WATERJET PUMP PERFORMANCE DETERMINATIONS

Arthur F. Garcia, Jr.
Mechanical Engineer
Propulsion Systems Analysis Branch
Naval Ship Engineering Center

This document has been approved for public release and sale; its distribution is unlimited.

The views expressed herein are the personal opinions of the author and are not necessarily the official views of the Department of Defense or of a Military Department.
Abstract

The following calculation procedure originated from the necessity of determining waterjet pump performance requirements for waterjet propelled sea vessels. The waterjet pump sizing procedure is basically in two parts. The purpose of the first part is to determine the requirements of flow rate and head required of any pump to produce the necessary thrust to achieve design speed. Secondly, where the pump requirements are known, studies to determine pump rpm, cavitation limitation, impeller diameter, case volume and wet weight can be made to estimate some detailed aspects of various pump types which might be applied. A computer program has been developed from this procedure which streamlines studying variations in design. The procedures and the computer program can also be used in evaluating waterjet systems presented in studies and proposals.
### Notation

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Units</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>ft²</td>
<td>Area</td>
</tr>
<tr>
<td>De</td>
<td>ft</td>
<td>Elevation deviation</td>
</tr>
<tr>
<td>Dimp</td>
<td>ft</td>
<td>Impeller diameter</td>
</tr>
<tr>
<td>E</td>
<td>ft</td>
<td>Efficiency</td>
</tr>
<tr>
<td>fl</td>
<td>ft/sec²</td>
<td>Inlet velocity ratio</td>
</tr>
<tr>
<td>g</td>
<td>ft/gal/min</td>
<td>Gravitational constant</td>
</tr>
<tr>
<td>gpm</td>
<td>ft</td>
<td>Flow rate per pump stage</td>
</tr>
<tr>
<td>HD</td>
<td>ft</td>
<td>Head</td>
</tr>
<tr>
<td>HL</td>
<td>ft</td>
<td>Head loss</td>
</tr>
<tr>
<td>Hld</td>
<td>ft</td>
<td>Differential head loss</td>
</tr>
<tr>
<td>NPSH</td>
<td>ft</td>
<td>Net positive suction head</td>
</tr>
<tr>
<td>Na</td>
<td>ft</td>
<td>Pump specific speed</td>
</tr>
<tr>
<td>Patm</td>
<td>ft</td>
<td>Atmospheric pressure</td>
</tr>
<tr>
<td>Fvap</td>
<td>ft</td>
<td>Vaporization pressure of water</td>
</tr>
<tr>
<td>Q</td>
<td>ft³/sec</td>
<td>Flow rate</td>
</tr>
<tr>
<td>R</td>
<td>ft³/sec</td>
<td>Jet Velocity Ratio</td>
</tr>
<tr>
<td>Rey</td>
<td>ft</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>RPM</td>
<td>ft</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>Subm</td>
<td>ft</td>
<td>Submergence</td>
</tr>
<tr>
<td>SUCT</td>
<td>ft</td>
<td>Suction specific speed</td>
</tr>
<tr>
<td>T</td>
<td>lbs</td>
<td>Thrust</td>
</tr>
<tr>
<td>Us</td>
<td>ft/sec</td>
<td>Impeller specific speed</td>
</tr>
<tr>
<td>V</td>
<td>ft/sec, kts.</td>
<td>Velocity</td>
</tr>
<tr>
<td>WHP</td>
<td>HP</td>
<td>Ideal horsepower (water horsepower)</td>
</tr>
<tr>
<td>$\beta$</td>
<td>ft²/ft³</td>
<td>Impeller blade discharge angle</td>
</tr>
<tr>
<td>$\phi$</td>
<td>$\phi$</td>
<td>Density</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kg/m³</td>
<td>Flow coefficient</td>
</tr>
<tr>
<td>$\rho$</td>
<td>$\rho$</td>
<td>Specific density</td>
</tr>
<tr>
<td>$\psi$</td>
<td>$\psi$</td>
<td>Cavitation factor</td>
</tr>
<tr>
<td>Subscripts</td>
<td>Units</td>
<td>Definition</td>
</tr>
<tr>
<td>------------</td>
<td>-------</td>
<td>------------</td>
</tr>
<tr>
<td>av</td>
<td></td>
<td>Available at the eye of the pump</td>
</tr>
<tr>
<td>ax</td>
<td></td>
<td>At the pump exit</td>
</tr>
<tr>
<td>fs</td>
<td></td>
<td>Free stream</td>
</tr>
<tr>
<td>i</td>
<td></td>
<td>Inlet</td>
</tr>
<tr>
<td>int</td>
<td></td>
<td>Intake</td>
</tr>
<tr>
<td>j</td>
<td></td>
<td>Jet</td>
</tr>
<tr>
<td>k</td>
<td></td>
<td>Craft</td>
</tr>
<tr>
<td>nz</td>
<td></td>
<td>Nozzle</td>
</tr>
<tr>
<td>pmp</td>
<td></td>
<td>Pump</td>
</tr>
<tr>
<td>s</td>
<td></td>
<td>Stage</td>
</tr>
<tr>
<td>t</td>
<td></td>
<td>Total</td>
</tr>
<tr>
<td>tr</td>
<td></td>
<td>Transmission</td>
</tr>
</tbody>
</table>
TABLE OF CONTENTS

Abstract
Notation
I. Development of Relationships
   Basic Approach
   Inlet Velocity Ratio
   Intake and Nozzle Losses
   Total Thrust Required
   Pump RPM and Cavitation Determinations
   Impeller Pump Diameter

II. Computer Program

III. Conclusions

IV. Acknowledgements

V. References

VI. Appendix A
   Sample Run
   Terms
   Calculation Procedure
   Program Print-Out

VII. Appendix B
   Inlet Area Sizing Guide
   Sensitivity Trends

VIII. Figures
The Basic Approach

The flow rate and the head differential required of the waterjet pump are based on three inter-related equations. The flow rate is a function of the velocity of flow across the inlet area (equ. 1). The thrust required is dependent on the flow rate, the jet velocity and craft velocity (equ. 2). The head differential the pump must provide is a function of the head of the jet stream, the head of the free stream, and system losses (equ. 6).

The head differential the pump must produce is simplified by equation 3 as the absolute head at the pump exit (equ. 4) minus the absolute head available at the eye of the pump (equ. 5). Head of the jet and head of the free stream are defined by equations 7 and 8.

Inlet Velocity Ratio

The inlet velocity ratio is the ratio of the inlet velocity $V_i$ to the free stream velocity $V_k$ at the point the inlet area is measured (equ. 9). Inlet velocity is calculated from the inlet velocity ratio at design speeds. Flow rate is calculated from the inlet velocity at the location along the inlet where the inlet area is measured (equ. 10). Inlet velocity ratio is usually smaller than one at cruising speeds but can be greater than one at take-off or when accelerating.

