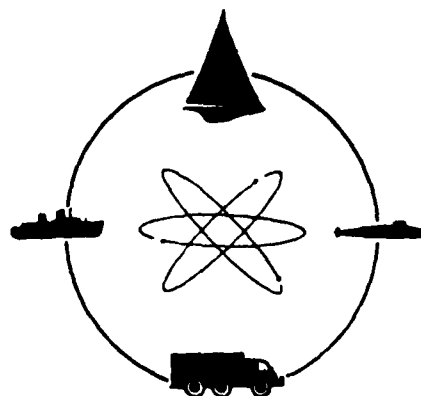
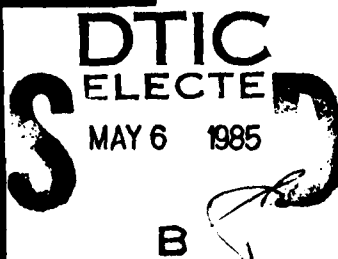




DAVIDSON LABORATORY

Technical Report SIT-DL-85-9-2328
March 1985

DEVELOPMENT OF **WATERJET** PROPULSION UNIT



by

John K. Roper

Prepared for

Code 112

David W. Taylor

Naval Ship Research and Development Center

Under

Office of Naval Research

Contract N00014-80-D-0890

Delivery Order 4, Item 1

(DL Project 4982/134)

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Technical Report SIT-DL-85-9-2328

March 1985

DEVELOPMENT OF **WATERJET** PROPULSION UNIT

By

John K. Roper

for

David W. Taylor Naval Ship Research and Development Center
Code 1120

under

Office of Naval Research
Contract N00014-80-D-0890
Delivery Order 4, Item 1
DL Project 4982/134

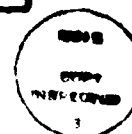
APPROVED: 

Daniel Savitsky
Director

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SUMMARY

Designs of three waterjet propulsion systems have been developed for application to amphibious vehicles. The first system was an axial-flow pump designed to be used in an existing amphibious vehicle, an LVTP-7A1 and it shows significant advantages over the existing waterjet unit in efficiency, thrust output and system weight.

The other two systems were to be used to propel a proposed high-speed amphibious vehicle. These pumps were designed to provide cavitation-free performance at propulsive coefficients in the region of 40 to 45 percent at a vehicle water speed of 20 mph. State-of-the-art composite material technology was used wherever possible to reduce weight.

INTRODUCTION

The U. S. Marine Corps plans to improve the mobility of amphibious vehicles. One aspect which requires attention is the need to increase the efficiency of existing waterjet propulsion units, along with improving the durability and reducing the cost of these units.

Existing waterjet installations in typical amphibious vehicles have low efficiencies due to numerous design constraints associated with their present stern locations. One of the contributors to their relatively poor performance is the location of the water intakes to an area which is seriously obstructed by the tracks. Although this is but one influence on total performance, it is desirable to define the sources of blockage, interference, ventilation, etc., in the intake area of presently installed waterjets and to then use these results to recommend suitable design changes which will ameliorate the undesirable effects.

It is also appropriate to consider a redesign of the present waterjet units using modern high strength plastic materials being developed by AMRAC at the Watertown Arsenal. These plastics should have a high resistance to erosion by sand or debris which can pass through the impeller and hence result in a more durable and potentially less costly propulsion unit.

This design effort proceeded through the following phases:

1. Feasibility study of composite plastic waterjet propulsion unit.
2. Design study of a waterjet unit with improved propulsion efficiency for:
 - a. An existing slow-speed-in-water amphibious vehicle
 - b. A proposed high-speed-in-water vehicle

FEASIBILITY STUDY OF
COMPOSITE PLASTIC WATERJET UNIT

In order to evaluate the practicality of constructing a composite plastic waterjet pump and to obtain expert advice on likely materials and fabrication methods, discussions were held with Mr. A. Alisio of the Army Materials Research Laboratory of Watertown, MA and Mr. A. Macander of the Naval Research & Development Center at Annapolis, MD.

There is no doubt that the concept is well within the state-of-the-art. Indeed, similar components such as composite plastic pump casings, impellers and large valves are in production and are used extensively in many industries. The question is whether the tooling costs, which are likely to be high, can be justified by the small number of units to be produced. This question can only be answered by obtaining cost estimates from manufacturers for specific components.

On the basis of advice received so far, the propeller duct, Figure 1, would be layed up of "pre-preg" fabric over a male mold or perhaps filament-wound. The propeller, support strut, rudder bearings and other small parts would be injection or transfer molded.

While composites of carbon fiber with polyurethane resin have been recommended because of their stiffness and abrasion resistance, it appears that components similar to those shown in Figure 1, have been fabricated successfully using glass fibers with a variety of other resins such as acetal, polycarbonate, epoxy, etc., and that these composites should be investigated further.

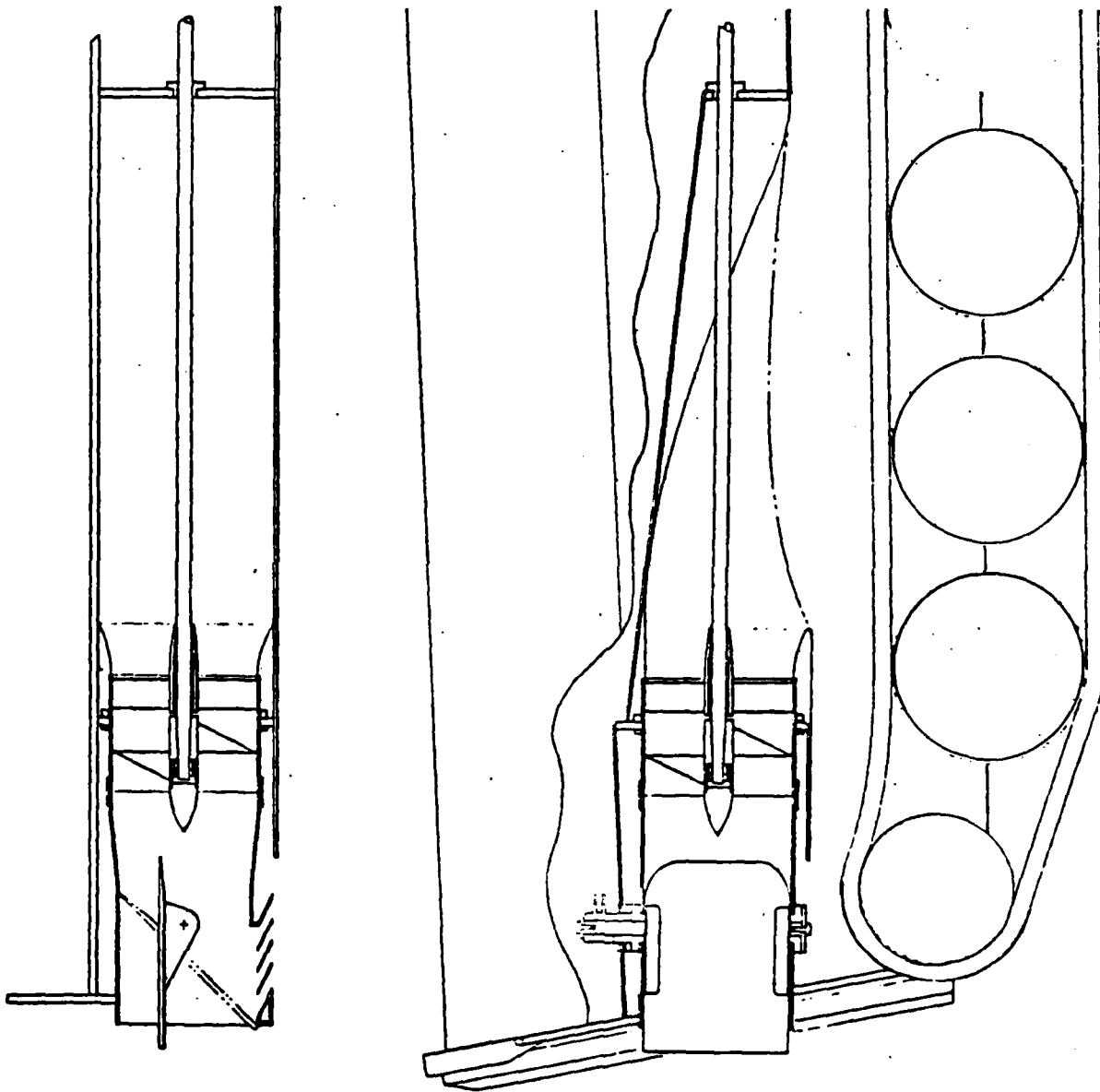


FIGURE 1 PROPOSED WATERJET PROPULSION SYSTEM FOR LVTP-7A1

DESIGN OF WATERJET PROPULSION SYSTEM
FOR LVTP-7A1 AMPHIBIOUS VEHICLE

The general objective was to design an axial flow pump, two of which would generate sufficient thrust to propel an existing amphibious vehicle - an LVTP-7A1 - at a cruise speed of 8 mph with a propulsive efficiency higher than that of the existing propulsor.

Figure 1 shows the configuration of the proposed propulsion system in the aft end of an LVTP-7A1. A 20-inch diameter impeller is to be housed in a horizontal cylindrical duct, and water would be drawn from the track well through a horizontal rectangular inlet. There is a transition from the 8 sq ft horizontal inlet area to a 22-inch by 24-inch vertical opening, and thence to the 20-inch nominal diameter of the cylindrical duct. A rudder and a reversing elbow are located at the duct outlet.

For purposes of calculating system performance, the impeller is assumed to be a marine screw propeller with wide tips, having a blade area ratio that is commercially available. Reference 1 presents open water characteristics of such a screw propeller in an axial cylinder. Figure 2 shows charts adapted from Reference 1 (Figures 28 and 29, respectively).

A matrix of design calculations involving the primary variables of pump input power, pump flow rate, and vehicle speed was completed to determine:

- (a) The pump head rise which can be produced by a given pump input horsepower for a range of flow rates.
- (b) The pump head rise at which cavitation begins to affect pump performance at given vehicle speeds for a range of flow rates.
- (c) The pump head rise required to produce a given flow rate through the duct system over a range of vehicle speeds.

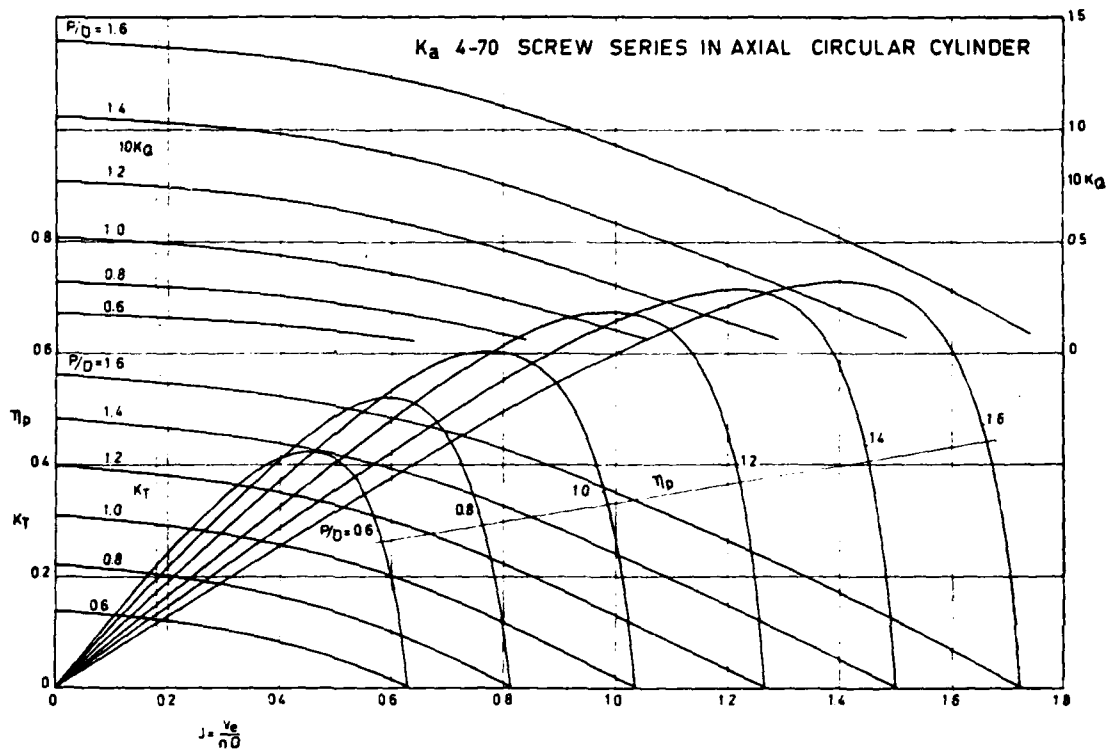


Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

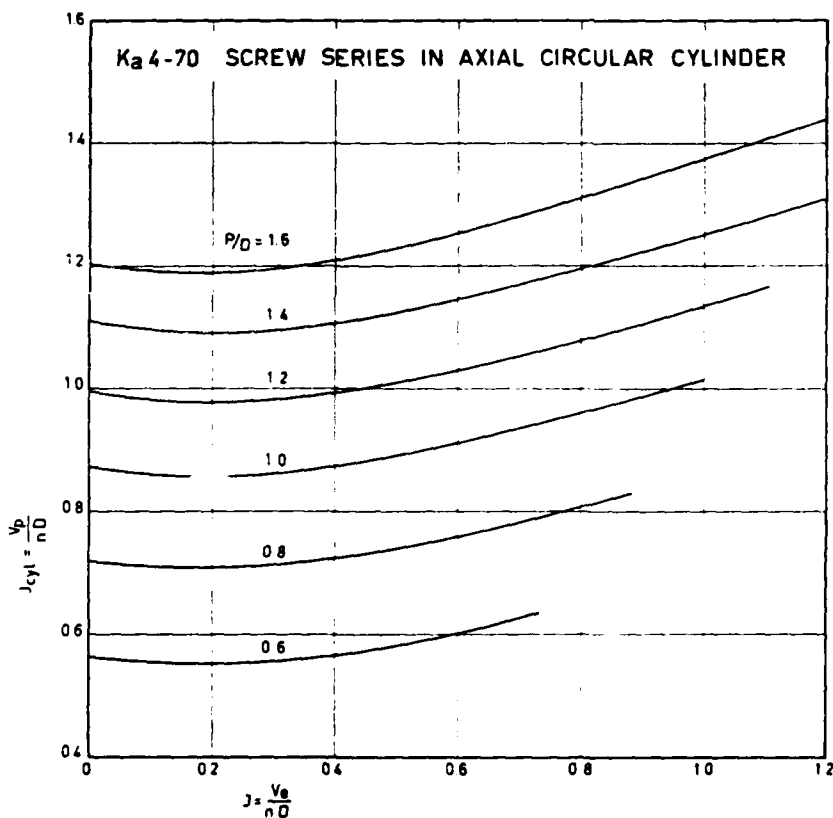


Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

FIGURE 2 "SCREW-IN-CYLINDER" DIAGRAMS (REFERENCE 1)

Appendix A presents the details of these calculations.

Figure 3 is a chart of pump head H_p versus flow rate Q in the form in two families of curves showing the results of calculations (b) and (c) above, with vehicle speed as a parameter. Equilibrium flow rate and pumphead rise were determined for a given vehicle speed at the intersection of the Required H_p and Available H_p curves for that speed.

Figure 4 is a chart of H_p versus Q showing results of calculations (a) and (c) above, with input power SHP and vehicle speed V_o , respectively, as parameters of the two families of curves. Entering Figure 4 with the equilibrium flow rate for a given speed from Figure 3, permits the determination of input power SHP required at equilibrium.

From the equilibrium flow rate Q cu ft/sec and the exit duct area, the exit jet velocity V_j was calculated. Jet thrust T , the time rate of change of fluid momentum, was then determined:

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

The ratio of output power, $TV_o/550$, to input SHP was then the propulsive coefficient, P.C.

Having determined hydrodynamic loads, required input power, and propeller operating conditions, a structural analysis of propeller, shaft, rudder and duct was performed to determine required sizes. Finally, weight estimates were made assuming (a) aluminum construction and (b) composite materials construction of the waterjet system.

