can be superimposed on the same coordinates. A typical curve for a surface effect ship is shown in Figure 13. By finding the pump speed to produce a certain ship speed from Figure 13, the pump operating point can be directly located in Figure 14.

The cavitation point, assuming this to be a function of suction specific speed, can also be shown in Figure 13. Typical values for three different values of suction specific speed are shown.

6.0 DISCUSSION

It would appear that the Lockheed approach, by taking into account the effect on the optimum jet velocity ratio of differences in the weight and drag characteristics of alternative propulsion systems, affords a more useful approach to system design than those of Johnson and Levy. Also, relating internal energy losses to the inlet velocity, rather than to the vessel velocity or the jet velocity, seems more realistic. Even so, the various k factors must be used with considerable caution. Relating inlet and duct losses to $V_1^2/2g$ is an obvious oversimplification. (Actually in the Lockheed study, a more direct approach of calculating and adding the individual sources of loss through the system is taken. The oversimplification is due to the effort of the present writer to indicate principles rather than details.)

In particular, use of V_i as the measure of internal energy losses tends to mask the effect of diffuser losses in relation to duct wall friction losses, which may in some cases trend in opposite directions. For example, with a given flow rate, increased diffusion will reduce frictional losses, due to reduced duct velocity; but this saving will be partially offset by the loss in the diffusion process. Diffusion losses in this sense are not correctly accounted for in the equations presented here, and differences in inlet velocity ratio probably have less effect than shown in Figure 7.

7.0 FUTURE PLANS

In support of the Surface Effects Ship Program, further studies will be carried forward along lines of:

7.1 Estimating the specific characteristics of surface effect ship water-jet systems based on use of conventional pumps of around 3000 specific speed suitable for use with the Pratt & Whitney FT-4A engine in a 4000-ton water-jet ship.

7.2 Similar estimates with respect to the 500-ton prototype ship.

7.3 Preliminary layout and weight estimates of the machinery plant for the 4000- and 500-ton ships.

7.4 Refinement of propulsion system drag and internal loss estimates in order to identify optimum pump characteristics with greater certainty. This will include quantifying the various loss coefficients and also developing better methods of taking into consideration the effects on required pump characteristics of diffuser losses and boundary layer ingestion.

7.5 Development of optimum nonconventional pump designs.

Appendix A

Nomenclature

C_1 - $1 + \frac{k_1}{2}$

C_2 - $k_2\left(\frac{V_i}{V_o}\right)^2 + \frac{2gh}{V_o^2} - 1$

g - Acceleration of gravity, fps^2

h - Difference in elevation between nozzle and mean water level, ft

H - Energy added to water by pump, ft-lb/lb

H^* - Ratio of energy added to water by pump to kinetic energy of water due to ship forward motion ($V_o^2/2g$), nondimensional

H_d - Energy expended to overcome drag of propulsion system, ft-lb/lb

H_L - Summation of internal energy losses, ft-lb/lb

H_{sv} - Net positive suction head, ft-lb/lb

k - Ratio of internal energy losses to entering kinetic energy, nondimensional

K - $k + \frac{2gh}{V_o^2}$, nondimensional

k_L - Ratio of internal energy losses to jet kinetic energy, nondimensional

k_1 - Ratio of power required to overcome propulsion system drag, to product of water flow rate and kinetic energy, nondimensional

k_2 - Ratio of internal energy losses to velocity head in inlet duct, nondimensional

n - Pump revolutions per minute

N_s - Pump specific speed, $\frac{n\sqrt{GPM}}{H^{\frac{3}{4}}}$

η - Propulsive efficiency x pump efficiency

η_p, η_{pu} - Pump efficiency

η_{pi} - Ideal propulsive efficiency

η_o - Overall efficiency, $\eta_{pu}\,\eta_{pi}\,\eta_s$

η_s - System efficiency

P - Shaft power applied to pump, ft-lb/sec

Q - Water flow, ft^3/sec
In pump specific speed formulae, water flow, gpm

r - $\frac{V_j - V_o}{V_o}$, nondimensional

ρ - Water density, $slugs/ft^3$, $lb\text{-}sec^2/ft^4$

S - Pump suction specific speed, $\frac{h\sqrt{GPM}}{(H_{sv})^{\frac{3}{4}}}$

T - Total drag of ship = thrust of propulsive system, lb

T_b - Drag of basic ship without propulsion system, lb

T_d - Drag attributable to propulsion system, lb

V_i - Inlet water velocity relative to ship, fps

V_j - Jet velocity relative to ship, fps

V_o - Approaching water velocity, relative to ship = ship forward velocity, at cruising speed, fps

ΔV - $V_j - V_o$, fps

Appendix B

Technical References

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2 - Levy, Joseph, "The Design of Water-Jet Propulsion Systems for Hydrofoil Craft," SNAME, 14 May 1964

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5 - Chaplin, Harvey, and Allen Ford, Preliminary Draft of Design Principles of Ground Effect Machines, Section H, Supplement 2, Performance Graphs, May 1965

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13. ABSTRACT

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Mathematical basis						
Pumps						
Velocity						
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Cruise speed						
Efficiency						

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COMPARISON OF THREE THEORIES OF WATER-JET PROPULSION, by J. H. Garrett. February 1967.
41 pp. Figs. UNCLASSIFIED

The three different approaches to derivation of formulae expressing the relations among speed, thrust, power, and efficiency of water-jet propulsion systems, as developed by Lockheed California Company; Virgil Johnson of Hydronautics, Incorporated; and Joseph Levy of Aerojet-General Corporation, are summarized and compared. Certain modifications and simplifications are incorporated, and terminology is modified as necessary to facilitate comparison. The Lockheed system, which provides a method for including the weight and drag of the propulsion system in the optimization (over)

1. Hydrofoil craft
2. Ground-effect machines
3. Water-jet propulsion
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procedure, appears to be the more useful. The problems of compromising the performance of the propulsion system at cruising speed in order to provide reasonable hump performance are briefly discussed.