Intake and Nozzle Losses

Intake losses include all friction and elevation effects by the inlet and ducting system which reduces the free stream head to the head available at the pump. Intake efficiency is essentially the ratio of the absolute head available at the pump to the free stream head in absolute pressure (equ. 11). From equation 5 and intake efficiency, an expression defining head loss of the intake can be written (equ. 12). The intake efficiency is a unique physical characteristic of each inlet and ducting system. Figures 4 and 5 show intake efficiency characteristics for two typical inlet-ducting systems.

Figure 5 shows intake efficiency versus craft speed for various inlet velocity ratios. Speeds at which cavitation occurs in the intake system can be indicated on the intake performance curves. The example curves tend to suggest that inlet velocity ratios greater than one have high intake efficiencies. However, intake efficiency will reflect head loss due to friction and elevation effects only. When the inlet velocity is greater than the free stream velocity the head at the inlet has to be greater than the free stream head. That head differential will be a head loss to the pump. So, when the inlet velocity is greater than the free stream velocity, an extra head loss factor for the inlet differential must be included in the head in determining pump head (equ. 13).
The nozzle efficiency term, $Enz$, expresses the ratio of the head of the jet absolute to the head at the exit of the pump where the nozzle is at a constant elevation only (equ. 14). Nozzle loss calculated from equation 16 will reflect frictional loss in the nozzle only. Equation 16 was developed from equation 4 and nozzle efficiency (equ. 15). Normally nozzles are short to minimize losses and do not significantly change in elevation with respect to the eye of the pump. The eye of the pump coincides with the maximum elevation of the ducting system. Since all intake losses are accounted for at that maximum duct elevation point, any further elevation losses must be measured with respect to that datum point. As shown in figure 8, elevation deviation of the nozzle can be measured from the eye of the pump. This method of compensating for elevation deviation does not reflect the internal frictional losses exactly since the elevation deviation will cause internal flow in piping to be faster or slower than at constant elevation. But elevation deviation will normally be very small with respect to jet head anyway, so the discrepancy will be insignificant. Nozzle efficiencies of .995 are typical for short nozzles (nozzle length is no longer than twice the throat diameter) where Reynolds number is greater than 400,000.

The basic pump head equation (equ. 6) can be modified to accept the above special losses.

$$HD_{pmp} = HD_j - HD_{fs} + HL_{nz} + HL_{int} + HL_d + D_e \quad (17)$$

Total Thrust Required

The total thrust required should be equal to the total drag effect at design craft speed. The total drag of the craft will include external drag effects of the strut and pod of a ram type inlet or the drag of a semiflush lip. An inlet area on a planing surface may affect planing characteristics which may affect trim and drag. The internal drag of the ducting is compensated for in the intake loss calculations. Where a study may include various inlet and duct sizes and shapes, the effect on drag should be included in determining the overall thrust required at design speed (figure 9). A check should be made on displacement and drag effect of reaction forces due to turns in ducting systems (figure 10). The drag or displacement effect of ducting turns may be significant in some craft at certain speeds. Also, where thrust is vectored, the total thrust required will be the same as the reaction resultant necessary to meet the lift and drag components.
Figure 12 shows how head per stage and flow rate per stage are determined for the various pump types. Centrifugal pumps can be banked with several stages to give a lower flow rate per stage for a given pump head. Mixed-flow and axial pumps have the advantage of decreasing the head per stage at a given flow rate per pump. The pump performance curves of figure 13 were taken from pump technology (ref. 1, page 34) and indicate performance per stage. The performance curves indicate specific speeds at which the three types of pumps perform the best. Where a centrifugal pump is being considered for design at sustained conditions, the specific speed per stage should be between 500 and 2,000. Mixed-flow pumps should fall in specific speeds of 2,000 to 10,000. Axial pumps should operate at specific speeds of 10,000 to 15,000. If the pump type and geometry is fixed, the specific speed can be estimated from the appropriate pump geometry region. Knowing the flow per stage and the head per stage, the estimated pump rpm can be calculated from the estimated specific speed (equ. 19).

The net positive specific speed available at the eye of the pump is the absolute head available at the eye of the pump from previous calculations (equ. 5) minus the vaporization pressure of water (equ. 20). The cavitation factor \( \sigma \) is calculated from the net positive suction head and the head per stage of the pump (equ. 21) (see ref 1, page 35). With centrifugal pumps, \( \sigma \) applies for each stage since each stage must produce the total pump head. However, with axial and mixed-flow pumps, only the first stage "sees" the NPSH so \( \sigma \) applies to the first stage only. Using NPSH and the rpm estimated from pump specific speed the estimated suction specific speed per stage can be calculated (equ. 22). As can be seen from figure 14, where \( \text{SUCT} \) is less than 8,000, the pump will be in a safe design region. Where \( \text{SUCT} \) is between 8,000 and 12,000, the pump will be a special design. Pumps having an suction specific speed between 12,000 and 20,000 will be in a critical design region. No conventional pumps are designed with suction specific speeds greater than 20,000. \( \text{SUCT} \) will be an indication of the complexity of pump design involved and whether a conventional design is possible with the given configuration.

Impeller Diameter Estimation

Impeller diameter is estimated by generalizing conventional pump design for the various pump types. A typical blade discharge angle for the three types can be assumed to be between \( \beta_2 = 20 \) degrees and \( \beta_2 = 22.5 \) degrees (see ref 1, pages 49 and 50). Figure 15 shows non-dimensional performance curves used in pump design (see ref. 1, page 61).
Pump performance curves can be simplified for the two typical blade discharge angles and plotted as the head coefficient versus the pump specific speed (figure 16). The head coefficient $\psi$ is a function of the head per stage and the impeller tip speed ($U_2$) in feet per second (equ. 23). By functionalizing $\psi$, impeller diameter can be calculated from the impeller tip speed and the estimated impeller rpm (equ. 24). Equations exist where case diameter and case length are estimated from impeller diameter (reference 1, page 60). The casing volume is calculated from the casing dimensions and using a weight density factor for pump wet weight, the pump wet weight can be estimated (see Program Pring-out Appendix A.).
Computer Program

A computer program has been written which is based on the relationships developed in this paper. The computer program is written to allow entering a range of input so that output will show a trend when studying variations. The program is a basic tool; i.e., a mechanism which only needs accurate empirical intake performance data and thrust requirement data to give accurate head, flow and horsepower values. The computer program also allows for estimating more detailed pump characteristics for various types of pumps with various stages. RPM, sigma, suction specific speed and impeller diameter are rough cut estimates which can serve to find critical areas, limitations and generally, the type of pump which may be best suited for a particular craft at design speeds. Wet estimates for aluminum type pumps are also part of the output. Wet weight estimates are based on density factors which may be outdated, too conservative or over estimated since some computational comparisors have shown that the computed wet pump weight has been twice what an actual on the shelf pump might weigh. The computer program included in this paper has the density factors which have shown this error. As more realistic wet weight factors become available, or a better method of estimating the wet weight of the pump becomes available, that part of the computer program should be revised and updated. Refer to Appendix A for a sample run, a program print-out and explanation of terms. The values calculated by the program have been validated by hand calculations.
Conclusion