From known characteristics of the existing waterjet system in the LVTP-7A1 amphibious vehicle, the following comparison was developed:

	Existing System	Proposed System
At 8 mph: Thrust, lb	2369	2846
Flow, gpm	14020	33346
P. C.	.25	.30
At 0 mph: Thrust, lb	3025	4278
Dry Weight, lb	435	284 (aluminum) 197 (composites)

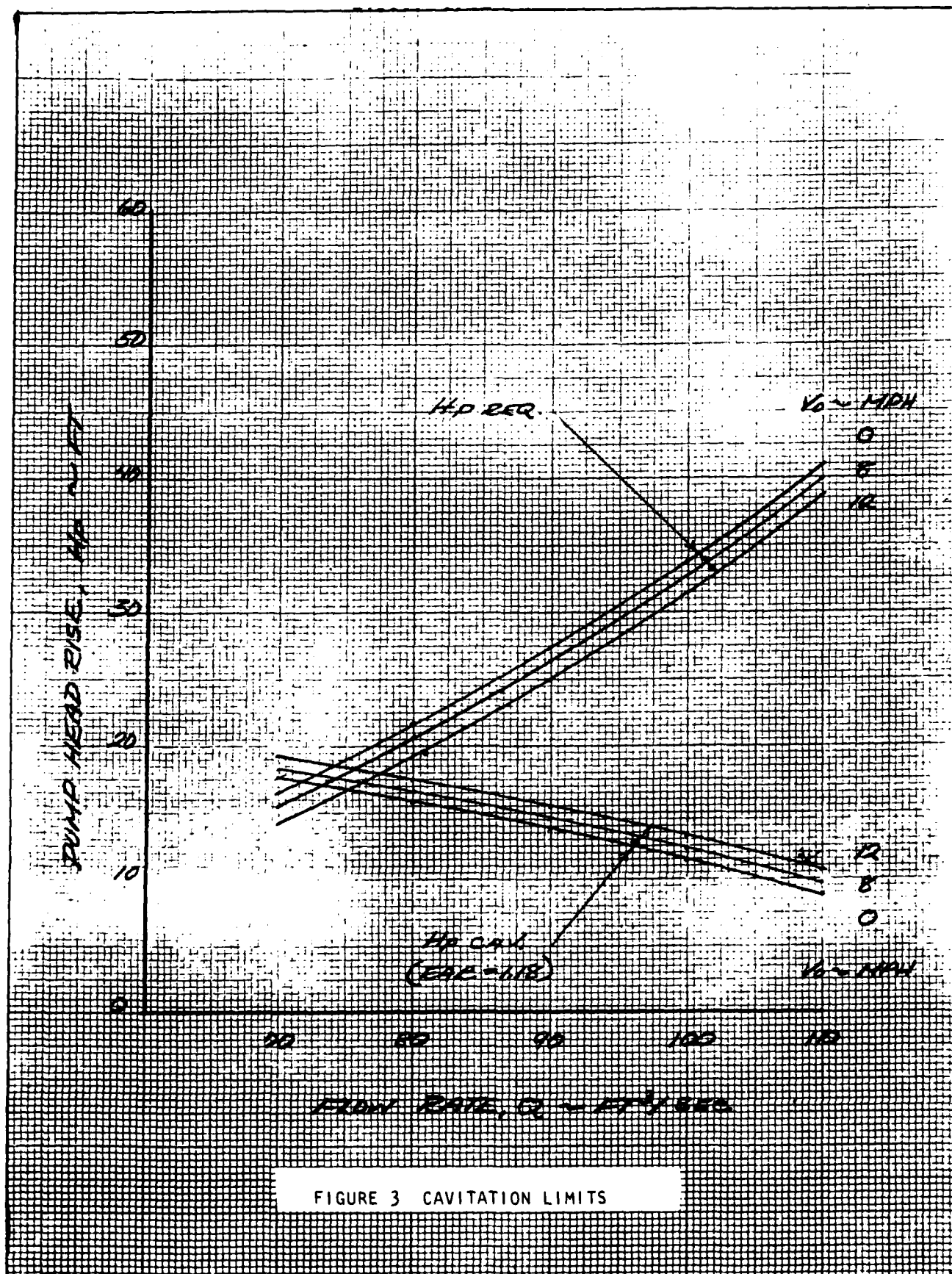
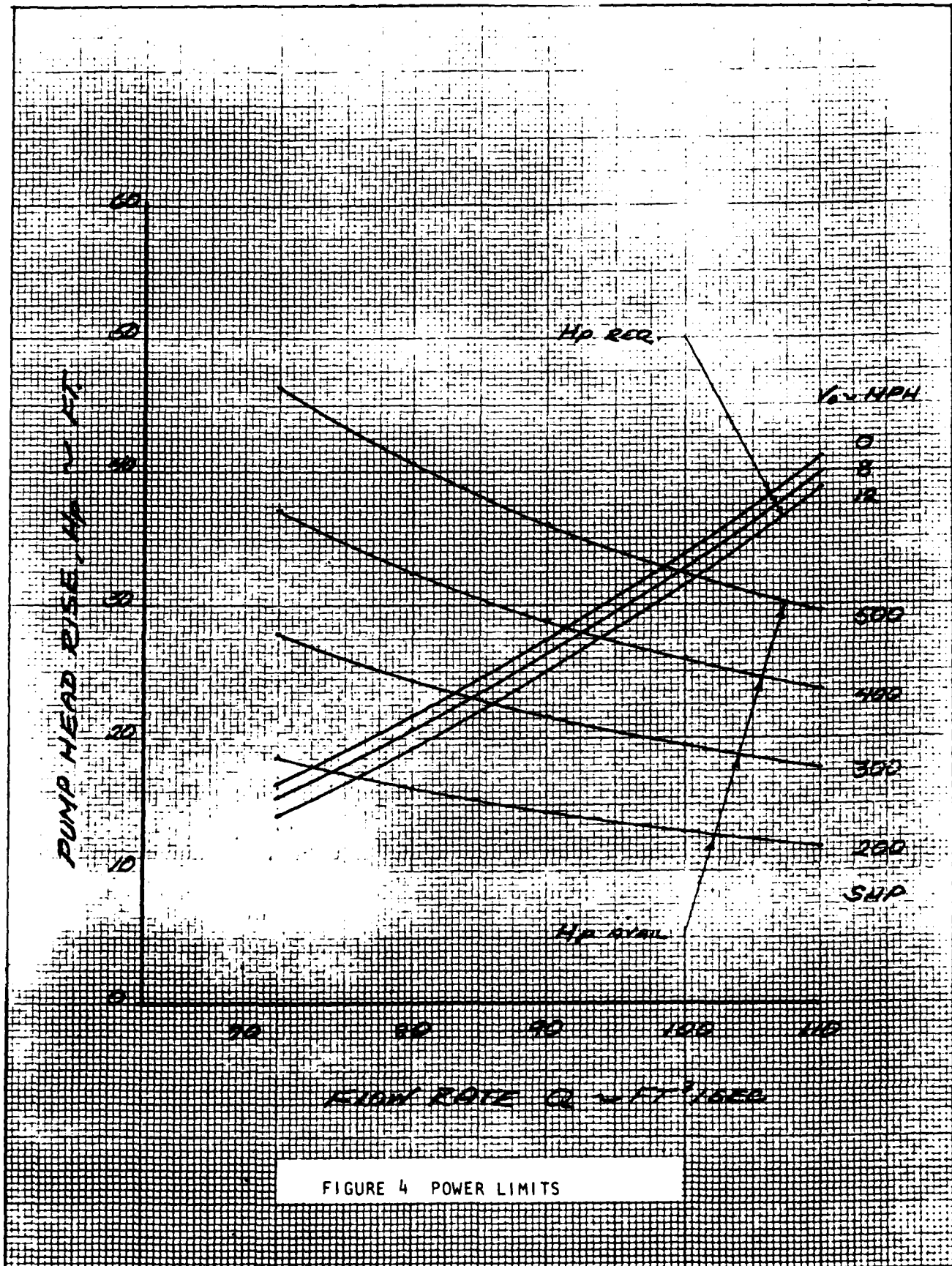


FIGURE 3 CAVITATION LIMITS



DESIGN OF PROPULSION PUMP SYSTEM
FOR HIGH-SPEED AMPHIBIAN

The general objective was to design an axial flow pump suitable for use in a multiple unit waterjet propulsion system in a high-speed amphibious vehicle to achieve a 20 mph speed.

Figure 5 shows an elevation sketch of such a unit which would draw water through a 42 inch x 20 inch rectangular port in the flat bottom of the amphibian. The flow then passes through a 24 inch x 20 inch inlet to a short transition and finally through a cylindrical duct with a nominal diameter of 20 inches in which the 20 inch diameter pump impeller is located.

For purposes of calculating system performance, the pump impeller was assumed to be a marine screw propeller with wide tips and a largest commercially-available blade area ratio. Appendix B presents details of calculations of:

- (a) Pump head rise versus flow rate for selected input powers, i.e., power-limited head rise.
- (b) Pump head rise versus flow rate for selected vehicle speeds, such that pump performance is not affected by cavitation, i.e., cavitation-limited head rise.
- (c) Pump head rise versus flow rate for selected vehicle speeds required to overcome system head losses.

These calculations made use of propeller performance data in cylindrical ducts, Figure 2 (from Reference 1), and a curve of inlet ram pressure recovery ratio in Appendix B, page B-22.

Equilibrium flow rate and pump head rise were determined, for a given vehicle speed, at the intersection of the curve of cavitation-limited head rise with the corresponding curve of head rise required to overcome system head losses.

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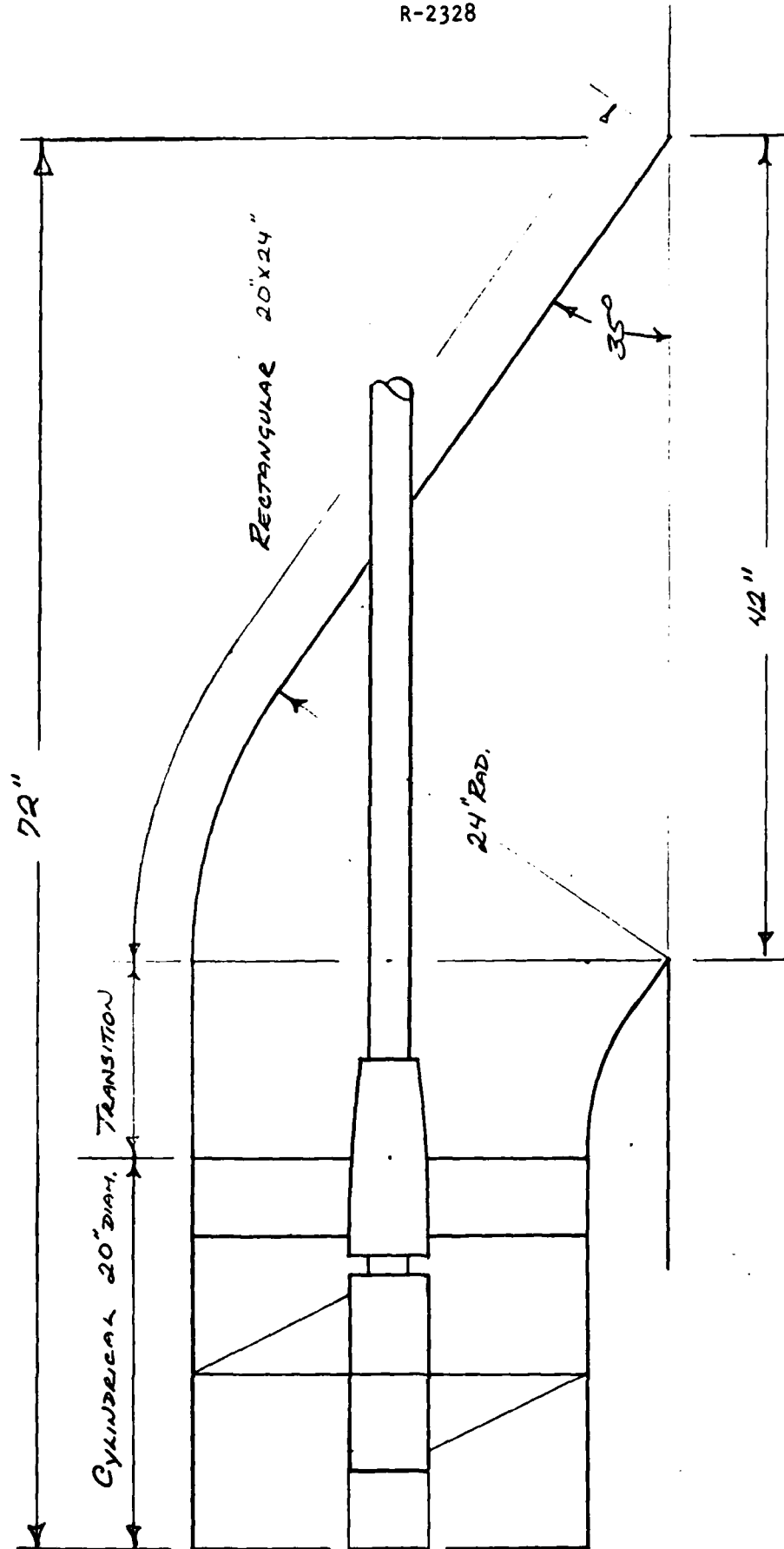


FIGURE 5 SKETCH OF 20 INCH DIAMETER PUMP

Knowing the equilibrium head rise and flow rate Q at a given vehicle speed, calculation procedure (a) was used to determine input power SHP. Jet thrust T was calculated after finding jet velocity from Q and the exit duct area. Finally, at a given vehicle speed, the ratio of thrust horsepower output to input SHP gave propulsive coefficient P.C.

Having determined hydrodynamic loads, required input power and propeller operating conditions for cavitation-free performance with an area ratio of 1.0, structural analyses of propeller, shafting and casing were performed to determine required sizes. Then weight estimates were made assuming (a) an aluminum casing, and (b) a composite-materials casing; aluminum alloy shafting and Ni-Al bronze propeller were used in each case. Selected performance characteristics were:

At zero mph:	Thrust	4507 lb
At 20 mph	Thrust	2365 lb
	Flow	40,080 gpm
	Input SHP	284 hp
	P.C.	.445

Composite construction of the casing reduced the dry weight of the waterjet system to 169 lb from a 239 lb weight for aluminum construction.

DESIGN OF 15-INCH DIAMETER PROPULSION PUMP SYSTEM FOR HIGH-SPEED AMPHIBIAN

The general objective was to design an axial flow pump suitable for installation in a multiple unit waterjet propulsion system in a high-speed amphibious vehicle to achieve a 20 mph speed.

Figure 6 is an elevation sketch of the proposed unit which would draw water through a $31\frac{1}{2}$ inch x 15 inch rectangular port in the flat bottom of the amphibian. The flow then passes through an 18 inch x 15 inch inlet to a short transition and finally through a cylindrical duct with a nominal diameter of 15 inches in which a 15 inch diameter pump impeller is located.

This 15 inch diameter impeller is to provide at least the same propulsive thrust as the 20 inch diameter propeller described in the previous section because the same vehicle is involved. To meet this loading requirement requires a significant increase in impeller blade area ratio if cavitation is to be avoided. Thus, the calculations in Appendix C include consideration of projected area ratios, PAR = 1.0, 1.5, 2.0, 2.5 and 3.0. By contrast the largest commercially available PAR is about 1.0.

Assuming a projected area ratio of 3.0 as an upper limit for extended cavitation-free operation, structural analyses of propeller, shaft and ducting were performed to determine required sizes. Weight estimates were made assuming (a) aluminum casing, and (b) a composite materials casing; Acquamet 22 shafting and Ni-Al bronze impeller were used in each case. Selected performance characteristics were:

At zero mph:	Thrust, lb	5,403
At 20 mph:	Thrust, lb	3,703
	Flow, gpm	30,120
	Input SHP	462
	P.C.	.428

Composite construction of the casing reduced the dry weight of the waterjet system to 134 lb from a 167 lb weight for aluminum construction.

The next step in design would require consideration of available engine powers, and the thrust needed to overcome vehicle drag in order to settle on a practical area ratio for the impeller.

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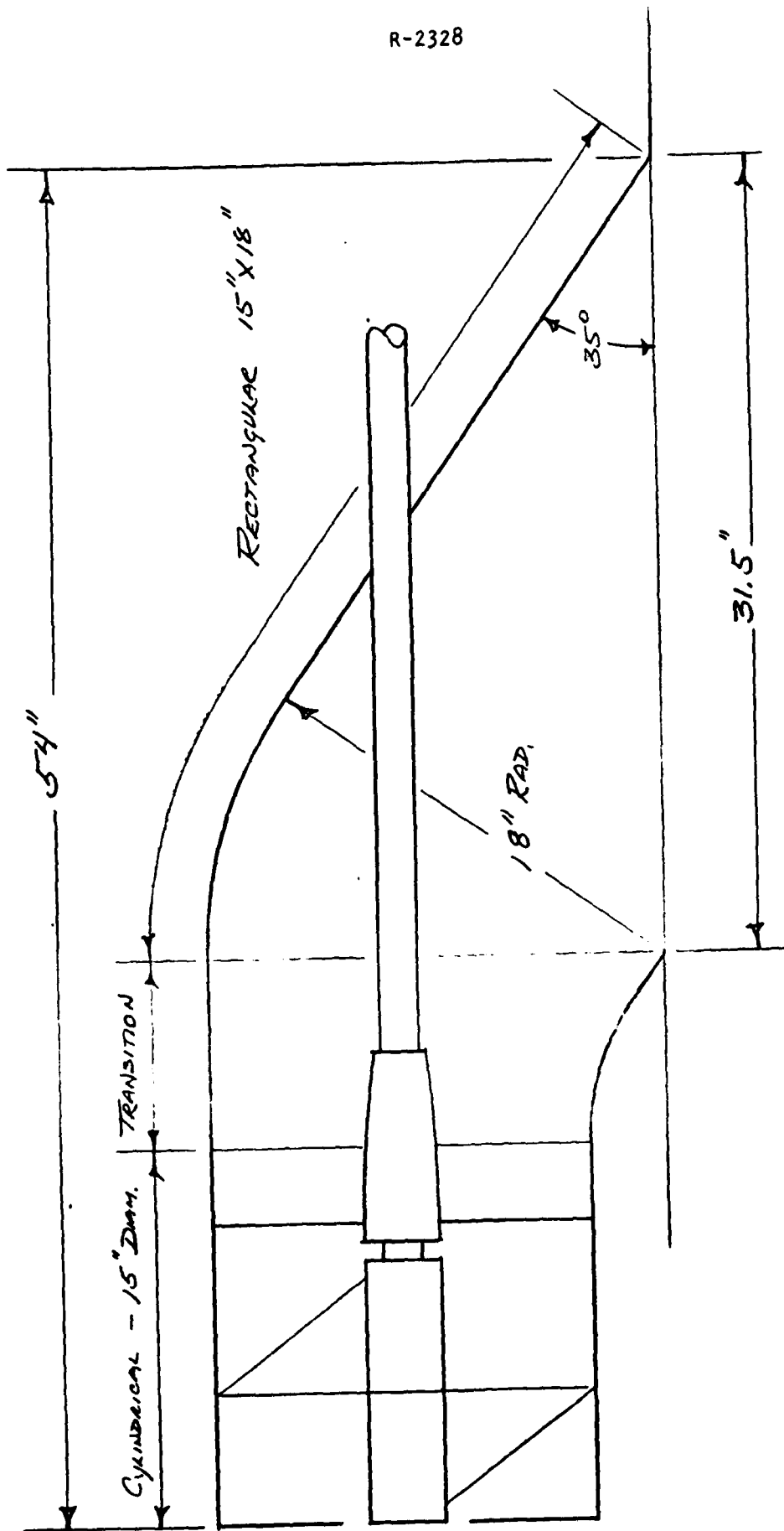


FIGURE 6 SKETCH OF 15 INCH DIAMETER PUMP

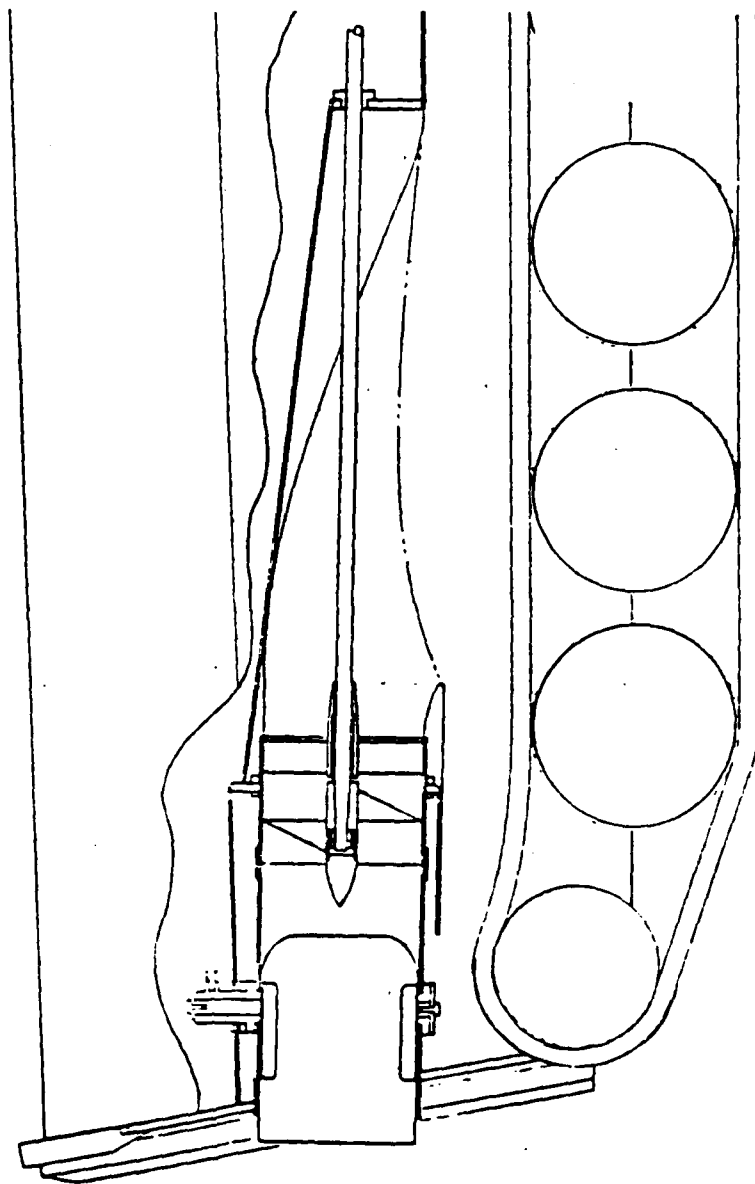
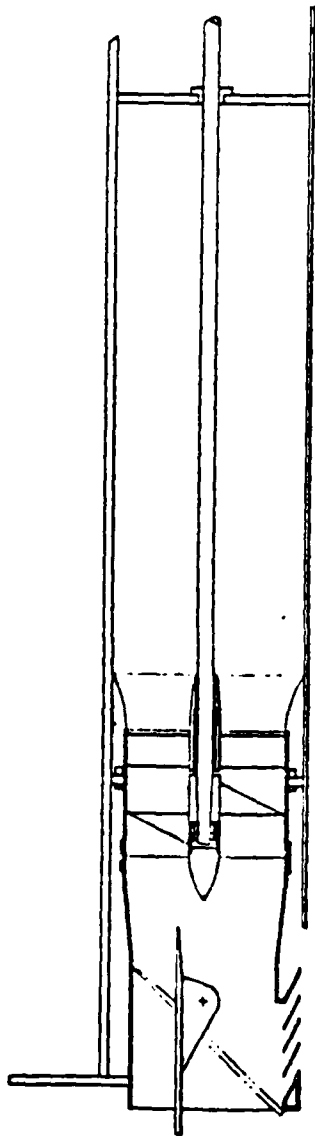
REFERENCES

1. van Manen, J.D. and Oosterveld, M.W.C., "Analysis of Ducted-Propeller Design," Trans. SNAME, Vol. 74, 1966.
2. "Flow of Fluids Through Valves, Fittings and Pipe," Crane Company Technical Paper 410.
3. Conolly, J.E., "Strength of Propellers," Trans. Royal Institution of Naval Architects, 1960.

APPENDIX A

OBJECTIVES:

- Design a replacement propulsion system for an LVTP-7A1 amphibious vehicle.
- Determine performance, weight and dimensional characteristics of propulsion system.
- Use simple "propeller-in-tube" approach
- Limit blade area ratio to that available in existing propeller series.
- Investigate use of composite materials.



PROPOSED WATERJET PROPULSION SYSTEM FOR LVTP-7A1

PERFORMANCE CHARACTERISTICS

- POWER LIMITS
- CAVITATION LIMITS
- SYSTEM PERFORMANCE
- REVERSE OPERATION
- DEMANDS ON SYSTEM

CALCULATION NOTE:

$$D = \text{PROP. DIAM.} = 20' = 240''$$

$$A_p = \text{PROP. AREA} = .785 [(600')^2 - (330')^2] = 2.0942 \text{ FT}^2$$

$$A_s = \text{INLET AREA} = \frac{\pi (144)}{4} = 3.667 \text{ FT}^2$$

$$J_{cyl} = \text{ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE}$$

$$J_e = \text{ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE}$$

$$\frac{J_{cyl}}{J_e} = \frac{A_s}{A_p} = \frac{3.667}{2.0942} = 1.7510$$

$$P/D = \text{PROP. PITCH DIAM. RATIO}$$

$$J_{cyl} = f(P/D, \frac{J_{cyl}}{J_e}) = .895 \quad (\text{VON KARMAN - FIG. 1})$$

$$J_e = J_{cyl} / (\frac{J_{cyl}}{J_e}) = .895 / 1.7510 = .511$$

$$\lambda_o = \text{PROPULSION EFFICIENCY OF D-D-TUBE COMB.} = f(J_e, P/D) \quad (\text{FIG. 2})$$

$$\lambda_{re} = \text{PROP. RELATIVE ROTATIVE EFFICIENCY} \quad (\text{GODDARD - FIG. 3})$$

$$\lambda_p = \text{PUMP EFFICIENCY} = \frac{J_{cyl}}{J_e}$$

$$\text{SHP} = \text{PUMP INPUT POWER}$$

$$Q = \text{FLOW RATE}$$

$$H_{pump} = \text{PUMP HEAD}$$

$$\frac{550 \text{ SHP}}{P \& Q}$$

CALCULATION

A-5

CAVITATION LIMIT

CALCULATION NOTES

$$D = \text{PROP. DIAM.} = 20" = 1.667'$$

$$A_p = \text{PROP. DISC AREA} = \pi [(1.667)^2 - (.533)^2] = 2.0942 \text{ ft}^2$$

$$A_i = \text{INLET AREA} = \pi \left(\frac{10}{12}\right)^2 = 3.667 \text{ ft}^2$$

$$V_{cyl} = \text{ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE}$$

$$V_e = \text{ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE}$$

$$\frac{V_{cyl}}{V_e} = \frac{A_i}{A_p} = \frac{3.667}{2.0942} = 1.7510$$

$$P/D = \text{PROP. DITCH DIAM. RATIO}$$

$$V_{cyl} = f(P/D, \frac{V_{cyl}}{V_e}) \quad (\text{VON KARMAN - FIG. 1})$$

$$Q = \text{FLOW RATE} \sim \text{FT}^3/\text{SEC}$$

$$n = \text{PROP. SPEED} = \frac{Q}{A_p V_{cyl} D}$$

$$H_{L1} = \text{INLET HEAD LOSS} = .000533 Q^2 \quad (\text{DERIVATION "1"})$$

$$V_i = \text{INLET VELOCITY} = Q/A_i$$

$$V_{tmax} = \text{TANGENTIAL VELOCITY AT DISC PERIPHERY} = 1.5 V_i$$

$$V_{tr} = \text{TOTAL VELOCITY AT DISC PERIPHERY} = \sqrt{V_i^2 + V_{tmax}^2}$$

$$H_{ATM} = \text{HEAD DUE TO ATMOSPHERIC PRESSURE}$$

$$H_2 = \text{HEAD DUE TO EXHAUSTION}$$

$$H_v = \text{HEAD DUE TO VAPOR PRESSURE}$$

Calculation Note

Calculation Note (Cont.)

V_0 = FREE STREAM V. (ft/sec) = 1000

RPR = RAM PRESS. RECOVERY RATIO = .50 (GUESS)

H_0 = RAM PRESS. RECOVERY = $(RPR) \frac{V_0^2}{2g}$

H_{I_s} = INLET STATIC HEAD (ABOVE VAP. PRESS.) = $H_{atm} + H_z - H_v + H_0 - H_{L_s} - \frac{V_s^2}{2g}$

P_{I_s} = INLET STATIC PRESS. (ABOVE VAP. PRESS.) = $\rho g H_{I_s}$

σ_{ir} = LOCAL CAV. NO. AT I.D. = $\frac{P_{I_s}}{\frac{1}{2} \rho V_{ir}^2}$

τ_{cav} = PROP. LOAD COEF. AT CAV. LIMIT = .70_{ir} (GAWN)

EAR = PROP. EXPANDED AREA RATIO = 1.1% (MAN. VAL.)

PAR = PROP. PROJECTED AREA RATIO = 1.01 (DEFINITION # 3)

$H_{p_{cav}}$ = PUMP HEAD RISE AT CAV. LIMIT = $\frac{(\tau_{cav})(PAR)(V_{ir})^2(\rho)}{\rho g}$

CAPITAL LIMIT

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CALCULATION NOTES

- REQUIRED PUMP HEAD RISE

$$V_0 = \text{CRAFT SPEED}$$

$$Q = \text{FLOW RATE} \sim \text{FT}^3/\text{SEC}$$

$$H_{PRQ} = .00339 Q^2 - .0078 V_0^2$$

(DERIVATION # 2)

- ESTIMATED THRUST (POWER LIMIT)

$$V_0 = \text{CRAFT SPEED}$$

$$\text{SHP} = \text{PUMP INPUT POWER}$$

$$Q_{EQ} = \text{EQUILIBRIUM FLOW RATE @ } H_{PAVAIL} = H_{PRQ}$$

$$A_s = \text{EXIT AREA} = 2.404 \text{ FT}^2$$

$$V_s = \text{EXIT VELOCITY} = Q_{EQ} / A_s$$

$$T = \text{THRUST} = \rho Q (V_s - V_0)$$

$$\text{P.C.} = \text{PROPULSIVE COEF.} = \frac{T V_0}{550 \text{ SHP}}$$

- ESTIMATED THRUST (CAVITATION LIMIT)

$$V_0 = \text{CRAFT SPEED}$$

$$Q_{EQ} = \text{EQUILIBRIUM FLOW RATE @ } H_{PRQ} = H_{PAVAIL}$$

$$A_s = \text{EXIT AREA} = 2.404 \text{ FT}^2$$

$$V_s = \text{EXIT VELOCITY} = Q_{EQ} / A_s$$

$$T = \text{THRUST} = \rho Q (V_s - V_0)$$

$$H_{PRQ} = \text{EQUILIBRIUM HEAD RISE @ } H_{PAVAIL} = H_{PRQ}$$

$$\eta_p = \text{PUMP EFFICIENCY} = .76$$

(POWER LIMIT CALC.)

$$\text{SHP} = \text{PUMP INPUT POWER} = \frac{\rho g H_{PRQ} Q_{EQ}}{550 \eta_p}$$

$$\text{P.C.} = \text{PROPULSIVE COEF.} = \frac{T V_0}{550 \text{ SHP}}$$

1. PROBLEM

ASSUMPTIONS

- REQUIRED PUMP HEAD

<u>V₀</u>	<u>Q</u>	<u>H_{req}</u>
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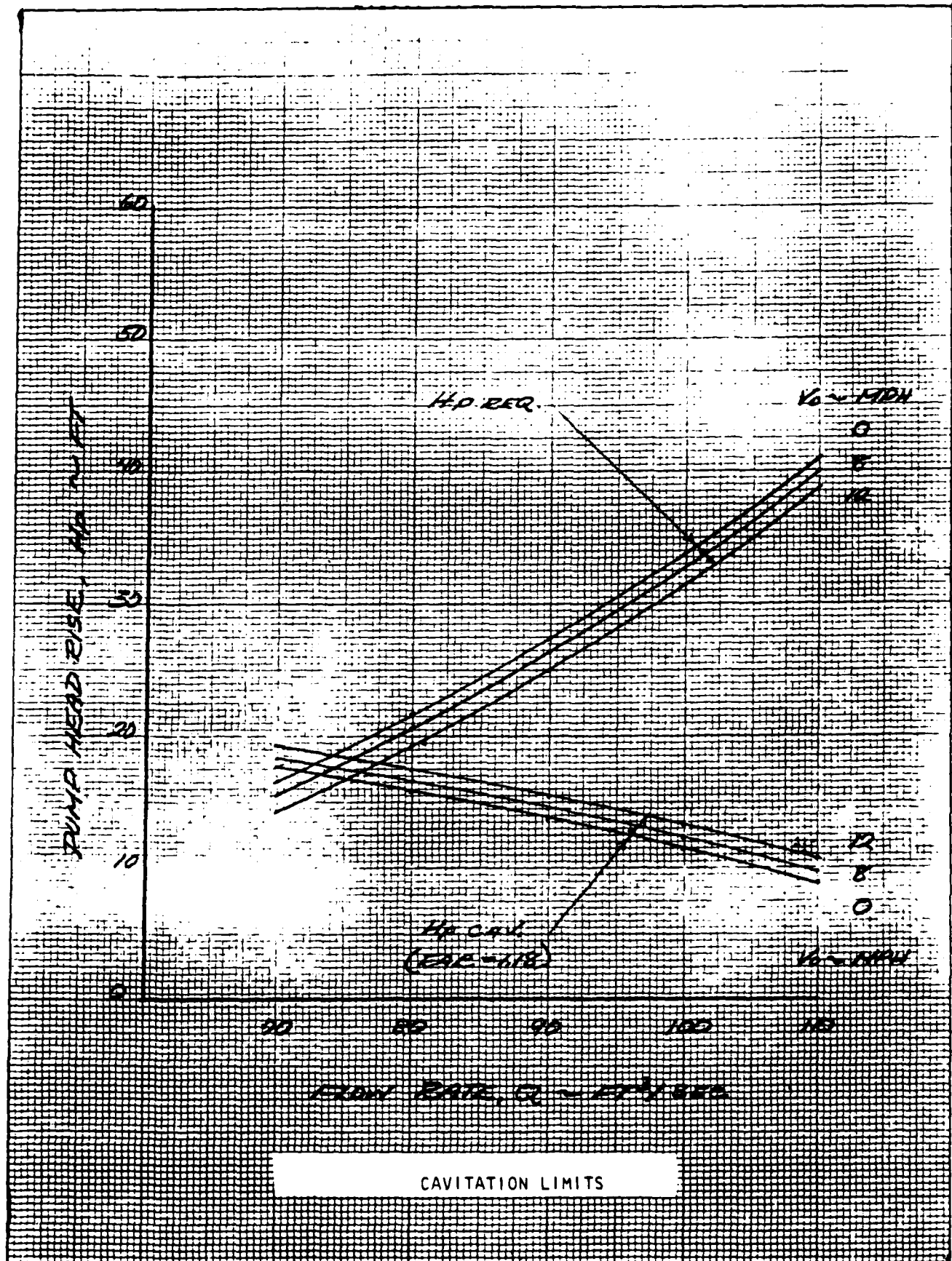
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	80	21.70
	90	27.46
	100	33.90
	110	41.02