The relationships defined in this paper were developed to support the computer program which has been developed for analyzing waterjet system performance. Using this computer program, the inlet area sizing diagram and the sensitivity chart in Appendix B and intake performance characteristics as shown in figures 4 and 5, a detailed study can be conducted which could determine the most optimum system for a given design speed. Once the inlet area, ducting configuration and pump type are fixed the program can be used to check for performance characteristics and limitations at hump speed, full speed and other important craft speeds. The procedure and the program are still in the development stage and are being submitted as the basic pattern to be improved as design data becomes available.
ACKNOWLEDGEMENTS

Berg, David J.
Brandau, John H.
Corso, James
Hale, Malcolm
Johnson, Anthony
Lombardi, Paul
Jones, Walt
Kimble Nancy (typist)
Williamson, Carol (typist)
REFERENCES


4) "Waterjet Propulsion - A critical Survey of the State of the Art and a Recommended Research and Development Program". Technical Note 70, Hydromechanics Laboratory, Naval Ship Research and Development Center; April 1967

Sample Run

The computer program is presently on a time sharing system. As indicated by the sample run figures A1 and A2, on-line input happens to be one of the features of the system used. The question marks indicate readiness for input according to headings as programmed. Pump efficiency and transmission efficiency are inputs which are used to calculate a pump horsepower and a shaft horsepower. An estimated pump efficiency and transmission efficiency is entered when the exact value is not known. The program is arranged to calculate for any combination of inlets, pumps and jets the craft may use. Most of the headings are self explanatory however, the following terms might be clarified: VEL RATIO is jet velocity ratio, MIL PE/NZL is Reynolds number at the nozzle throat in millions, AVLHD FT is the gage pressure available at the eye of the pump in feet. IDL EF is ideal system efficiency, IMPX GAL/M is flow rate per stage in gallons per minute, WET WT is pump wet weight for an aluminum type pump in thousands of pounds.
### List of Terms in Computer Program

<table>
<thead>
<tr>
<th>Term</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>ARINL</td>
<td>Al</td>
</tr>
<tr>
<td>ARJET</td>
<td>Aj</td>
</tr>
<tr>
<td>AVLHD</td>
<td>HD avg</td>
</tr>
<tr>
<td>DC</td>
<td>Pump case dia</td>
</tr>
<tr>
<td>DIAM</td>
<td>Dtip</td>
</tr>
<tr>
<td>D2, D2, Dj2</td>
<td>Nozzles throat, Mean impeller dia squared</td>
</tr>
<tr>
<td>D22,</td>
<td></td>
</tr>
<tr>
<td>ED</td>
<td>Dpmp</td>
</tr>
<tr>
<td>EFIDL</td>
<td>Ideal system efficiency</td>
</tr>
<tr>
<td>EFINT</td>
<td>Efficiency</td>
</tr>
<tr>
<td>EFNZL</td>
<td>Enz</td>
</tr>
<tr>
<td>EFPMP</td>
<td>Epmop</td>
</tr>
<tr>
<td>EFTRN</td>
<td>Etr</td>
</tr>
<tr>
<td>EHP</td>
<td>Effective HP</td>
</tr>
<tr>
<td>ELOI, Q1</td>
<td>Flow rate ft³/min</td>
</tr>
<tr>
<td>FLO2, Q2</td>
<td>Flow rate lb/sec</td>
</tr>
<tr>
<td>FLO3, Q3</td>
<td>Flow rate gal/min</td>
</tr>
<tr>
<td>FLO4</td>
<td>gal/min per stage</td>
</tr>
<tr>
<td>FSHD</td>
<td>Hdf's</td>
</tr>
<tr>
<td>HDFPMP</td>
<td>HDpmp</td>
</tr>
<tr>
<td>HLD</td>
<td>Hld</td>
</tr>
<tr>
<td>HLINT</td>
<td>HLnt</td>
</tr>
<tr>
<td>HLNZL</td>
<td>Hlnz</td>
</tr>
<tr>
<td>ICKTL</td>
<td>control</td>
</tr>
<tr>
<td>MODE</td>
<td>control</td>
</tr>
<tr>
<td>NTYE</td>
<td>control</td>
</tr>
<tr>
<td>OPEN5, Cps</td>
<td>Number of inlets</td>
</tr>
<tr>
<td>PATM</td>
<td>Patm</td>
</tr>
<tr>
<td>PG</td>
<td>Propulsion Coefficient</td>
</tr>
<tr>
<td>PHP</td>
<td>Pump HP</td>
</tr>
<tr>
<td>PMFHD</td>
<td>Number of pumps</td>
</tr>
<tr>
<td>PMPS, Pmp(s)</td>
<td>Number of pumps</td>
</tr>
<tr>
<td>PNFPSH</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>REJET, Ray</td>
<td>Revolutions per min</td>
</tr>
<tr>
<td>RPM</td>
<td>Shaft HP per pump</td>
</tr>
<tr>
<td>SHUT, SHPt</td>
<td>Total SHP</td>
</tr>
<tr>
<td>SIG</td>
<td></td>
</tr>
<tr>
<td>SIG</td>
<td></td>
</tr>
<tr>
<td>STGS</td>
<td></td>
</tr>
<tr>
<td>SUBM</td>
<td>Number of stages per pump</td>
</tr>
<tr>
<td>SUCT</td>
<td>Suction specific speed</td>
</tr>
<tr>
<td>TR, T</td>
<td>Impeller tip speed</td>
</tr>
<tr>
<td>U2</td>
<td></td>
</tr>
<tr>
<td>VI</td>
<td></td>
</tr>
<tr>
<td>VIOVK</td>
<td></td>
</tr>
<tr>
<td>VJ</td>
<td></td>
</tr>
<tr>
<td>VJOVK</td>
<td></td>
</tr>
<tr>
<td>VK</td>
<td></td>
</tr>
<tr>
<td>VOL</td>
<td></td>
</tr>
<tr>
<td>WP</td>
<td>Wet weight of aluminum type pump</td>
</tr>
<tr>
<td>WL</td>
<td>Wet weight of aluminum type pump</td>
</tr>
<tr>
<td>WWAL</td>
<td>Wet weight of aluminum type pump</td>
</tr>
<tr>
<td>XIST, Tst</td>
<td>Pump case volume</td>
</tr>
<tr>
<td>XLC</td>
<td>Static thrust</td>
</tr>
<tr>
<td>XNS</td>
<td>Pump case length</td>
</tr>
<tr>
<td>Z</td>
<td>Axial pump length factor</td>
</tr>
</tbody>
</table>
IN *FIRE*

<table>
<thead>
<tr>
<th>UK(NTS)</th>
<th>INLETS</th>
<th>PUMPS</th>
<th>JETS</th>
<th>SUBM(FT)</th>
</tr>
</thead>
<tbody>
<tr>
<td>45</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>PUMP EFF</th>
<th>TRNSMN EFF</th>
<th>NOZZLE EFF</th>
<th>ELEV DEV(FT)</th>
</tr>
</thead>
<tbody>
<tr>
<td>.87</td>
<td>1.0</td>
<td>.995</td>
<td>0.0</td>
</tr>
</tbody>
</table>

**MODE=1 - ADDITIONAL ENTRY, =0 - LAST ENTRY**

<table>
<thead>
<tr>
<th>AREA/INL(FT2)</th>
<th>VI0USK</th>
<th>INLK EFF</th>
<th>TOT THRUST REQ'D(LBS)</th>
<th>MOGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>455</td>
<td>.7</td>
<td>.45</td>
<td>10628+.</td>
<td>1</td>
</tr>
<tr>
<td>54</td>
<td>.7</td>
<td>.45</td>
<td>10000+</td>
<td>0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>A/INL</th>
<th>VI</th>
<th>VEL</th>
<th>VJ</th>
<th>JET</th>
<th>A/USK</th>
<th>MIL RE/</th>
</tr>
</thead>
<tbody>
<tr>
<td>FT2</td>
<td>FT/S</td>
<td>RATIO</td>
<td>FT/S</td>
<td>EFF</td>
<td>FT2</td>
<td>NUSK</td>
</tr>
<tr>
<td>.45</td>
<td>53-20</td>
<td>2-45</td>
<td>186-29</td>
<td>5795</td>
<td>.13</td>
<td>5-87</td>
</tr>
<tr>
<td>.54</td>
<td>53-20</td>
<td>2-24</td>
<td>170-43</td>
<td>6164</td>
<td>.17</td>
<td>6-12</td>
</tr>
</tbody>
</table>

**PER PUMP VALUES**

<table>
<thead>
<tr>
<th>A/INL</th>
<th>FLO RT</th>
<th>FLO HT</th>
<th>FLO RT</th>
<th>IDEAL STAT</th>
</tr>
</thead>
<tbody>
<tr>
<td>FT2</td>
<td>LBS/S</td>
<td>FT3/M</td>
<td>GAL/M</td>
<td>THRUST-LB</td>
</tr>
<tr>
<td>.45</td>
<td>1550-25</td>
<td>1452-40</td>
<td>10863-98</td>
<td>8976-01</td>
</tr>
<tr>
<td>.54</td>
<td>1839-85</td>
<td>1723-73</td>
<td>12893-52</td>
<td>9746-12</td>
</tr>
</tbody>
</table>

**FREE STREAM HEAD (FT-CAGE)=** 91.7674

<table>
<thead>
<tr>
<th>A/INL</th>
<th>NUSK</th>
<th>INLK</th>
<th>AVL HD</th>
<th>PMP HD</th>
<th>NPSH</th>
<th>WHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>FT2</td>
<td>LOS-FT</td>
<td>LOS-FT</td>
<td>FT</td>
<td>FT</td>
<td>FT</td>
<td>HP</td>
</tr>
<tr>
<td>.45</td>
<td>2-88</td>
<td>69-12</td>
<td>22-64</td>
<td>519-55</td>
<td>54-75</td>
<td>1464-41</td>
</tr>
<tr>
<td>.54</td>
<td>2-44</td>
<td>69-12</td>
<td>22-64</td>
<td>431-20</td>
<td>54-75</td>
<td>1442-46</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>A/INL-FT2</th>
<th>EHP</th>
<th>PHP/PMP</th>
<th>SHP TOTL</th>
<th>PC</th>
<th>IDL EFF</th>
</tr>
</thead>
<tbody>
<tr>
<td>.45</td>
<td>1468-64</td>
<td>1683-23</td>
<td>3366-46</td>
<td>.4363</td>
<td>.5014</td>
</tr>
<tr>
<td>.54</td>
<td>1492-41</td>
<td>1458-06</td>
<td>3316-81</td>
<td>.4501</td>
<td>.5173</td>
</tr>
</tbody>
</table>

**ENTER CONTROL=1 FOR RPM AND DIA ESTIMATES, CONTROL= 2 TO SKIP HEADING, CONTROL= 0 TO STOP PROGRAM.**

? 1
DESIGN ESTIMATES BASED ON CONVENTIONAL PUMP PERFORMANCE

CENTRIFUGAL MULTISTAGE PUMPS (TYPE 1)
ASSUMPTIONS: BETA=22.5 DEG, CASE DIA=1.90X IMPLR DIA,
CASE VOL BASED ON DOUBLE SUCTION IMPLRS

MIXED-FLOW PUMPS (TYPE 2)
ASSUMPTIONS: BETA=22.5 DEG, CASE DIA=1.50X IMPLR DIA
ONE STAGE ONLY

AXIAL-FLOW PUMPS (TYPE 3)
ASSUMPTIONS: BETA=20 DEG, CASE DIA=1.20X IMPELLER DIA
6 BLADES/STAGE

PUMP WT WT WILL BE 167 PER CENT OF THE ALUMINUM WT WT
FOR STAINLESS STEEL OR OTHER BASIC PUMP MATERIALS

VOL AND WT ARE BASED ON THE ESTIMATED DIAMETER
KLB IS THOUSANDS OF LBS
NS=0 STOPS PROGRAM

<table>
<thead>
<tr>
<th>TYPE</th>
<th>NS</th>
<th>IMPLRS OR STAGES / PUMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1400</td>
<td>2</td>
</tr>
</tbody>
</table>

A/INL SIG IMPLR RPM SUCT IMPLR CASE WET WT
FT2 GAL/M SP SPD DIA-FT VOL-FT3 ALM-KLB
45 1054 5431.99 2067.13 7568.85 1.68 29.34 2.79
54 1270 6446.76 1649.95 6581.49 1.92 43.62 4.15

<table>
<thead>
<tr>
<th>TYPE</th>
<th>NS</th>
<th>IMPLRS OR STAGES / PUMP</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>
Calculation Procedure

Input:

\( V_k (\text{Kts}), \ O_{\text{pns}}, \ P_{\text{mps}}, \ I_{\text{s}}, \)
\( \text{Subm}, \ EF_{\text{noz}} > D_o, \ A_1 \) (area per inlet)
\( f_i, \ E_{\text{int}}, \ Tr, \)