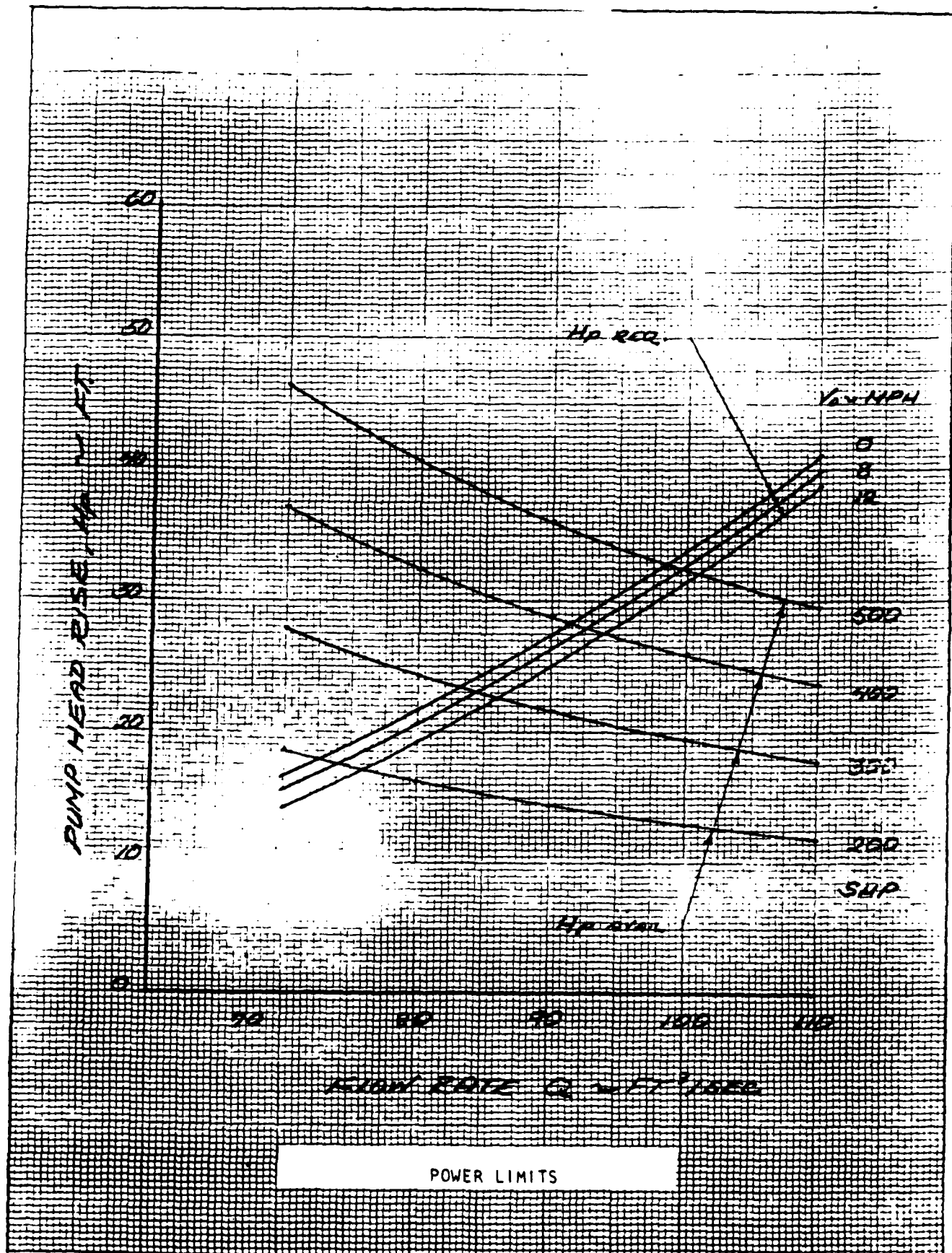
5.48	70	16.34
	80	21.43
	90	27.19
	100	33.63
	110	40.75

} NOT PLOTTED (CLARITY)

11.76	70	15.53
	80	20.62
	90	26.38
	100	32.82
	110	39.94

17.64	70	14.18
	80	19.27
	90	25.03
	100	31.47
	110	38.59





CALCULATIONS (CONT.)

• ESTIMATED THRUST (POWER LIMIT)

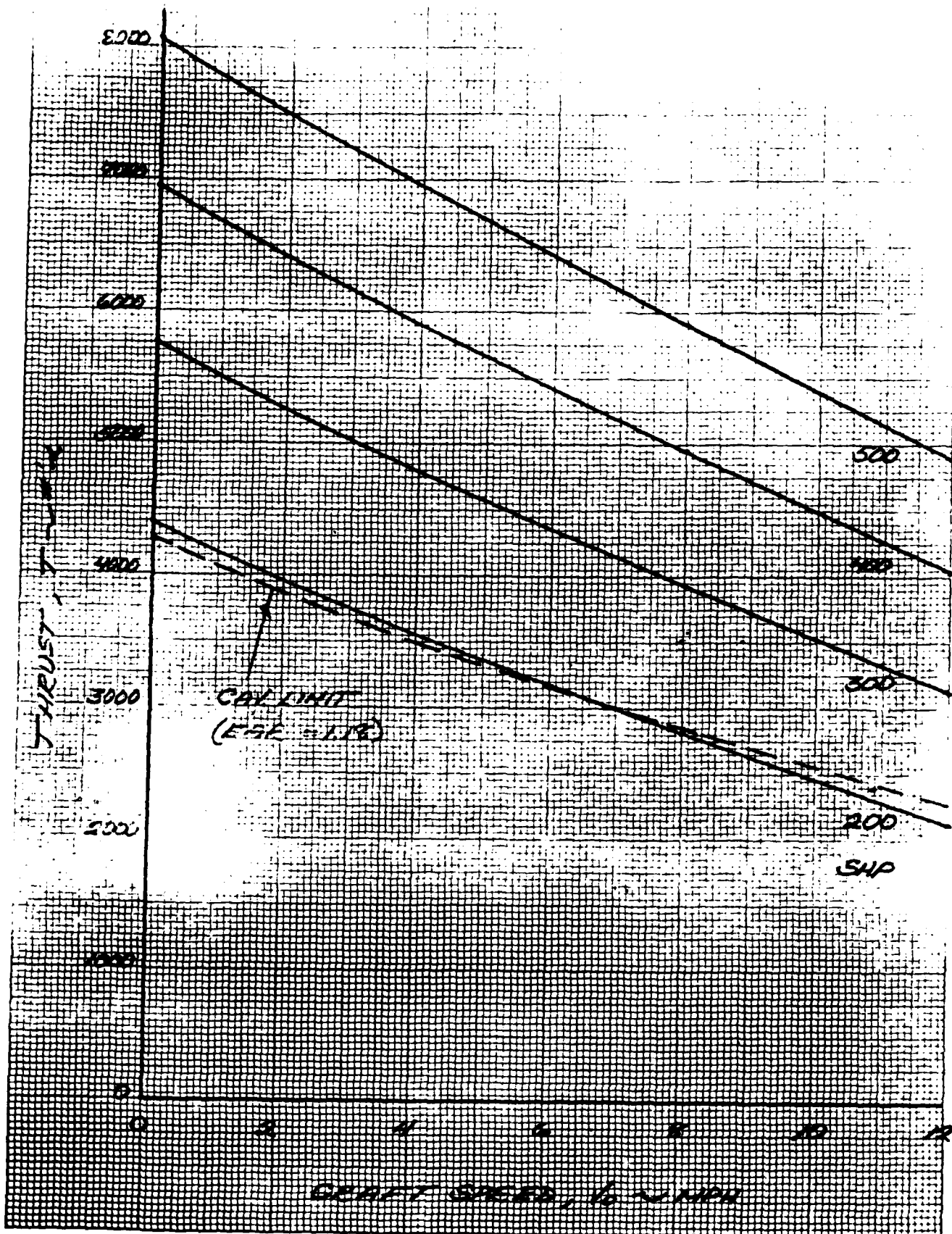
<u>V₀</u>	<u>SHP</u>	<u>Q_{REQ}</u>	<u>A_J</u>	<u>V_J</u>	<u>T</u>	<u>P.C.</u>
0	500	98.5	2.404	40.97	9071	0
	400	91.4		38.02	6950	0
	300	83.2		34.61	5759	0
	200	72.7		30.24	4397	0
11.76	500	99.6		41.43	5910	.2527
8.4 MPH	400	92.6		38.52	4956	.2649
	300	84.4		35.11	3941	.2409
	200	74.0		30.78	2815	.3009
17.64	500	100.8		41.93	4897	.3141
12.4 MPH	400	94.0		39.10	4034	.3235
	300	86.0		35.77	3118	.3333
	200	76.0		31.61	2123	.3405

CALCULATIONS (CONT.)

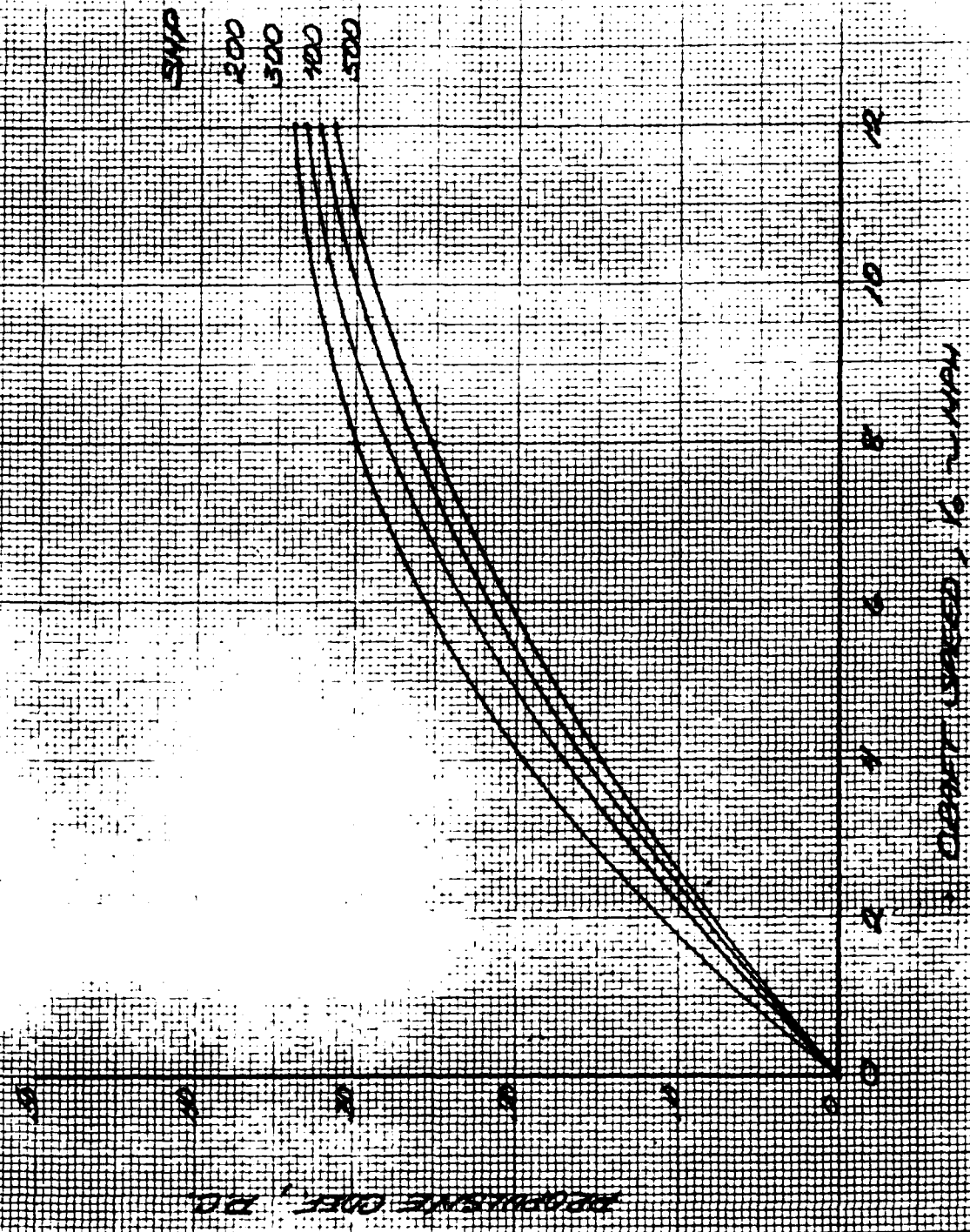
• ESTIMATED THROUST (CAVITATION LIMIT)

<u>V₀</u>	<u>Q_{req}</u>	<u>A₀</u>	<u>V₀</u>	<u>T</u>	<u>H_{req}</u>	<u>λ_p</u>	<u>SHR</u>	<u>P.C.</u>
0	71.7	2,404	29.83	4278	17.4	.26	192	0
11.26	24.3		30.91	2846	17.6		201	.3027
12.64	22.6		32.28	2272	18.0		215	.3389

12 THRUST

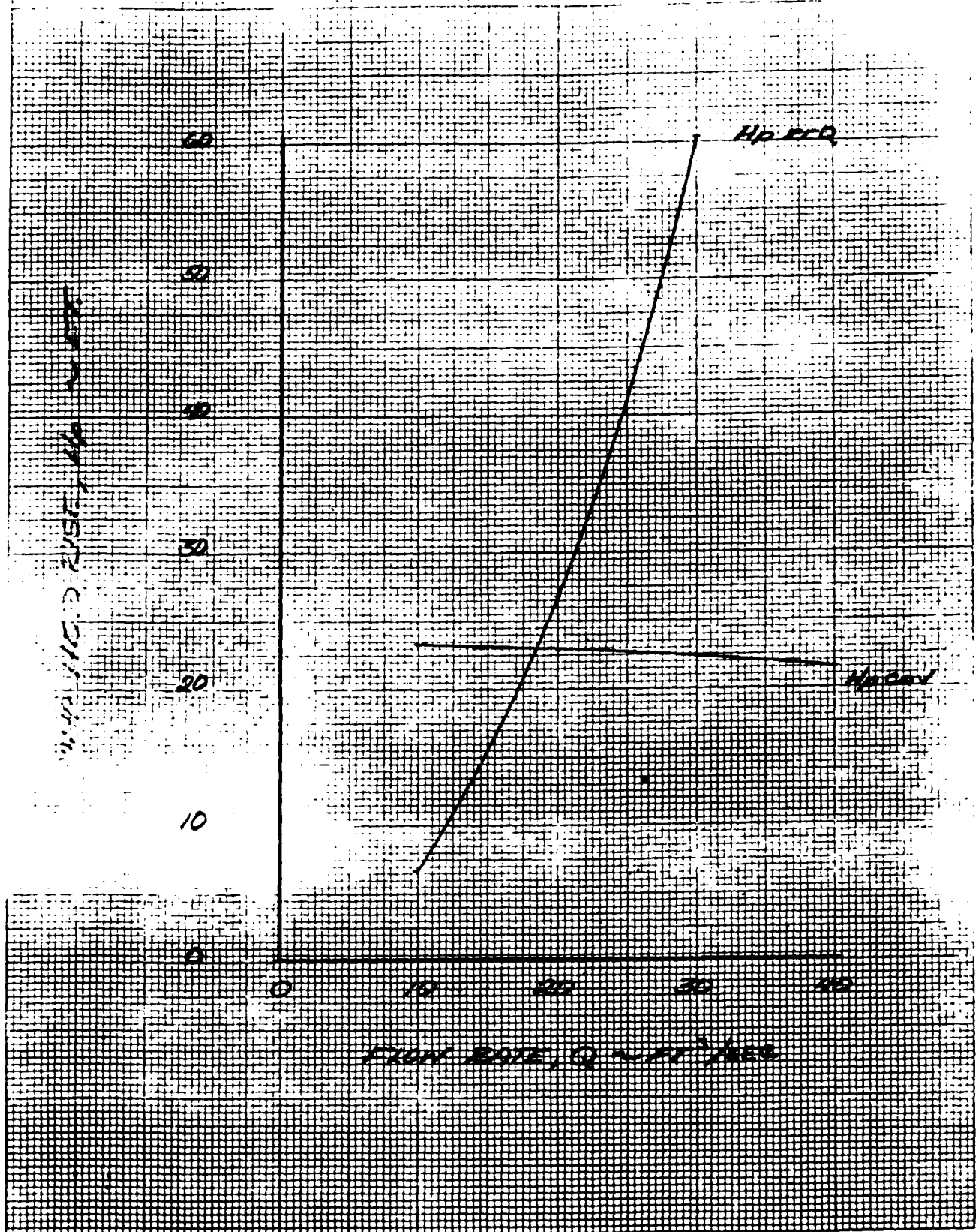


ESTIMATED PROPORTIONATE COEFFICIENT



A-16

REVISION LIMIT



REQUIRED PUMP HEAD: H_P

$$H_{PRQ} = H_L + H_L - H_0$$

$$\left. \begin{aligned} H_L &= .0666 Q^2 \\ H_L &= .000533 Q^2 \\ H_0 &= 0 \end{aligned} \right\} \text{DERIVATION \#1}$$

$$H_{PRQ} = .0666 Q^2 + .000533 Q^2 - 0 = .0671 Q^2$$

Q H_{PRQ}

10	6.71
20	26.84
30	60.39
40	107.36

CAVITATION LIMIT (SEE CAV. LIMIT CALCS.)