Calculations:

1) \( V_i = \left( V_i / V_k \right) (V_k \times 1.688) \)
\( V_i = f_i V_k \times 1.688, \) inlet velocity

2) \( T_r = \rho Q (V_j - V_k) \)
\( Q = AV = A_1 V_i = A_2 V_{nz} \)
\( T_r = \rho \left( \text{total inlet area} \right) V_i (V_j - V_k) \)
\( T_r = \rho \ O_{\text{pns}} x A_1 V_i (V_j - V_k) (V_k) \)
\( V_j / V_k - 1 = T_r / (\rho \ O_{\text{pns}} A_1 V_i V_k \times 1.688) \)
\( V_j / V_k = 1 + T_r / (\rho \ O_{\text{pns}} A_1 V_i V_k \times 1.688) \)
jet velocity ratio

3) \( V_j = \left( V_j / V_k \right) (V_k \times 1.688), \) jet velocity

4) \( E_j = \frac{2}{1 + V_j / V_k}, \) jet efficiency

5) \( A_j = Q / V_j = A_1 O_{\text{pns}} V_i / (V_j \times Jts) \)
\( A_j = A_1 O_{\text{pns}} / (Jts \times V_j / V_k), \)
throat area per nozzle

6) \( A_j = \pi D_j^2 / 4 \)
\( D_j = 4A_j / \pi, \) nozzle dia

\( \text{Rey} = \frac{V_i D_j}{\nu}, \) reynolds no.

\( \nu = 12.9 \times 10^{-6} \text{ ft}^2/\text{sec} @ 59^\circ \text{F sea water} \)
7) $Q_1 = 60. A_i V_{i\text{ns}} V_i/P_{\text{mps}}$, (ft$^3$/min) 
flow rate per pump 

$Q_2 = Q_1 \, y/60.$, (lbs/sec) 

$Q_3 = Q_2 (7.48)$, (gal/sec) 

8) $T_{st} = Q (V_i-0)$, ideal static thrust 

9) $HD_{fs} = (V_k \times 1.688)^2/2g + S_{hbm}$ 
free stream HD 

10) $HL_{nz} = (1-E_{nz}) (V_j^2/2g + P_{atm})/E_{nz}$ 

$HL_{ns}/P_{mp} = HL_{nz} J_{ts}/P_{mps}$ 
nozzle head loss per pump 

11) $HL_{int} = (1-E_{int}) (HD_{fs} + P_{ajm})$ 

$HL_{int}/P_{mp} = HL_{int} \times 0_{\text{pns}}/P_{mps}$ 
intake head loss per pump 

12) When $V_i > V_k \times 1.688$: 

$HL_d = (V_i^2/2g - (V_k \times 1.688)^2/2g$ 

$HL_d/P_{mp} = HL_d \times 0_{\text{pns}}/P_{mps}$ 
inlet differential head loss per pump 

13) $HD_{avg}/P_{mp} = HD_{fs} - HL_{int}/P_{mp} - HL_d/P_{mps}$ 
head available (gage) per pump 

14) $D_e/P_{mp} = D_e J_{ts}/P_{mps}$ 
elevation deviation head per pump 

15) $HD_{pmp} = V_j^2/2g + HL_{nz}/P_{mp} - HD_{avg}/P_{mp}$ 

$+ D_e/P_{mp}$ 
head differential each pump will have to provide
16) NPSH = HD_{avg} + P_{atm} - P_{vap} \\
HD_{avg} + 32.11 \\
net positive suction head

17) WHP = Q_1 \times HD_{pmp}/550, water - horsepower or the 
ideal horsepower per pump

18) EHP = T \times (V_k \times 1.688)/550 
effective horsepower or resistance horsepower