Q H_{PRQ}

10	6.71
20	26.84
30	60.39
40	107.36

DESIGN DATA

<u>Q_{req}</u>	<u>A_R</u>	<u>V_R</u>	<u>T_R</u>	<u>H_{PRQ}</u>	<u>γ_p</u>	<u>SH_P</u>
18.5	.764	24.2	896	23.1	.76	66

Deviation

Fig. 1 & 2

1513 LAMEREN FIG. 1 } MOIST. TESTS OF PROP. IN AXIAL CYLINDER
" FIG. 2 }

DEVIATIONS

- #1 ESTIMATED INLET, CRANK & RELEASE SYSTEM LOSSES
- #2 REQUIRED PUMP HEAD ETC
- #3 PROPELLER BLADE AREA

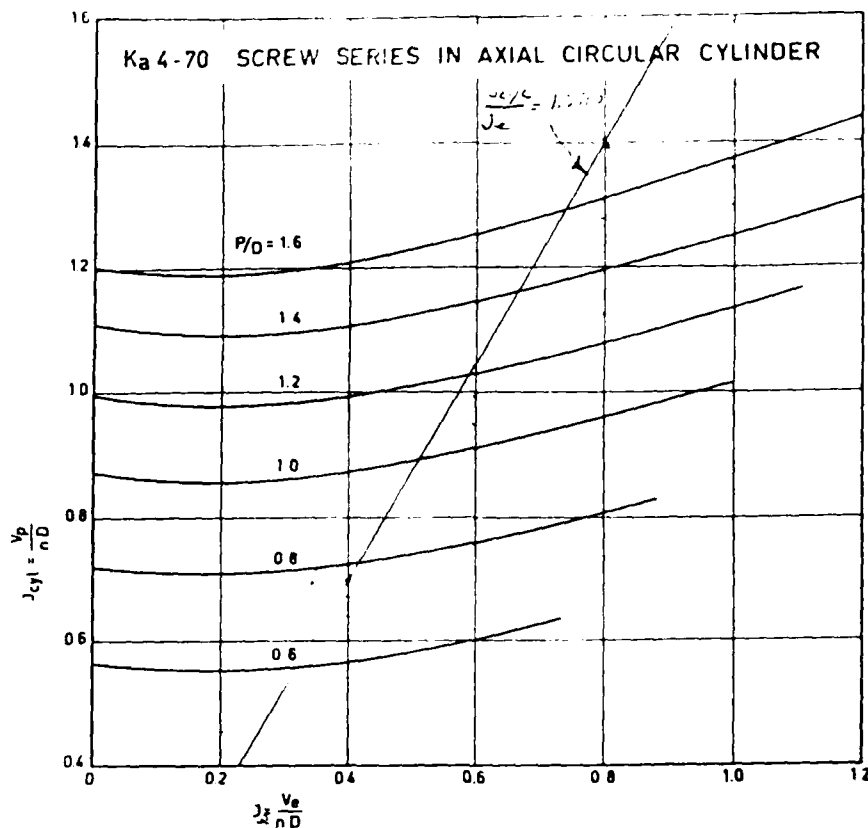


Fig. 1

Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

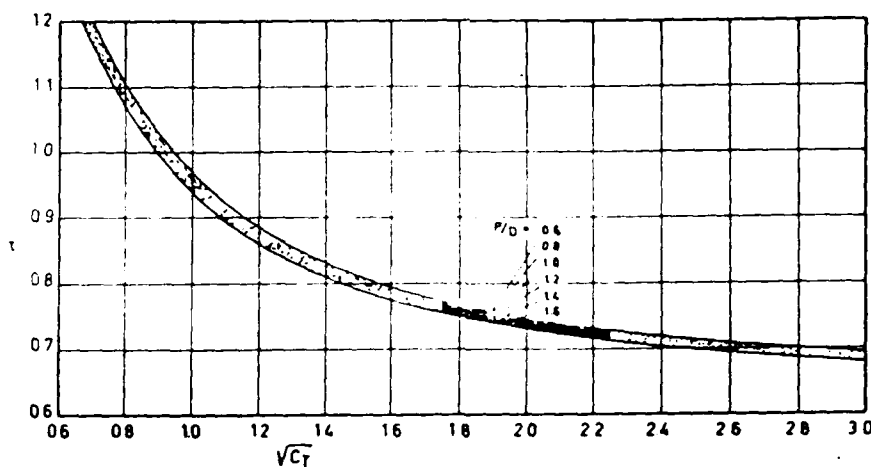


Fig. 30 Relation between thrust coefficient C_T and thrust ratio τ of nozzle no. 19a

figure have been obtained by substituting nozzles with different length-diameter ratios by systems of annular vortexes and calculating the induced velocities in the screw disk.

If the radial displacement of the streamlines is small, we can consider the streamlines as lying approximately on cylindrical planes. If internal friction and turbulence are neglected, the radial

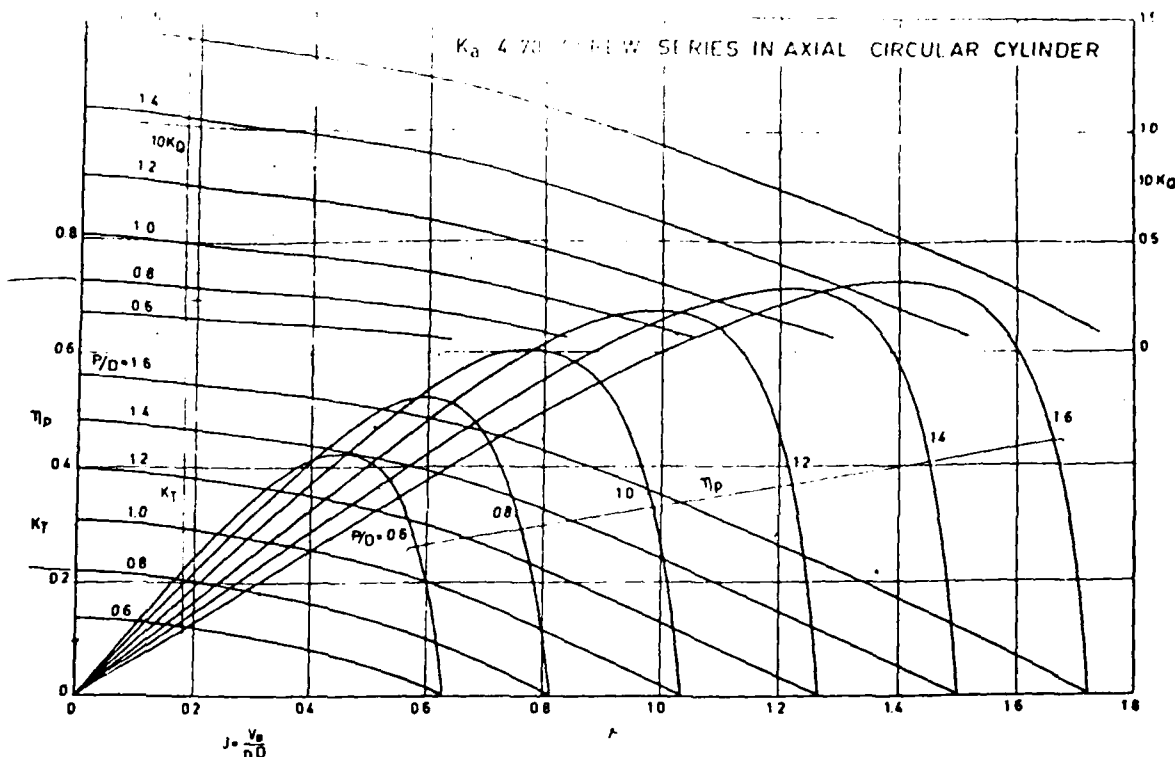


Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

been obtained from the experiments with the Ka 4-70 screw series in an axial circular cylinder and from the application of the momentum theorem.

From the comparison of the axial velocities obtained with these methods, we see that

1 The velocities agree reasonably well at high loadings of the ducted propeller system ($C_T > 1$).

2 The difference between the axial velocities becomes very large at low loadings ($C_T < 1$).

In regard to the second conclusion, the following remark may be made. From Fig. 13 it can be seen that the nozzle drag due to friction becomes substantial at low loadings of the ducted-propeller system. Then, it is no longer permitted to neglect the effect of friction on the force action between nozzle and fluid.

The design of a screw in a nozzle may now be carried out as follows:

With given thrust T or power P , intake velocity V_0 , and number of revolutions n , the B_p and consequently the optimum diameter coefficient D can be determined with the aid of open-water test results of the nozzle considered, in combination with a systematic screw series (see, for instance, Fig. 24). In addition, the thrust coefficient C_T and the propeller thrust-total thrust ratio τ can be determined. With the aid of the experiments

of the systematic screw series in the axial circular cylinder or using the momentum theorem, the axial velocity V_p in the way of the screw can be found. In addition, the mean axial velocity in the vicinity of the screw due to the nozzle action, U_k , and due to the screw action U_p , can be calculated.

The pressure difference created by the screw becomes

$$\Delta p = \frac{T_p}{\frac{\pi}{4} (D^2 - d_a^2)}$$

In order to avoid an excessive loading of the inner radii of the screw blades, the usual assumption for axial pumps that the head is constant for all radii is abandoned. The following radial $\Delta p(r/R)$ distribution is suggested for the screws in nozzle no. 19a:

$$\Delta p(r/R) = [4.88 - 4r/R] \cdot [r/R - 0.133] \Delta p$$

The radial distribution of the axial and tangential velocities at the screw may be approximated as follows:

A reasonable radial distribution of the axial velocities due to the nozzle action can be determined from Fig. 32. The results given in this

INLET FRICTION

$$A_e = \text{ENTRANCE AREA} = (672)(1) = 8.00 \text{ FT}^2$$

$$Q = \text{NOMINAL FLOW RATE} = 75 \text{ FT}^3/\text{SEC}$$

$$V_e = \text{ENTRANCE VELOCITY} = Q/A_e = 75.00/8.00 = 9.375 \text{ FT/SEC}$$

$$K_e = \text{LOSS COEFF.} = 1.50$$

(CRANE - SQ. EDGED ORIFICE)

$$H_{L_e} = K_e \frac{V_e^2}{2g} = \frac{(1.50)(9.375)^2}{2(32.2)} = 2.047'$$

INLET FRICTION & BEND

$$A_i = \text{INLET AREA} = \frac{(22)(24)}{144} = 3.667 \text{ FT}^2$$

$$V_i = \text{INLET VELOCITY} = Q/A_i = 75/3.667 = 20.45 \text{ FT/SEC}$$

$$d_e = \text{EQUIV. INLET DIAM.} = \frac{4A_i}{2\left(\frac{22+24}{12}\right)} = \frac{4(3.667)}{2\left(\frac{22+24}{12}\right)} = 1.9132'$$

$$Re = \frac{V_i d_e}{\mu} = \frac{(20.45)(1.9132)}{1.24 \times 10^{-5}} = 3.16 \times 10^6$$

$$\epsilon/Re = \text{RELATIVE SURFACE ROUGHNESS} = .000005/1.9132 = .00000261$$

$$f = \text{FRICTION FACTOR} = .00915$$

$$L_i = \text{INLET LENGTH} = 5'$$

$$L_e = \text{EQUIV. LENGTH} = \left(\frac{L}{D}\right)_E \left(\frac{V_i}{V_e}\right) \left(\frac{\mu}{\mu_e}\right) = (36)(1.9132)\left(\frac{25}{90}\right) = 19.13'$$

$$L = \text{TOTAL EQUIV. LENGTH} = L_i + L_e = 5 + 19.13 = 22.13'$$

$$H_L = f \left(\frac{L}{d_e}\right) \left(\frac{V_i^2}{2g}\right) = (.00915) \left(\frac{22.13}{1.9132}\right) \left(\frac{20.45^2}{2(32.2)}\right) = .732'$$

SHAFT

$$C_D = \text{DRAG COEF. DUE TO CROSS FLOW} = 1.1 \frac{\text{ft}^2}{\text{ft}} \frac{V}{2} = 1.1 \frac{\text{ft}^2}{\text{ft}} \frac{20.45}{2} = .012$$

$$V_E = 20.45 \text{ FT/SEC}$$

$$L = \text{SHAFT LENGTH} = 4'$$

$$d = \text{SHAFT DIAM.} = 2'' = .1667'$$

$$D = \text{SHAFT DIA.} = C_D C_L V_E^2 L d = (.012)(.34)(20.45)^2(4)(.1667) = 3.12''$$

$$HL = \frac{D}{\rho g A_T} = \frac{3.12}{2(62.4)(3.14)} = .0132'$$

TRANSITION

$$A_T = \text{CROSS SECTION AREA} = \frac{5.567 + 2.0242}{2} = 2.8806 \text{ FT}^2$$

$$V_T = \text{VELOCITY} = Q/A_T = 75/2.8806 = 26.04 \text{ FT/SEC}$$

$$d_e = \text{EQUIV. DIAM.} = \sqrt{\frac{4A_T}{\pi}} = .19152'$$

$$Re = \frac{V_T d_e}{\nu} = \frac{(26.04)(.19152)}{1.24 \times 10^{-5}} = 4.02 \times 10^6$$

$$Q/d_e = \text{RELATIVE ROUGHNESS} = \frac{.00015}{.19152} = .0000261$$

$$f = \text{FRICTION FACTOR} = .0074$$

$$L_T = \text{TRANSITION LENGTH} = 1.58 \times 10^4'$$

$$HL = f \left(\frac{L_T}{d_e} \right) \frac{V_T^2}{2g} = .0074 \left(\frac{1.58 \times 10^4}{.19152} \right) \frac{(26.04)^2}{2(32.2)} = .0301'$$

BRASS TUBE

$$A_p = \text{CROSS SECTION AREA} = 2.0742 \text{ ft}^2$$

$$V_p = \text{VELOCITY} = Q/A_p = 7 / 2.0742 = 35.81 \text{ ft/sec}$$

$$L = \text{TUBE LENGTH} = 2'$$

$$Re = \frac{V_p L}{\mu} = \frac{(35.81)(2)}{1.24 \times 10^{-5}} = 5.78 \times 10^6$$

$$C_f = .00321$$

$$S = \text{TUBE WETTED SURF.} = (2)(3.3) \text{ ft} = 2.07 \text{ ft}$$

$$D = \text{TUBE DRAG} = (C_f + 0.0008)(S)(\frac{1}{2}) V_p^2 = (.00321 + .0008)(2.07)(\frac{1}{2})(35.81)^2 = 10.64$$

$$HL = \frac{D}{\rho g A_p} = \frac{10.64}{(2)(32.2)(2.0742)} = .0289'$$

STRUTS

$$A_p = \text{CROSS SECTION AREA} = 1.2 \text{ ft}^2$$

$$V_p = \text{VELOCITY} = Q/A_p = 12.08 / 1.2 = 35.81 \text{ ft/sec}$$

$$L = \text{STRUT LENGTH} = 1.2'$$

$$Re = \frac{V_p L}{\mu} = \frac{(35.81)(1.2)}{1.24 \times 10^{-5}} = 9.62 \times 10^5$$

$$C_f = .00442$$

$$\frac{1}{Re} = \text{STRUT FRICTION} = 1.04 \times 10^{-6} \quad \frac{1}{Re} = .0938 \times 10^{-5}$$

$$C_D = 2(C_f + 0.0008)(1 + 1.2 \frac{1}{Re}) = 2(.00442 + .0008)(1 + 1.2 \times .0938) = .0116$$

$$S = \text{STRUT PLANFORM AREA} = 2(1.665 - .333)(3.33) = .8884 \text{ ft}^2$$

$$D = \text{STRUT DRAG} = C_D S (\frac{1}{2}) V_p^2 = (.0116)(.8884)(\frac{1}{2})(35.81)^2 = 13.22$$

$$HL = \frac{D}{\rho g A_p} = \frac{13.22}{(32.2)(1.2)} = .0980'$$

PIPE INLET LOSS

PIPE ENTRANCE	2.0470
PIPE FRICTION & FINE	.0320
ELBOW	.0132
TERMINATION	.0301
PIPE LOSS TOTAL	.0783
PIPE	<u>.0783</u>

$$H_{L_I} = 2.9992'$$

$$K = \frac{H_{L_I}}{Q_{max}^2} = \frac{2.9992}{(75)^2} = .000533$$

$$H_{L_I} = .000533 Q^2$$

Friction

$$Q = \text{NOMINAL FLOW RATE} = 251 \text{ gpm}$$

$$A_p = \text{PIPE CROSS SECTION AREA} = 2.0942 \text{ ft}^2$$

$$V_p = \text{PIPE VELOCITY} = Q/A_p = 251 \text{ gpm} \times 1.48 \text{ ft}^3/\text{min} / 2.0942 \text{ ft}^2 = 35.81 \text{ ft/sec}$$

$$d = \text{CASING DIAM.} = 20 \text{ in} = 1.667 \text{ ft}$$

$$Re = \frac{V_p d}{\nu} = \frac{(35.81 \text{ ft/sec})(1.667 \text{ ft})}{1.24 \times 10^{-5} \text{ ft}^2/\text{sec}} = 4.81 \times 10^6$$

$$e/d = \text{RELATIVE ROUGHNESS} = .00005 / 1.667 = .00003$$

$$f = \text{FRICTION FACTOR} = .0092$$

$$L_e = \text{CASING LENGTH} = 4'$$

$$HL = f \left(\frac{L_e}{d} \right) \frac{V_p^2}{2g} = (.0092) \left(\frac{4}{1.667} \right) \frac{(35.81)^2}{2(32.2)} = .4396'$$

CASING DIVERGENCE

$$A_p = \text{CROSS SECT. AREA OF PIPE} = 2.0942 \text{ ft}^2$$

$$A_j = \text{CROSS SECT. AREA OF JUNCTION} = \frac{(30)(18) - (8)^2 + (285)(8)^2}{144} = 2.404 \text{ ft}^2$$

$$K = \text{LOSS COEFF.} = \left(1 - \frac{A_p}{A_j} \right)^2 = \left(1 - \frac{2.0942}{2.404} \right)^2 = .0166 \quad (\text{CRANE - SUDDEN EXP.})$$

$$V_p = \text{PIPE VELOCITY} = Q/A_p = 251 \text{ gpm} \times 1.48 \text{ ft}^3/\text{min} / 2.0942 \text{ ft}^2 = 35.81 \text{ ft/sec}$$

$$HL = K \frac{V_p^2}{2g} = .0166 \frac{(35.81)^2}{2(32.2)} = .0305'$$

Example 2

2.0

$$A_R = \text{RUDDER SECT. AREA} = 7.404 \text{ ft}^2$$

$$V_R = \text{VELOCITY} = Q/A_R = 75/2.404 = 31.20 \text{ FT/SEC}$$

$$C = \text{RUD. CHORD} = 26.25/12 = 2.19'$$

$$Re = \frac{V_R C}{\mu} = \frac{(31.20)(2.19)}{1.24 \times 10^{-5}} = 5.51 \times 10^6$$

$$C_f = .00323$$

$$t/c = \text{RUDDER THICKNESS RATIO} = .25/26.25 = .03$$

$$C_D = 2(C_f + .0005)(1 + 1.2 t/c) = 2(.00323 + .0005)(1 + 1.2 \times .03) = .0084$$

$$S = \text{RUDDER PLANFORM AREA} = (2.19)(1.66) = 3.66 \text{ ft}^2$$

$$D = \text{RUDDER DRAG} = C_D S \rho/2 V_R^2 = (.0084)(3.66)(1/2)(31.20)^2 = 29.93 \text{ lb}$$

$$HL = \frac{D}{\rho g A_R} = \frac{29.93}{2(62.4)(2.404)} = .197'$$

TOTAL CASING LOSS

CASING FRICTION .55'
CASING DRAG LOSS .
RUDDER .

$$HL_c = .55'$$

$$k = \frac{HL_c}{Q_{min}^2} = .000171$$

$$HL_c = .000171 Q^2$$

REVERSE ELBOW

$$K = \text{LOSS COEFF.} = 1 + 1.5 = 2.5 \quad (\text{STANDARD LOSS COEFF. - CORRECTION})$$

$$A_R = \text{REVERSE NET AREA} = \frac{(20)(5.5)}{144} = .764 \text{ FT}^2$$

$$V_R = \text{REVERSE NET VELOCITY} = Q/A_R = 25/.764 = 98.17 \text{ FT/SEC}$$

$$HL = K \frac{V_R^2}{2g} = (2.5) \frac{(98.17)^2}{2(32.2)} = 374.12'$$

TOTAL REVERSE SYSTEM LOSS

CASING FRICTION	.44
CASING DIVERGENCE	.33
REVERSE ELBOW	<u>374.12</u>

$$HL_R = 374.89'$$

$$k = \frac{HL_R}{Q_{nom}^2} = \frac{374.89}{(25)^2} = .0666$$

$$HL_R = .0666 Q^2$$

Required Head

$$H_{REQ} = H_{L_1} + H_{L_2} + H_{L_3} - H_0$$

$$H_{L_1} = f \frac{L}{D} \frac{V^3}{2g}$$

$$V = Q/A$$

$$Q = 2.404 \text{ cfs}$$

$$H_{L_1} = \frac{Q^2}{(2.404)^2 (2) (32.2)} = .00269 Q^2$$

$$H_{L_2} = .000553 Q^2$$

$$H_{L_3} = .000171 Q^2 \quad \left. \begin{array}{l} \\ \end{array} \right\} \text{DEVIATION} \approx 1$$

$$H_0 = (RPL) \frac{V_0^2}{2g}$$

$$RPL = .50$$

$$H_0 = \frac{(.50) V_0^2}{2 (32.2)} = .0078 V_0^2$$

$$\begin{aligned} H_{REQ} &= .00269 Q^2 + .000553 Q^2 + .000171 Q^2 - .0078 V_0^2 \\ &= .00337 Q^2 - .0078 V_0^2 \end{aligned}$$

PROJ. AREA

1. PROJECTIONS

r	C	T.H.	f(1/r)
2	10.70	1/2	5.35
3	12.25	1	12.25
4	13.70	1	13.70
5	14.94	1	14.94
6	16.00	1	16.00
7	16.90	1	16.90
8	17.50	1	17.50
9	17.85	1	17.85
10	18.00	1/2	9.00
			123.49

$$A_x = (3)(1)(123.49) = 370.47 \text{ in}^2$$

$$A_0 = (.785)(20)^2 = 314 \text{ in}^2$$

$$L.R. = \frac{370.47}{314} = 1.18$$

PROJECTED AREA

r	C	α	Corr.	T.H.	f(1/r)
2	10.70	27.86	5.69	1/2	2.84
3	12.25	33.70	8.40	1	8.40
4	13.70	39.51	10.72	1	13.72
5	14.94	44.11	12.60	1	14.94
6	16.00	47.95	14.13	1	16.00
7	16.90	51.15	15.38	1	16.90
8	17.50	53.70	16.26	1	17.50
9	17.85	55.15	16.83	1	17.85
10	18.00	56.11	17.15	1/2	8.57
					105.74

$$\alpha = \arctan\left(\frac{P}{2\pi r}\right) / r$$

$$= \arctan\left(\frac{20}{2\pi r}\right) / r$$

$$= \arctan 3.1831 / r$$

$$A_p = (3)(1)(105.74) = 317.22 \text{ in}^2$$

$$A_0 = 314 \text{ in}^2$$

$$P.A.R. = \frac{317.22}{314} = 1.01$$

STRUCTURAL APL.

- PROPELLER
- SHAFT
- RUDDER
- DECK

PROPELLER

$$R = \text{PROP. RADIUS} = 10''$$

$$Z = \text{NO. OF BLADES} = 3$$

$$P = \text{PITCH} = 20''$$

$$T = \text{PROP. THRUST} = \rho g H_p A_1 = (64.4)(1)(2.0942) = 3115^*$$

$$N = \text{PROP. SPEED} = \frac{60 Q}{\rho_0 V_{tip} D} = \frac{(60)(18.5)}{(2.0942)(18.5)(11.4)} = 355 \text{ RPM}$$

$$Q' = \text{PROP. TORQUE} = 63024 \frac{54P}{N} - \frac{(63024)(6W)}{55} = 11717 \text{ IN}^*$$

$$a = \frac{2\pi R}{P} = \frac{2\pi(10)}{20} = 3.1416$$

$$x = r/R$$

$$K = f(x)$$

TABLE I

CONOLLY

$$A_1 = f(a, x)$$

" II

"

$$A_2 = f(a, x)$$

" III

"

$$B_1 = f(a, x)$$

" IV

"

$$B_2 = f(a, x)$$

" V

"

$$C_1 = f(x)$$

" VI

"

$$C = \text{SECTION CHORD}$$

$$t = \text{MAX. SECTION THICKNESS}$$

$$\sigma_R = \text{SPANWISE BENDING STRESS} = \frac{RK}{Z C t^2} \left[A_1 \left(\frac{2\pi R T}{P} \right) + A_2 \left(\frac{Q}{x} \right) \right]$$

$$\sigma_0 = \text{CHORDWISE BENDING STRESS} = \frac{RK}{Z C t^2} \left[B_1 \left(\frac{2\pi R T}{P} \right) + B_2 \left(\frac{Q}{x} \right) \right]$$

$$\sigma_c = \text{CENTRIFUGAL STRESS} = \frac{2240 N^2 R^2 C_1}{10^{10}}$$

$$\sigma_{T \text{ MAX}} = \text{MAX. TENSILE STRESS (FACE)} = \sigma_R + \sigma_c$$

$$\sigma_{S \text{ MAX}} = \text{MAX. SHEAR STRESS (CORE)} = \frac{\sigma_T - \sigma_0}{2}$$

PROPELLER STRESS CALCULATION

BENDING STRESSES

R	Z	P	T	Q	a	X	K	A ₁	A ₂	c	t	OR	B ₁	B ₂	O ₀
10.00	3	20	3115	11717	3.1416	.20	.1203	6.90	57.95	10.70	.725	9586	2.26	19.30	3189
						.30	.1207	6.94	39.26	12.25	.638	9192	3.13	18.69	4230
						.40	.1007	7.27	31.20	13.70	.550	8723	3.97	19.25	5291
						.50	.0720	7.96	27.63	14.94	.463	8208	4.79	19.88	52157
						.60	.0437	8.74	22.00	16.00	.375	7585	5.63	20.33	5755
						.70	.0214	10.00	27.95	16.90	.288	6647	6.58	20.90	4601
						.80	.0075	11.61	20.10	17.50	.201	5265	7.65	23.50	3633
						.90	.0013	12.53	30.32	17.65	.113	3007	8.44	24.24	2121

CENTRIFUGAL STRESS

R	N	X	C ₁	Y	OR	O ₀	O ₂	A _{TRAY}	A _{TRAY}
10.00	355	.20	16.6	.20	9586	3189	47	9633	3222
		.30	12.0	.30	9192	4238	34	9226	2494
		.40	9.5	.40	8723	4974	27	8750	1888
		.50	7.7	.50	8208	5267	22	8230	1482
		.60	6.2	.60	7585	5755	18	7603	1224
		.70	4.8	.70	6647	4601	14	6661	1030
		.80	3.4	.80	5265	3633	10	5275	821
		.90	1.9	.90	3007	2121	5	3012	446

TORSIONAL STRESS

$$Q = \text{TORSION MOM. IN SHAFT} = 11717 \text{ IN}^{\#}$$

$$d = \text{SHAFT DIAM.} = 2', \quad r = 1''$$

$$J = \text{POLAR MOM. OF INERTIA} = \frac{\pi}{32} d^4 = \frac{\pi}{32} (2)^4 = 1.5708$$

$$s_s = \text{TORSIONAL STRESS IN SHAFT} = \frac{Qr}{J} = \frac{(11717)(1)}{1.5708} = 7459 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{20000}{7459} = 2.68 \text{ ON SHEAR YIELD (6061-T6)}$$

WHIRLING FREQUENCY

$$W = \text{WEIGHT PER UNIT LENGTH OF SHAFT} = (.285)(2)(.09) = .3014 \text{ #/IN}$$

$$L = \text{DISTANCE BETWEEN SUPPORTS} = 82''$$

$$I = \text{MOM. OF INERTIA OF SHAFT} = .049 d^4 = (.049)(2)^4 = .284 \text{ IN}^4$$

$$D = \text{STATIC DEFLECTION OF SHAFT DUE TO OWN WEIGHT}$$

$$= \frac{.00542 W L^4}{EI} = \frac{(.00542)(.3014)(82)^4}{(10,200,000)(.284)} = .0092'' \quad (\text{Euler - first})$$

$$f = \text{WHIRLING FREQUENCY} = \frac{3.55}{D^{1/2}} = \frac{3.55}{(.0092)^{1/2}} = 36.2 \text{ cps}$$
$$= 2216 \text{ RPM}$$

$$N_{DES} = \frac{(24.3)(60)}{(2.0412)(.875)(1.667)} = 1421 \quad (\text{Empirical})$$

STRUCTURE

PLATE

$$p = \text{DESIGN PRESSURE} = 23.1' = 10.5 \text{ psi} \quad (\text{REVISIT Q.7 CLOS.})$$

$$l = \text{SPAN} = 19.5''$$

$$b = \text{PANEL WIDTH} = 26''$$

$$M = \text{BENDING MOMENT IN PLATE} = \frac{p l^2 b}{8} = \frac{(10.5)(19.5)^2(26)}{8} = 12803 \text{ in}^2$$

$$t = \text{PLATE THICKNESS} = .75''$$

$$Z = \text{SECTION MODULUS OF PLATE} = \frac{b t^2}{6} = \frac{26(.75)^2}{6} = 2.4375 \text{ in}^3$$

$$S_M = \text{BENDING STRESS IN PLATE} = \frac{M}{Z} = \frac{12803}{2.4375} = 5253 \text{ psi}$$

$$Q = \text{EFFECTIVE MOMENT ARM} = 2.66''$$

$$Q = \text{RUDDER TORQUE} = p l b Q = (10.36)(20)(26)(2.66) = 14330 \text{ in}^2$$

$$S_Q = \text{SHEAR STRESS IN PLATE} = \frac{3 Q}{b t^2} = \frac{3(14330)}{26(.75)^2} = 2939 \text{ psi}$$

$$S_{SMY} = \text{MAX. COMBINED SHEAR STRESS IN PLATE} = \sqrt{\left(\frac{S_M}{2}\right)^2 + (S_Q)^2}$$
$$= \sqrt{\left(\frac{5253}{2}\right)^2 + (2939)^2}$$
$$= 3942 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{35000/2}{3942} = 4.44 \text{ ON YIELD (6061-T6)}$$

$$= \frac{40000/2}{3942} = 5.07 \text{ ON YIELD (POLYMER G.A.)}$$

STOCK

$$Q = \text{RUDDER TORQUE} = 14330 \text{ in}^2$$

$$d = \text{STOCK DIAM.} = 2.00'' , r = 1.00''$$

$$J = \text{POLAR MOM. OF INERTIA} = \frac{\pi r^4}{2} = \frac{\pi (1)^4}{2} = 1.5708$$

$$S_s = \text{SHEAR STRESS IN STOCK} = \frac{Q r}{J} = \frac{(14330)(1)}{1.5708} = 9123 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{30,000}{9123} = 3.29 \text{ ON WELDED YIELD (316 STAINLESS)}$$

FLANGE

$$Q = \text{TORSION IN RUDDER STOCK} = 14330 \text{ IN}$$

$$d = \text{STOCK DIAM} = 2.00" \quad r = 1.00"$$

$$t = \text{FLANGE THICKNESS} = .25"$$

$$A_s = \text{SHEAR AREA IN FLANGE} = \pi d t = \pi (2)(.25) = 1.5708 \text{ IN}^2$$

$$s_s = \text{SHEAR STRESS IN FLANGE} = \frac{Q}{A_s} = \frac{14330}{(1.5708)(1)} = 9123 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{30,000}{9123} = 3.29 \text{ ON WELDED YIELD (316 STAINLESS)}$$