19) E_{id1} = EHP/(P_{mps} \times WHP), ideal system efficiency

20) PHP = WHP/E_{pmp}, pump horsepower

21) SHP = PHP/E_{tr} shaft horsepower per pump

22) SHP_t = SHP \times P_{mps}, total SHP

23) PC = EHP/SHP_t

At this point, the type of pump, the pump specific speed and 
the number of stages per pump must be entered. Head and flow rate 
per stage is then calculated according to the pump type as shown in 
Figure 12. RPM is then calculated from equation 19. NPSH is 
retrieved from previous calculations and used to calculate sigma 
and SUCT as shown in equations 21 and 22. Then using a function 
for si versus pump specific speed, tip speed can be calculated (U_2) 
and used in calculating impeller diameter as illustrated in figures 
15 and 16. From the impeller diameter, casing diameter and casing 
length are calculated according to size estimation factors given in 
reference 1 (page 60). Refer to the computer program print-out for 
the size relationships used. The computer program is simple to read 
and is straightforward in arrangement.
020  WJPB (WATER-JET PUMP PART A) CASDAC NO.-
030  PREDICTS WATER-JET PUMP REQUIREMENTS FOR A GIVEN APPLICATION
040  AND SHOULD BE USED WITH WJPB FOR RPM AND DIA ESTIMATES
10  DIMENSION ARINL(20), VI0V(20), EFINT(20), TR(20)
20  DIMENSION VI(20), VJ(20), ARJET(20), FLO2(20)
30  DIMENSION FLO3(30), PNPSH(20), HDPMP(20), WHP2(20)
35  DIMENSION EFJET(20)
40  CONV=1.68894
45  PI=3.14159
50  VISC=12.9
55  DENS=64.042
70  VISC=12.9
57  PATM=33.91
60  RHOm=0.9905
65  VISC=12.9
70  G=32.174
75  RHO=1.9905
80  PRINT":" " VK(KTS) INLETS PUMPS JETS SUBM(FT)"
90  INPUT, VK, OPENS, RPM, XJTS, DENS, PATM, RHOm, ED
100  INPUT, VK, OPENS, RPM, XJTS, SUBM
110  CONV=1.68894
120  PRINT": " PUMP EFF TRANSMN EFF NOZZLE EFF ELEV DEV(FT)"
130  INPUT, EFPPM, EFTRN, EFNEL, ED
140  I=1
150  PRINT":" " MODE=1- ADDITIONAL ENTRY, =0- LAST ENTRY"
160  PRINT":" " AREA/INL(FT2), VK, TR, MODE"
170  I=N=1
180  IF(MODE) 20, 10, 10
190  DO 25 J=1, N
200  VI(J)=VI0V(J)*CONV
210  VJ(J)=VJ0V(J)*CONV
220  TR(J)=TR(J)/RHO*ARINL(J)*OPENS*VI(J)/VK*CONV
230  EFJET(J)=2./RHO/VJ(J)
240  ARJET(J)=ARINL(J)*OPENS/VJ(J)*XJTS/VI(J)
250  D2=4.*ARJET(J)/PI
260  REJET=VJ(J)*SQRT(2)/VISC
270  PRINT 3,ARINL(J),VI(J), VJ0V(J), VJ(J), EFJET(J), ARJET(J), REJET
280  PRINT":" " PER PUMP VALUES"
290  PRINT":" " AREA/INL, FLO RT, FLO RT"
300  PRINT":" " FT2, FT2, LBS/S, LBS/S"
310  PRINT":" " GAL/M, THRUST-LB"
320  DO 30 J2=1, N
330  FLO1=60.*ARINL(J2)*OPENS*VI(J2)/PMPS
340  FLO2(J2)=FLO1*DENS/60.
350  FLO3(J2)=FLO1*7.48
360  XIST=FLO2(J2)*VJ(J2)/G
370  PRINT 4, ARINL(J2), FLO2(J2), FLO1, FLO3(J2), XIST
FSHD=(VK*CONV)**2/(2.*G)+SUBM
PRINT " FREE STREAM HEAD (FT-GAGE)=" , FSHD
PRINT " A/INL N2L INTK AVL HD PMP HD
+ NPSH WHP"
PRINT " FT2 LOS-FT LOS+FT FT FT
+ FT HP"
DO 40 K=1,N
HLN2L=(1.-EFNEL)*(VJK)**2/(2.*G)+PATM*XJTS/(PMP*EFNEL)
HLINT=(1.-EFINT(K))*(FSHD+PATM)*OPENS/PMP
HLD=0.0
B=VI(K)**2
C=(VI*CONV)**2
IF(B-C) 37, 37, 35
35 HLD=(B-C)*OPENS/(2.*G)*PMPS
37 AVLHD=FSHD-FLINT-HLD
HDMP(K)=VJK**2/(2.*G)-AVLHD+HLN2L+ED*XJTS/PMP
PNDSH(K)=AVLHD+32.11
WH2(K)=HDMP(K)*FLO2(K)/550.
40 IF(8-C) 37, 37, 35
PRINT 5, ARINL(K),HLN2L,HLINT,AVLHD,HDMP(K),PNDSH(K),WH2(K)
PRINT " A/INL-FT2 EHMP PHP/PMP SHP TOTL
+ PC IDL EFF"
49 DO 45 L=1,N
50 EHMP=TR(L)*VK*CONV/550.
51 PHP=WH2(L)/EFTRN
52 SHP=PHP/EFTRM
53 SHPT=SHP*PMPS
54 PC=EHMP/SHPT
55 EFIDL=EHMP/(WH2(P)*PMPS)
45 IF(8-C) 37, 37, 35
PRINT 7, ARINL(L), EHMP, PHP, SHPT, PC, EFIDL
PRINT " ENTER CONTROL=1 FOR RPM AND DIA ESTIMATES, CONTROL=
+ 2 TO SKIP"
PRINT " HEADING, CONTROL=0 TO STOP PROGRAM. CONTROL ">
INPUT *, ICNTL
60 IF(ICNTL-1) 47, 48, 50
50 CONTINUE
62 PRINT " DESIGN ESTIMATES BASED ON CONVENTIONAL PUMP PERFORMANCE"
63 USE WPJB
64 IF(CNTL) 47, 48, 50
70 IF(CNTL=1) 47, 48, 50
70 3 FORMAT(4F10.2, F10.4, 2F10.2)
71 4 FORMAT(SF14.2)
72 5 FORMAT(7F10.2)
73 7 FORMAT(4F11.2,2F11.4)
74 9 FORMAT(F52.2, FS*.4, 6F10.2)
WJPB (WATER-JET PUMP PART B) CASDAC NO.-
MUST BE USED WITH WJPA TO ESTIMATE RPM AND DIA FOR WATER-
JET PUMPS BASED ON CALCULATIONS FROM WJPA.

A. F. GARCIA NAVSEC 6144 10 JUNE 1969

PRINT "CENTRIFUGAL MULTISTAGE PUMPS (TYPE 1)"
ASSUMPTIONS: BETA=22.5 DEG, CASE DIA=1.90
+X IMPLR DIA,
CASE VOL BASED ON DOUBLE SUCTION IMPLRS
PRINT "MIXED-FLOW PUMPS (TYPE 2)"
ASSUMPTIONS: BETA=22.5 DEG, CASE DIA=1.50
+X IMPLR DIA
ONE STAGE ONLY
PRINT "AXIAL-FLOW PUMPS (TYPE 3)"
ASSUMPTIONS: BETA=20 DEG, CASE DIA=1.20X
+ IMPELLER DIA
6 BLADES/ STAGE
PRINT "PUMP WET WT WILL BE 167 PER CENT OF THE ALUMINUM
+WET WT"
PRINT "FOR STAINLESS STEEL OR OTHER BASIC PUMP MATERIALS"
PRINT "VOL AND WT ARE BASED ON THE ESTIMATED DIAMETER"
PRINT "KLB IS THOUSANDS OF LBS"
PRINT "NS=0 STOPS PROGRAM"
220 50 PRINT " TYPE NS IMPLRS OR STAGES / PUMP"
230 INPUT NTYPE, XNS, STGS
240 IF (XNS=500) 95, 55, 55
250 55 PRINT "A/INL SIG IMPLR RPM SUCT IMPLR"
260 CASE WET WT"
270 PRINT " FT2 GAL/M SP SPD DIA-FI"
280 VOL-FT3 ALM-XLB"
290 DO 90 M=N
300 PMPHD=HDPMP(M)
310 IF (NTYPE=2) 65, 65, 60
320 60 PMPHD=PMPHD/STGS
330 FLO4=FLO3(M)
340 65 SIG=PNPSH(M)/PMPHD
350 RPM=XNS*(PMPHD**.75)/FLO4**.5
360 SUCT=RPM*(FLO4**.5)/(PNPSH(M)**.75)
370 X=XNS/1080.*
380 SI=.5873-X*7.1528E-2+(X**2)*3.8194E-3-(X**3)*6.944E-5
385 + +.01
390 DZ=PMPHD/6/SI
400 DIAM=SQRT(U2)*60./(PI*RPM)
405 IF (NTYPE=2) 70, 72, 75
410 70 DC=1.5*DIA M
420 XLC=STGS*1.5*DIAM/2.5*DIAM
430 WC=.855
440 GO TO 85
460 72 DC=1.5*DIA M
470 XLC=2.7*DIAM
480 WC=.841
490 GO TO 85
500 75 D22=2.5*DIAM**2/25
510 DC=364*.80*PI*SQRT(D22)/6.
520 DC=1.2*SQRT(D22)
530 XLC=2.4*PI*STGS+2.2*SQRT(D22)
540 WC=.839
550 85 VOL=XLC*PI*DC**2/4.
560 WWAL=VOL*WC*.1+728
570 PRINT 9. ARINL(M), SIG, FLO4, RPM, SUCT, DIAM, VOL, WWAL
580 90 CONTINUE
590 GO TO 50
600 95 CONTINUE
Inlet Area Sizing