ATTACH BOLTS

$$p = \text{DESIGN PRESSURE} = 10.36 \text{ psi}$$

$$S = \text{RUDDER AREA} = (20)(26) = 520 \text{ IN}^2$$

$$F = \text{LOAD NORMAL TO RUDDER} = p S = (10.36)(520) = 5387 \text{ #}$$

$$A_t = \text{TENSILE AREA IN BOLTS} = 6(.047) = .2820 \text{ IN}^2 \quad 12, 5/16-18 \text{ BOLTS}$$

$$s_t = \text{TENSILE STRESS IN BOLTS} = \frac{F}{A_t} = \frac{5387}{.2820} = 19103 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{30,000}{19103} = 1.57 \text{ ON YIELD (316 STAINLESS)}$$

PLATE

$$P = \text{DESIGN PRESSURE} = 23.1' = 10.36 \text{ psi}$$

$$L = \text{SPAN} = 20''$$

$$W = \text{UNIT WIDTH} = 1''$$

$$M = \text{BENDING MOMENT IN PLATE} = P \frac{L^2 W}{12} = \frac{(10.36)(20)^2(1)}{12} = 345.33 \text{ in}^3/\text{in width}$$

$$t = \text{PLATE THICKNESS} = .375''$$

$$Z = \text{SECTION MODULUS OF PLATE} = \frac{W t^2}{6} = \frac{(1)(.375)^2}{6} = .0234 \text{ in}^3/\text{in width}$$

$$S = \text{BENDING STRESS IN PLATE} = \frac{M}{Z} = \frac{345.33}{.0234} = 14758 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{19000}{14758} = 1.29 \text{ ON WELDED YIELD (SDS-1116)}$$

$$= \frac{40,000}{14758} = 2.71 \text{ ON FLEX. YIELD POLYESTER/GLASS FIBER}$$

VERTICAL LOUVRE

$$p = \text{DESIGN PRESSURE} = 10.36 \text{ psi}$$

$$L = \text{SPAN} = 5''$$

$$C = \text{CHORD} = 5''$$

$$M = \text{BENDING MOMENT} = \frac{p L^2 C}{12} = \frac{(10.36)(5)^2(5)}{12} = 107.92 \text{ in}^3$$

$$t = \text{PLATE THICKNESS} = .1875''$$

$$Z = \text{SECTION MODULUS} = \frac{wt^3}{6} = \frac{(5)(.1875)^3}{6} = .0293 \text{ in}^3$$

$$S = \text{BENDING STRESS} = \frac{M}{Z} = \frac{107.92}{.0293} = 3683 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{19000}{3683} = 5.16 \text{ ON NICKEL VELD (5086-H116)}$$

HORIZONTAL LOUVRES

$$p = \text{DESIGN PRESSURE} = 10.36 \text{ psi}$$

$$L = \text{SPAN} = 13''$$

$$b = \text{PANEL WIDTH} = 5''$$

$$K = \text{DISTRIBUTION FACTOR} = \frac{12-5}{12} = 1.23$$

$$M = \text{BENDING MOMENT} = \frac{1}{8} p \frac{L^2 b}{K} = \frac{(1.23)(10.36)(13)^2(5)}{8} = 1346 \text{ in}^3$$

$$Z = \text{SECTION MODULUS} = \frac{(.1875)(3)^3}{6} = .125 \text{ in}^3$$

$$S = \text{BENDING STRESS} = \frac{M}{Z} = \frac{1346}{.125} = 10768 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{19000}{10768} = 1.76 \text{ ON NICKEL VELD (5086-H116)}$$

Example Cylinder (2.50" diam. x 4.00" long)

$$M = \text{REQUIRED MOMENT} = 13522 \text{ in}^2$$

$$A_{\text{CYL}} = \text{CYL. AREA} = (.785)(2.50)^2 = 4.9063 \text{ in}^2$$

$$P_{\text{CYL AVAIL}} = \text{CYL. PRESS.} = 1000 \text{ psi}$$

$$F_{\text{CYL AVAIL}} = \text{AVAILABLE CYL. FORCE} = P_{\text{CYL}} A_{\text{CYL}} = (1000)(4.9063) = 4906 \text{ lb}$$

$$r = \text{CRANK RADIUS} = 3.50 \text{ in}$$

$$\theta = \text{TOTAL TRAVEL} = 45^\circ$$

$$a = \text{EFFECTIVE MOMENT ARM} = r \cos \frac{\theta}{2} = (3.50) \cos 22.5^\circ = 3.23 \text{ in}$$

$$M_{\text{AVAIL}} = (F_{\text{CYL AVAIL}})(a) = (4906)(3.23) = 15846 \text{ in}^2 \quad (\text{O.K.})$$

$$L_{\text{REQ}} = \text{REQ. STROKE} = 2r \sin \frac{\theta}{2} = 2(3.50) \sin 22.5^\circ = 2.68 \text{ in} \quad (\text{O.K.})$$

ESTIMATED VOLUME

$$A = (20.7 - 16) / 144 = 6.913 \text{ ft}^2$$

$$x = .375", \quad w = (.375)(144)(10.2) = 5.18 \text{ #/ft}^2$$

$$W = (6.913)(5.18) = 36.16 \text{ #}$$

BACKING RING

$$V = .785 [(22.75)^2 - (20.875)^2] (1) = 64.21 \text{ in}^3$$

$$w = .096 \text{ #/in}^3$$

$$W = 64.21 (0.096) = 6.16 \text{ #}$$

FLANGE

$$V = .785 [(22.75)^2 - (20.75)^2] (.5) = 34.15 \text{ in}^3$$

$$w = .096 \text{ #/in}^3$$

$$W = 34.15 (0.096) = 3.28 \text{ #}$$

ORIFICE

$$V = 4(4)(.375)(.71)(.5) = 34.08 \text{ in}^3$$

$$w = .096 \text{ #/in}^3$$

$$W = 34.08 (0.096) = 3.27 \text{ #}$$

BEARING LUGS

STA.	R	C	T.M.	F(Y)
1	4.00	12.56	1	5.28
2	4.00	12.56	1	12.56
3	3.50	9.62	1	9.62
4	2.75	5.94	1/2	2.97
				31.43

$$V = 4(31.43) - (8)(.785)(2.625)^2 - (4)(.785)(2.75)^2 = 66.15$$

$$w = .096 \text{ #/in}^3$$

$$W = 6.39 \text{ #}$$

TOTALS

$$W = 36.16 \quad \times 1.50 / 2.66$$

$$6.16$$

$$3.28$$

$$3.27$$

$$6.39$$

$$\underline{55.26 \text{ #}}$$

CONV.

$$\underline{31.16 \text{ #}}$$

PLAST.

CIRCULAR DUCT

$$A = (20)\pi(3)/144 = 1.57 \text{ FT}^2$$

$$W = .875' \quad w = (.875)(5.18) = 5.18 \text{ #/FT}^2$$

$$W = (1.309)(5.18) = 6.78 \text{ #}$$

RECTANGULAR DUCT

$$A = [8\pi + 2(10+12)]5/144 = 2.4004 \text{ FT}^2$$

$$W = 5.18 \text{ #/FT}^2$$

$$W = (2.4004)(5.18) = 12.43 \text{ #}$$

TRANSITION DUCT

$$A = \left(\frac{62.832 + 69.1328}{2} \right) 10/144 = 4.5821 \text{ FT}^2$$

$$W = 5.18 \text{ #/FT}^2$$

$$W = (4.5821)(5.18) = 23.74 \text{ #}$$

REVERSE DUCT

$$A = \left[\frac{8\pi}{2} + 15+12+15 \right] 14/144 = 5.3051 \text{ FT}^2$$

$$W = 5.18 \text{ #/FT}^2$$

$$W = (5.3051)(5.18) = 27.48 \text{ #}$$

REVERSE DUCT FLANGE

$$A_1 = 20(4+5) - \frac{(285)(5)}{2} - (2)(4) = 106.88$$

$$A_2 = 20(12) = 240$$

$$A_3 = 2(12)(2) = 64$$

$$A = \frac{410.88}{144} = 2.8533 \text{ FT}^2$$

$$W = 5.18 \text{ #/FT}^2$$

$$W = (2.8533)(5.18) = 14.78 \text{ #}$$

REVERSE DUCT LOUVRES (HORIZ.)

$$A = 3(2)(13)/144 = .5417 \text{ FT}^2$$

$$W = 2.592 \text{ #/FT}^2$$

$$W = (.5417)(2.592) = 1.40 \text{ #}$$

Rel. (V₁)

$$A = 3(20)(4)/144 = 1.667 \text{ ft}^2$$

$$st = .125", w = (.125)(144)(.092) = 1.228 \text{ #/ft}^2$$

$$W = (1.667)(1.228) = 2.88 \text{ #}$$

Duct Connector

$$A = (20.75)\pi(4)/144 = 1.8108 \text{ ft}^2$$

$$st = 5.18 \text{ #/ft}^2$$

$$W = (1.8108)(5.18) = 9.38 \text{ #}$$

TOTALS

W =	6.78	x 1.50/2.66	3.82
	12.43	"	7.01
	23.74	"	13.39
	27.48	"	15.50
	14.78	"	8.33
	1.40	x 1.50	1.40
	2.84	"	2.84
	<u>9.38</u>	"	<u>9.38</u>
	98.87		61.71

Conv.

Plastic

BLADES

<u>X</u>	<u>C</u>	<u>f</u>	<u>a</u>	<u>T.M.</u>	<u>f(r)</u>
.20	10.70	.725	5.51	1/2	2.76
.30	12.25	.638	5.55	1	5.55
.40	13.70	.550	5.35	1	5.35
.50	14.94	.463	4.91	1	4.91
.60	16.00	.375	4.26	1	4.26
.70	16.90	.288	3.46	1	3.46
.80	17.50	.201	2.50	1	2.50
.90	17.85	.113	1.43	1	1.43
1.00	18.00	.080	1.02	1/2	.51
					30.73

$$V = 3(1)(30.73) = 92.19 \text{ in}^3$$

$$W = .096 \text{ #/in}^3$$

$$W = (92.19)(.096) = 8.85 \text{ #}$$

HUB

$$V = (10)(.785)(4)^2 - (6)(.785)(1.25)^2 - (4)(.785)(3.25)^2 = 78.01 \text{ in}^3$$

$$W = .096 \text{ #/in}^3$$

$$W = (78.01)(.096) = 7.49 \text{ #}$$

TOTAL

$$8.85$$

$$7.49$$

$$16.34 \text{ #}$$

$$1.40 \text{ #/in}$$

$$8.60 \text{ #}$$

$$D_{\text{max}}$$

PLATE

$$A = (20)(1.5) / 144 = 3.125 \text{ FT}^2$$
$$f = .25", W = (.25)(144)(.092) = 10.37 \text{ #/FT}^2$$
$$W = (3.125)(10.37) = 32.44 \text{ #}$$

FLANGES

$$A = \left[\frac{(2+2)}{2} 9 + 2(12)(2) \right] 2 / 144 = 1.5417 \text{ FT}^2$$
$$f = .25", W = (.25)(144)(.22) = 10.08 \text{ #/FT}^2$$
$$W = (1.5417)(10.08) = 15.54 \text{ #}$$

SOCKS

$$L = (8.0 + 1.5) / 12 = .7917'$$
$$W = (.7917)(2)(12)(.28) = 10.5504 \text{ #/IN}$$
$$W = (.7917)(10.5504) = 8.35 \text{ #}$$

TOTALS

$$W = 32.44$$
$$15.54$$

$$47.98$$

$$67.13 \text{ #}$$

2.00

$$\times 1.00 / 2.00$$
$$\times 1.00$$
$$"$$

$$19.71$$

$$15.54$$

$$8.35$$

$$43.60 \text{ #}$$

COMPOSITE

SINCE

$$L = 10'$$

$$W = (785)(2)^2(12)(60\%) = 3,6173 \text{ #/FT}$$

$$W = 60(3,6173) = 36,173 \approx$$

265.77

STUT (alum)

(base for support)

$$A = (17.5)(2.50)/111 = .8351$$

$$W = (.50)(144)(.09) = 6.91 \text{ #/ft}^2$$

$$W = (.8351)(6.91) = 5.77 \text{ #}$$

TILLER (alum)

$$\nabla = (1)(.785)(4^2 - 2^2) + 2(2)(2)(.5) = 31.68 \text{ m}^3$$

$$W = .096 \text{ #/m}^3$$

$$W = (31.68)(.096) = 3.04 \text{ #}$$

BEAMS, SPARKS, WEIR (TERRAIN)

$$\nabla = (3.6)(.785)(4.50)^2 + (5.50 - 3.60)(.785)(4)^2 - (5.5)(.785)(2)^2 = 125.99 \text{ m}^3$$

$$W = (.096)(\frac{4.50}{2.50}) = .054 \text{ #/m}^3$$

$$W = (125.99)(.054) = 6.80 \text{ #}$$

TOTALS

$$W = 5.77$$

$$3.04$$

$$6.80$$

$$15.61 \text{ #}$$

<u>Sta</u>	<u>Ord.</u>	<u>a</u>	<u>T.M.</u>	<u>(a²)</u>
0	11.5	226	1/2	158
1	9	216	1	216
2	6	144	1	144
3	3.25	28	1	28
4	1.25	30	1	30
5	0	0	1/2	0
				<u>476</u>

$$T = (1.5)(600)/1200 = 4.21 \text{ FT}^3$$

$$W = 6 \text{ FT}^2$$

$$V = (4.21)(6) = 25.26 \text{ FT}^3$$

Duct Fairing

$$V = (12)[(24)(24) - (28.5)(20)^2](.67) = 2107.23 = 1.22 \text{ FT}^3$$

$$W = 60 \text{ FT}^2$$

$$V = (1.22)(30) = 36.60 \text{ FT}^3$$

1.22

1.22

61.80

REVISION

$$W = 233.58 - 10.08 - 2.35 - .561 = 219.54 \text{ #}$$

$$\nabla = 219.54 / 166 = 1.30 \text{ ft}^3$$

$$B = (64)(1.30) = 96 \text{ #}$$

WIND

$$W = 10.08 + 8.35 = 18.43 \text{ #}$$

$$\nabla = 18.43 / 484 = .0381 \text{ ft}^3$$

$$B = (.0381)(64) = 2.44 \text{ #}$$

INLET FAIRING

$$\nabla = 4.21 \text{ ft}^3$$

$$B = (4.21)(64) = 269.44 \text{ #}$$

DUST FAIRING

$$\nabla = 1.22 \text{ ft}^3$$

$$B = (1.22)(64) = 78 \text{ #}$$

TOTAL

$$B = 96.00$$

$$2.44$$

$$269.44$$

$$\underline{78.08}$$

$$445.96 \text{ #}$$

Drive Shaft & Joint	42.50
Drive Shaft Cover	17.00
Art U-Joint Support	2.90
Art Drive Shaft	34.04
Worm & Assembly	304.10
Flange Nut	18.50
Min.	<u>11.00</u>
	435.34

WEIGHT SUMMARY

	<u>CONVENTIONAL CONSTR.</u>		<u>COMPOSITE CONSTR.</u>	
	<u>WT.</u>	<u>Notes</u>	<u>WT.</u>	<u>Notes</u>
PROP, DUCT	55.36	Alum. (30% - 1116)	31.16	POLYCARB - GLASS (20% + Alum. 60%)
RUDDER DUCT	0.00	" "	61.01	" " "
PROPELLER	16.34	Alum. (30% - 70)	8.60	POLYCARB - 30% GLASS
RUDDER	0.00	Alum. (40% - 70)	43.60	POLYCARB - GLASS (20% - 70)
SHAFT	0.15	" "	36.15	Alum. (100% - 70)
MISC.	0.11	" "	15.61	Various Metals, Alum. Sheet, etc.
			196.85	" - 40% - 70 = 196.85
FAIRINGS	61.86	Various	61.86	Various
	345.44		258.21	
BUOYANCY	100.52		1415.96	" - 162.97 = 1415.96
	- 100.52		- 182.25	" + 262.12 = 1415.96