A method of estimating realistic inlet area for certain craft is necessary since an infinite range of inlet areas and hence flow rates in combination with appropriate jet velocities could exist to meet the required thrust. From the thrust and flow equations (1) and (2):

\[ T_r = \rho Q(V_j - V_k) \]
\[ Q = A_i V_i, V_i = f_1 V_k \]
\[ T_r = \rho f_1 A_i V_k (V_j - V_k) \]
\[ T_r = \rho f_1 A_i V_k^2 (V_j/V_k - 1) \]
\[ T_r/f_1 A_i = \rho V_k^2 (R-1) \]

If:

Total thrust/f_1 (Total inlet area) = \rho V_k^2 (R-1)

Then:

Thrust per nozzle/f_1 (Total inlet area per nozzle) = \rho V_k^2 (R-1)

\( \varphi \) is then a function of the inlet area required per jet nozzle on the craft to meet a particular thrust. \( \varphi \) is also a function of jet velocity and jet velocity ratio. By generating a family of curves of \( \varphi \) versus ship speed and jet velocity ratio, inlet area can be determined from \( \varphi \) where the design speed and design jet velocity ratio are known and a certain thrust per nozzle is required (fig. B.1). Regions of typical design velocities and design jet velocity ratios for certain waterjet propelled craft can be shown on figure B.1. The conventional design regions can be used for "first-cut" estimations of inlet area from \( \varphi \) for specific craft. The design regions for the various craft as shown in figure B.1 are general and based on limited information and should be revised as more exact design jet velocity ratios are known at design speeds for the various craft.
Sensitivity Trends

The required pump head differential and flow rate are sensitive to the jet velocity ratio. It can be shown that for low jet velocity ratio \( R \to 1 \) the ideal jet efficiency will be high, but the flow rate required will become very high and the pump head differential will become unrealistically low. However, as the jet velocity ratio increases, the ideal jet velocity ratio will drop so as \( R \to 3 \), \( \varepsilon_j = 0.5 \) An ideal jet efficiency of fifty per cent may be undesirably low for design cruising speeds for most craft. For most applications then, \( R \) will be between one and three.

\[ \varepsilon_j = \frac{2}{R+R} \]

From equations (2) and (4):

\[ T_r = Q (V_j - V_k) \]

\[ T_r = A_{fi} V_k^2 (R-1) \]

If \( T_r \) and \( V_k \) are constant,

\[ A_{fi} \approx \frac{f_1}{(R-1)} \]

And

\[ Q = V_1 A_{fi} = f_1 V_k A_{fi} \]

So,

\[ Q \approx \frac{f_1 V_k}{f_1(R-1)} \]

And since \( V_k \) is constant,

\[ Q \approx \frac{1}{(R-1)} \]

The equation for pump head (equ. 6) can be simplified to be:

\[ H_{dpump} = V_j^2/2g + E_{int} (V_k^2/2g) + C \]

"C" is miscellaneous head loss such as nozzle submergence, and deviation loss which tends to be small and constant. If "C" can be considered insignificant with respect to the total head the pump must produce, let \( C = 0 \) then:
\[
\text{HDpmp} = \frac{1}{2g} \left( \frac{v_j^2}{2g} + \text{Eint} \frac{v_k^2}{2g} \right)
\]

\[
\text{HDpmp} \approx R^2 = \text{Eint}
\]

Now,

\[
\text{WHP} = Q \times \text{HDpmp}
\]

From the specific speed equation for pumps, the RPM can be estimated (equ. 19):

\[
\text{RPM} = \frac{N_s \text{HDpmp}^{.75}}{\sqrt{\text{gpm}}}
\]

But gpm is just flow rate Q, so if \(N_s\) is constant:

\[
\text{RPM} \approx \frac{\text{HDpmp}^{.75}}{\sqrt{Q}}
\]

The above equation shows the effect of head and flow rate on pump RPM where pump specific speed is kept constant (pump geometry, or the pump type is kept constant). Suction specific speed can then be related to the RPM (equ. 22):

\[
\text{SUCT} = \frac{\text{RPM} \sqrt{\text{gpm}}}{(\text{NPSH})^{.75}}
\]

or:

\[
\text{SUCT} = \frac{N_s \text{HDpmp}^{.75} \sqrt{\text{gpm}}}{\sqrt{\text{gpm}} (\text{NPSH})^{.75}}
\]

Then if NPSH is constant:

\[
\text{SUCT} \approx \text{HDpmp}^{.75}
\]

Also from equation 21:

\[
\sigma = \frac{\text{NPSH}}{\text{HDpmp}}
\]

so:

\[
\sigma \approx \frac{1}{\text{HDpmp}}
\]
The above relationships can be depicted graphically as shown in figure B.2 as relative change in magnitude versus change in jet velocity ratio. Trends in head, flow, ideal horsepower, pump RPM, and suction specific can be seen as the design jet velocity ratio changes. The thrust required, pump specific speed and net positive suction speed must be held constant which means that duct configuration and the pump type (pump geometry) is kept constant. See the following Sensitivity Table as an example of the effect of changing a system from a design velocity ratio of 2 to 3.

Sensitivity Table

<table>
<thead>
<tr>
<th></th>
<th>R = 2</th>
<th>R = 3</th>
<th>% change from R = 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_1, f_1 = 1.0$</td>
<td>1</td>
<td>.5</td>
<td>50%</td>
</tr>
<tr>
<td>$Q'$</td>
<td>1</td>
<td>.5</td>
<td>50</td>
</tr>
<tr>
<td>HDpump, Eint = 1.0</td>
<td>3</td>
<td>8</td>
<td>266</td>
</tr>
<tr>
<td>WHP</td>
<td>3</td>
<td>4</td>
<td>133</td>
</tr>
<tr>
<td>RPM</td>
<td>2.3</td>
<td>6.1</td>
<td>265</td>
</tr>
<tr>
<td>SUCT</td>
<td>2.3</td>
<td>4.8</td>
<td>207</td>
</tr>
<tr>
<td>$\sigma_1$</td>
<td>.5</td>
<td>.39</td>
<td>.38</td>
</tr>
</tbody>
</table>
BASIC WATERJET EQUATIONS

\[ Q = A_f V_f \]  \hspace{1cm} (1)

\[ T_f = \rho Q (V_f - V_k) \]  \hspace{1cm} (2)

\[ H_{D_{\text{ex}}} = H_{D_{\text{j}}} + H_{L_{\text{nz}}} + P_{\text{atm}} \]  \hspace{1cm} (3)

\[ H_{D_{\text{av}}} = H_{D_{\text{fs}}} - H_{L_{\text{int}}} + P_{\text{atm}} \]  \hspace{1cm} (4)

\[ H_{D_{\text{pmp}}} = H_{D_{\text{j}}} + H_{L_{\text{nz}}} + P_{\text{atm}} - (H_{D_{\text{fs}}} - H_{L_{\text{int}}} + P_{\text{atm}}) \]  \hspace{1cm} (5)

\[ H_{D_{\text{pmp}}} = H_{D_{\text{j}}} - h_{D_{\text{fs}}} + H_{L_{\text{nz}}} + H_{L_{\text{int}}} \]  \hspace{1cm} (6)

\[ H_{D_{\text{j}}} = \frac{V_{j}^2}{2g} \]  \hspace{1cm} (7)

\[ H_{D_{\text{fs}}} = \frac{V_{k}^2}{2g} + \text{Subm} \]  \hspace{1cm} (8)

FIG. 1

27
**FIG. 2**

**INTAKE EFFICIENCY:**

\[
\dot{E}_{int} = \frac{H_{Dav}}{V_{k}^2/2g + S_{ubm} + P_{atm}} = \frac{H_{Dav}}{H_{Dfs} + P_{atm}}
\]

\[
H_{Dav} = \dot{E}_{int} (H_{Dfs} + P_{atm})
\]

FROM EQU. (5)

\[
H_{Lint} = H_{Dfs} + P_{atm} - H_{Dav}
\]

\[
H_{Lint} = (1 - \dot{E}_{int}) (H_{Dfs} + P_{atm})
\]

\[
H_{Lint} = (1 - \dot{E}_{int}) (H_{Dfs} + P_{atm})
\]

**FIG. 3**
FLUSH & SEMIFLUSH PERFORMANCE CURVES

FIG. 4
STRUT INTAKE PERFORMANCE CURVES

FIG. 5
HEAD DIFFERENTIAL LOSS AT INLET

WHEN \( V_i > V_k \) :

\[
\frac{V_i^2}{2g} > \frac{V_k^2}{2g}
\]

\[
HL_d = \frac{V_i^2}{2g} - \frac{V_k^2}{2g}
\]

\[
HL_d = \frac{V_k^2}{2g} \left( \frac{V_i^2}{V_k^2} - 1 \right)
\]

\[
HL_d = \frac{V_k^2}{2g} \left( f_i^2 - 1 \right)
\]

FIG. 6
NOZZLE LOSS

\[ \text{NOZZLE} \]

\[ \text{PUMP} \]

\[ \text{HDj} + \text{Patm} \]

\[ \text{HDex} \]

NOZZLE EFFICIENCY:

\[ \text{Enz} = \frac{\text{HDj} + \text{Patm}}{\text{HDex}} \] \hspace{1cm} (14)

\[ \text{HDex} = \frac{\text{HDj} + \text{Patm}}{\text{Enz}} \] \hspace{1cm} (15)

FROM EQU. (4)

\[ \text{HLnz} = \text{HDex} - (\text{HDj} + \text{Patm}) \]

\[ \text{HLnz} = \frac{\text{HDj} + \text{Patm}}{\text{Enz}} - (\text{HDj} + \text{Patm}) \]

\[ \text{HLnz} = (\text{HDj} + \text{Patm})(1 - \text{Enz}) / \text{Enz} \] \hspace{1cm} (16)

FIG. 7

ELEVATION DEVIATION OF THE NOZZLE

\[ \text{De} \quad + \text{De} \]

\[ \text{PUMP} \]

FIG. 8

\[ \text{De} \quad - \text{De} \]
$T_t = \text{TOTAL DRAG AT DESIGN } V_k$

**FIG. 9**

$T_t = \text{REACTION} = \sqrt{\text{LIFT}^2 + \text{DRAG}^2}$

**FIG. 11**
HEAD AND FLOW RATE PER STAGE

I. CENTRIFUGAL PUMPS

\[ A. \ Q_t = Q_1 + Q_2 + Q_3 + Q_4 \]
\[ B. \ Q_s = \frac{Q_t}{\text{Stgs}} \]
\[ C. \ HD_s = HD_1 = HD_2 = HD_3 = HD_4 = HD_t \]

II. SINGLE-STAGE MIXED FLOW PUMPS

\[ A. \ Q_s = Q_t \]
\[ B. \ HD_s = HD_t \]

III. AXIAL PUMPS

\[ A. \ Q_s = Q_1 = Q_2 = Q_t \]
\[ B. \ HD_t = HD_1 + HD_2 \]
\[ C. \ HD_s = \frac{HD_t}{\text{Stgs}} \]

FIG. 12
**SPECIFIC SPEED VS PUMP GEOMETRY**

![Graph showing specific speed vs pump geometry](image)

- **Equations**:
  1. \[ N_s = \text{RPM} \sqrt{\text{gpm}/\text{HD}_{0.75}} \]  
  2. \[ \text{RPM} = \frac{N_s \text{HD}_{0.75}}{\sqrt{\text{gpm}}} \]

**FIG. 13**

---

![Diagram showing cavitation](image)

- **Equations**:
  1. \[ \text{NPSH} = \text{H}_{\text{av}} - P_{\text{vap}} \]  
  2. \[ \sigma = \frac{\text{NPSH}}{\text{HD}_0} \]  
  3. \[ \text{SUCT} = \frac{\text{RPM} \times \text{gpm}}{\text{NPSH}^{0.75}} \]

**CAVITATION**

**FIG. 14**
\[ \gamma = \frac{H_d}{2} \cdot \frac{1}{g} \]  

\[ U_2^2 = g \cdot H_d / \gamma, \quad \gamma \text{ a function of } N_s \]

\[ D_{\text{imp}} = U_2 \times \frac{60}{\pi \times \text{RPM}} \]

FIG. 15

FIG. 16
WATERJET INLET
SIZING GUIDE

\[ \tau = \frac{\text{Thrust/nozzle}}{f_i \times \text{Inlet area/nozzle}} \]

\[ f_i = \frac{V_i}{V_k} \]

\[ R = \frac{V_j}{V_k} \]

FIG. B.1
THE RELATIVE EFFECT OF R

$E_i = \frac{2}{1+R}$

$E_{int} = 50\%$

$E_{int} = 100\%$

$HD_{pmp} \approx R^2 - E_{int}$

$WHP = HD_{pmp} \times Q$

$Q \approx \frac{1}{R-1}$

$A_i \approx \frac{1}{(R-1)f_i}$

$A_i, f_i = 0.5$

$A_i, f_i = 1.0$

$\sigma, SUCT & RPM$

AT a CONSTANT $N_5$

& a CONSTANT NPSH

FIG. B.2