EXISTING PUMP

- Static Pressure
- Gauge Pressure
- Comparison

EXERCISE

STATIC PERFORMANCE

$$T_{STAT} = 3025 \text{ lbf} \quad (\text{KNOWN})$$

$$A_0 = (.785)(10.66)^2 / 144 = .6453 \text{ ft}^2 \quad (\text{KNOWN})$$

$$V_0 = \sqrt{\frac{T_{STAT}}{\rho A_0}} = \sqrt{\frac{3025}{2(.6453)}} = 48.41 \text{ ft/sec}$$

$$Q = V_0 A_0 = (48.41)(.6453) = 31.24 \text{ ft}^3/\text{sec} = 14020 \text{ GPM}$$

$$H_p = H_L + H_L + H_0$$

$$H_L = \frac{V_0^2}{2g} = \frac{(48.41)^2}{2(32.2)} = 36.39'$$

$$H_L = .000533 Q^2 = (.000533)(31.24)^2 = .52'$$

$$H_0 = 0 \quad (\text{STATIC})$$

$$H_p = 36.39 + .52 - 0 = 36.91'$$

$$WHP = \frac{\rho g Q H_p}{550} = \frac{(2)(32.2)(31.24)(36.91)}{550} = 135$$

$$SHP = 200$$

$$\eta_{PUMP + NOZZLE} = \frac{WHP}{SHP} = \frac{135}{200} = .675$$

SEEMS LOW PERHAPS
PUMP IS NOT USING ALL
AVAILABLE POWER

CRUISE PERFORMANCE

$$V_0 = 8 \text{ MPN} = 11.76 \text{ ft/sec}$$

$$RPR = .50$$

$$H_0 = (RPR) \frac{V_0^2}{2g} = (.50) \frac{(11.76)^2}{2(32.2)} = 1.07'$$

$$H_L = H_p - H_L + H_0 = 36.91 - .52 + 1.07 = 37.46'$$

$$V_0 = \sqrt{2g H_L} = \sqrt{2(32.2)(37.46)} = 49.12 \text{ ft/sec}$$

$$Q = V_0 A_0 = (49.12)(.6453) = 31.70 \text{ ft}^3/\text{sec}$$

$$T = \rho Q (V_0 - V_0) = 2(31.70)(49.12 - 11.76) = 2369 \text{ lbf}$$

$$P.E. = \frac{T V_0}{550 SHP} = \frac{(2369)(11.76)}{(550)(200)} = .2533$$

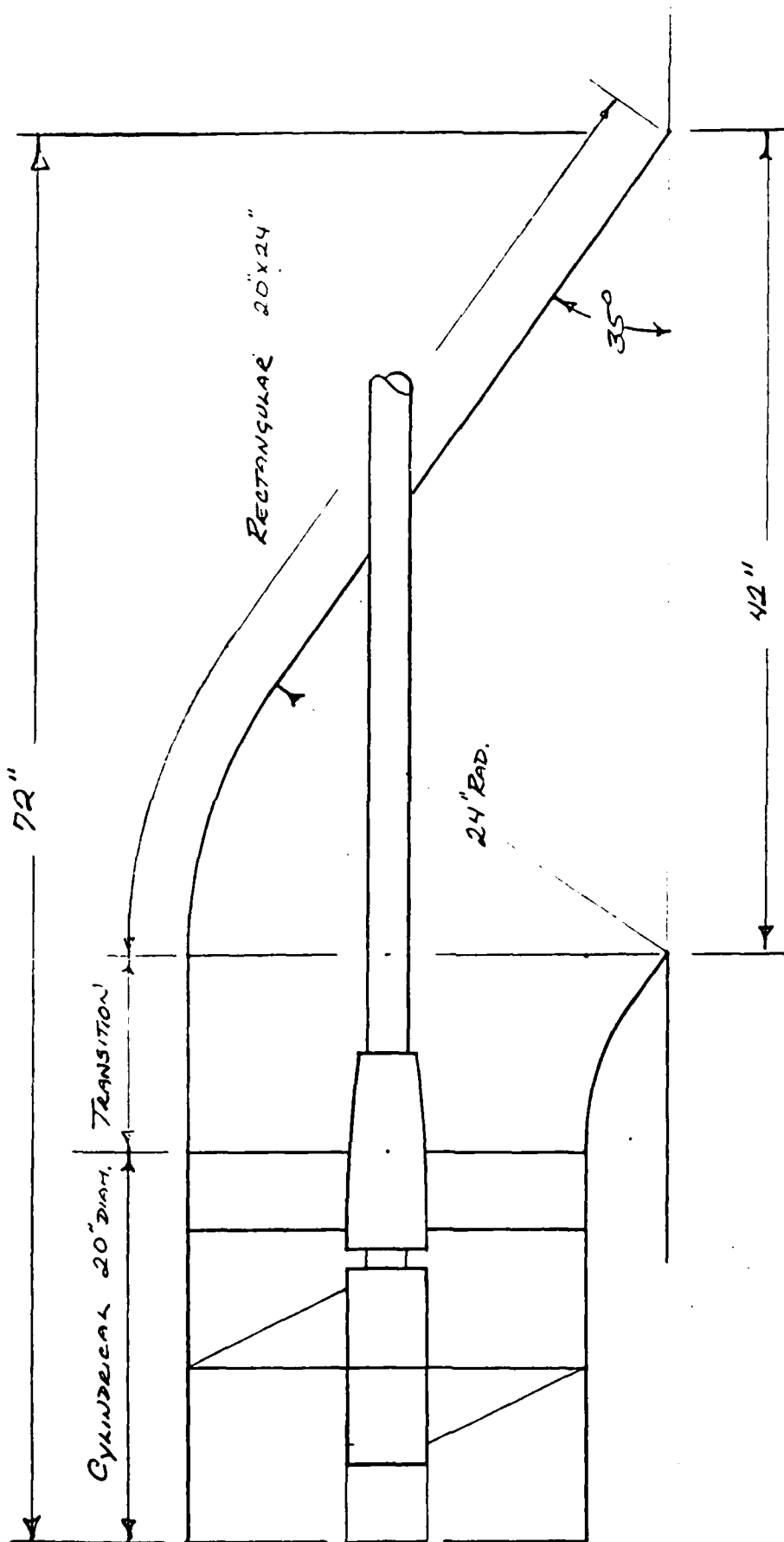
Comparison

	<u>Field System</u>	<u>Factory</u>
THRUST AT 8 MPH	2845 [#]	2369 [#]
P.C. AT 8 MPH	.30	.25
FLOW RATE AT 8 MPH	33346 GPM	14020 GPM
STATIC THRUST	4128 [#]	3025 [#]
DRY WEIGHT	284 [#] Conv. Const.	435 [#]
	197 [#] Corp. Const.	

APPENDIX B

OBJECTIVES:

- Determine performance, weight and dimensional characteristics of a propulsion pump, about the same size as the PJ-16, for use in a high-speed (20 mph) amphibian.
- Use simple "propeller-in-tube" approach.
- Limit blade area ratio to that available in existing propeller series.
- Investigate use of composite materials.



SKETCH OF 20 INCH DIAMETER PUMP

PERFORMANCE CHARACTERISTICS

- POWER LIMITS
- CAVITATION LIMITS
- SYSTEM PERFORMANCE
- SYSTEM OPERATION
- DERIVATIONS & REF. MATL.

CALCULATION / 1072

$$D = \text{PROP. DIAM.} = 2D' = 1.667'$$

$$A_p = \text{PROP. DISC. AREA} = .785[(1.667)^2 - (.333)^2] = 2.0942 \text{ FT}^2$$

$$A_i = \text{INLET AREA} = (1.667)(2.00) = 3.333 \text{ FT}^2$$

$$J_{cyl} = \text{ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE}$$

$$J_e = \text{ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE}$$

$$\frac{J_{cyl}}{J_e} = \frac{A_i}{A_p} = \frac{3.333}{2.0942} = 1.5915$$

$$P/D = \text{PROP. PITCH DIAM. RATIO}$$

$$J_{cyl} = f(P/D, \frac{J_{cyl}}{J_e}) = .905 \quad (\text{VON KARMAN} - \text{FIG. 1})$$

$$J_e = J_{cyl} / (\frac{J_{cyl}}{J_e}) = .905 / 1.5915 = .5686$$

$$\eta_o = \text{PROPULSION EFFICIENCY OF PROP-TUBE COMB.} = f(J_e, P/D) \quad (\text{FIG. 2})$$

$$\eta_{rel} = \text{PROP. RELATIVE RORR EFFICIENCY} = .95 \quad (\text{GROSS-TUNNEL PWR.})$$

$$\eta_p = \text{PUMP EFFICIENCY} = \frac{J_{cyl} \eta_o \eta_{rel}}{J_e}$$

$$\text{SHP} = \text{PUMP INPUT POWER}$$

$$Q = \text{FLOW RATE IN FT}^3/\text{SEC}$$

$$\text{PUMP HORSEPOWER} = \frac{550 \text{ SHP } \eta_p}{\rho g Q}$$

Power Limit

CALCULATION

[illegible]

CALCULATION NOTES

$$D = \text{PROP. DIAM.} = 20'' = 1.667'$$

$$A_p = \text{PROP. DISC AREA} = .785 [(1.667)^2 - (.333)^2] = 2.0942 \text{ ft}^2$$

$$A_i = \text{INLET AREA} = (1.667)(2.00) = 3.333 \text{ ft}^2$$

J_{cyl} = ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE

J_e = ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE

$$\frac{J_{\text{cyl}}}{J_e} = \frac{A_i}{A_p} = \frac{3.333}{2.0942} = 1.5915$$

P/D = PROP. PITCH DIAM. RATIO

$$J_{\text{cyl}} = f(P/D, \frac{J_{\text{cyl}}}{J_e})$$

(VON KARMAN - FIG. 1)

$$Q = \text{FLOW RATE} \sim \text{FT}^3/\text{SEC}$$

$$N = \text{PROP. SPEED} = \frac{Q}{A_p J_{\text{cyl}} D}$$

$$H_{L_i} = \text{INLET HEAD LOSS} = .000234 Q^2 \quad (\text{DERIVATION \#1})$$

$$V_i = \text{INLET VELOCITY} = Q/A_i$$

$$V_{\text{TR}} = \text{TANGENTIAL VELOCITY AT DRAG OF PROP.} = .707 D N$$

$$V_{\text{TR}} = \text{TOTAL VELOCITY AT DRAG OF PROP.} = \sqrt{V_i^2 + V_{\text{TR}}^2}$$

$$H_{\text{LH}} = \text{HEAD LOSS DUE TO HEAD LOSS}$$

$$H_e = \text{HEAD DUE TO ELEVATION}$$

$$H_v = \text{HEAD DUE TO VAPOR PRESSURE}$$

CALCULATION NOTE: (CONT.)

V_0 = FREE STREAM VELOCITY = CRAFT SPEED

RPR = RAM PRESS. RECOVERY RATIO = .70 (JACUZZI - FIG. 3)

H_0 = RAM HEAD RECOVERED = $(RPR) \frac{V_0^2}{2g}$

H_{I_3} = INLET STATIC HEAD (ABOVE VAP. PRESS.) = $H_{ATM} + H_2 - H_V + H_0 - H_{L1} - \frac{V_1^2}{2g}$

P_{I_3} = INLET STATIC PRESS. (ABOVE VAP. PRESS.) = $\rho g H_{I_3}$

σ_{LR} = LOCAL CAV. NO. AT 1/2 RAD. = $\frac{P_{I_3}}{\rho/2 V_{LR}^2}$

T_{EMAX} = PRODP. LOAD COEF. AT CAV. LIMIT = .7 σ_{LR} (GAWN)

EAR = PRODP. EXPANDED AREA RATIO = 1.18 (MAX. AVAILABLE)

PAR = 1/1 PROJECTED AREA RATIO = 1.01 (DERIVATION #3)

$H_{PCAV.}$ = PUMP HEAD RISE AT CAV. LIMIT = $\frac{(T_{EMAX})(PAR)(V_{LR})^2(\rho/2)}{\rho g}$

CALCULATION NOTES

• REQUIRED PUMP HEAD

$$V_0 = \text{CRAFT SPEED}$$

$$Q = \text{FLOW RATE} \sim \text{FT}^3/\text{SEC}$$

$$H_{PRQ} = .00381 Q^2 - .0109 V_0^2 \quad (\text{DERIVATION \#2})$$

• ESTIMATED THRUST (POWER LIMIT)

$$V_0 = \text{CRAFT SPEED}$$

$$\text{SHP} = \text{PUMP INPUT POWER}$$

$$Q_{EQ} = \text{EQUILIBRIUM FLOW RATE @ } H_{PAVAIL} = H_{PRQ}$$

$$A_j = \text{JET AREA} = A_p = 2.0942 \text{ FT}^2$$

$$V_j = \text{JET VELOCITY} = Q_{EQ} / A_j$$

$$T = \text{THRUST} = \rho Q (V_j - V_0)$$

$$\text{P.C.} = \text{PROPULSIVE COEF.} = \frac{T/V_0}{\text{SHP} \times 550}$$

• ESTIMATED THRUST (MOMENTUM LIMIT)

$$V_0 = \text{CRAFT SPEED}$$

$$Q_{EQ} = \text{EQUILIBRIUM FLOW RATE @ } H_{PCAV} = H_{PRQ}$$

$$A_j = \text{JET AREA} = A_p = 2.0942 \text{ FT}^2$$

$$V_j = \text{JET VELOCITY} = Q_{EQ} / A_j$$

$$T = \text{THRUST} = \rho Q (V_j - V_0)$$

$$H_{PRQ} = \text{EQUILIBRIUM HEAD RISE @ } H_{PCAV} = H_{PRQ}$$

$$\lambda_p = \text{PUMP EFFICIENCY} = .77 \quad (\text{POWER LIMIT CALC.})$$

$$\text{SHP} = \text{PUMP INPUT POWER} = \frac{\rho g H_{PRQ} Q_{EQ}}{550 \lambda_p}$$

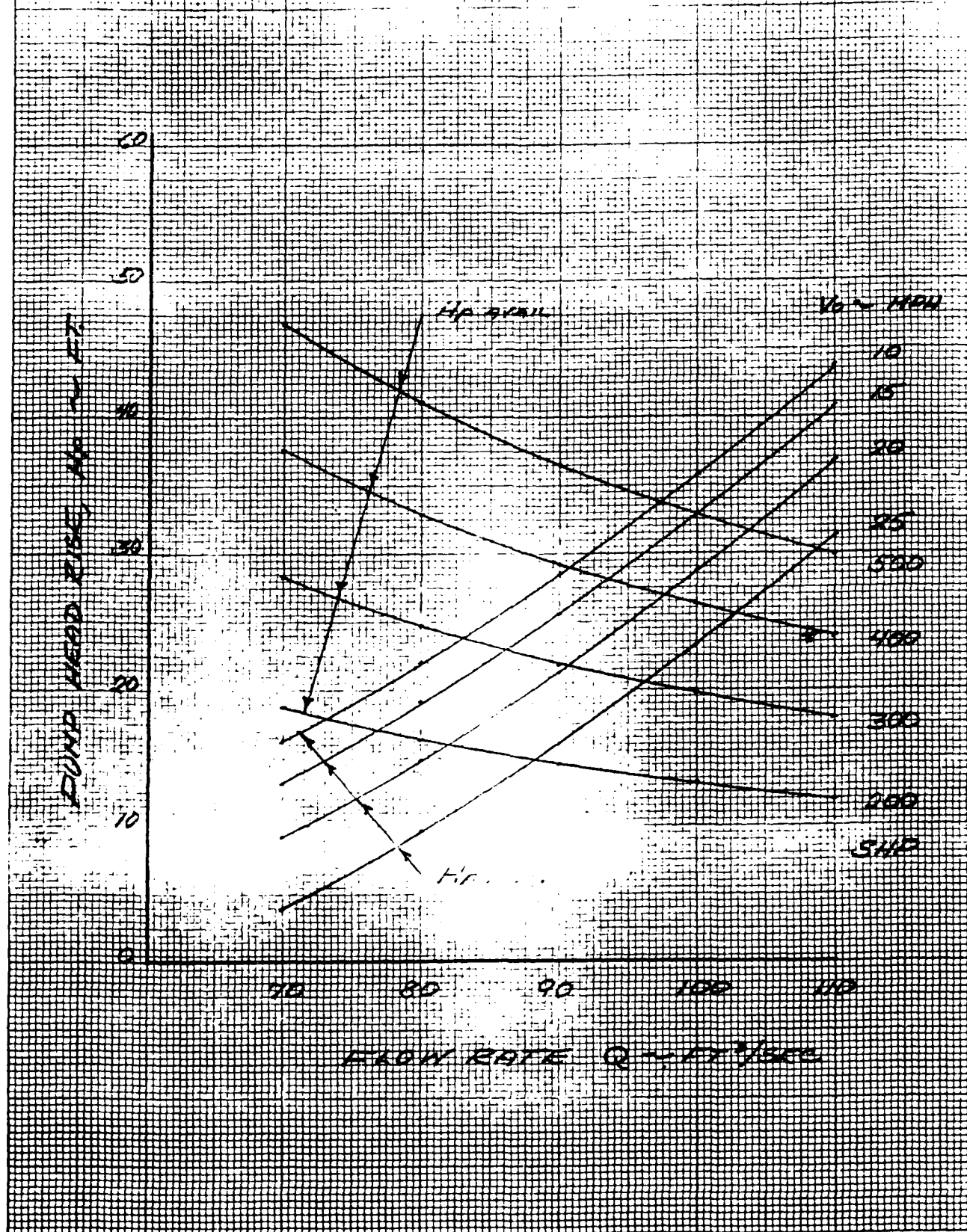
$$\text{P.C.} = \text{PROPULSIVE COEF.} = \frac{T/V_0}{\text{SHP} \times 550}$$

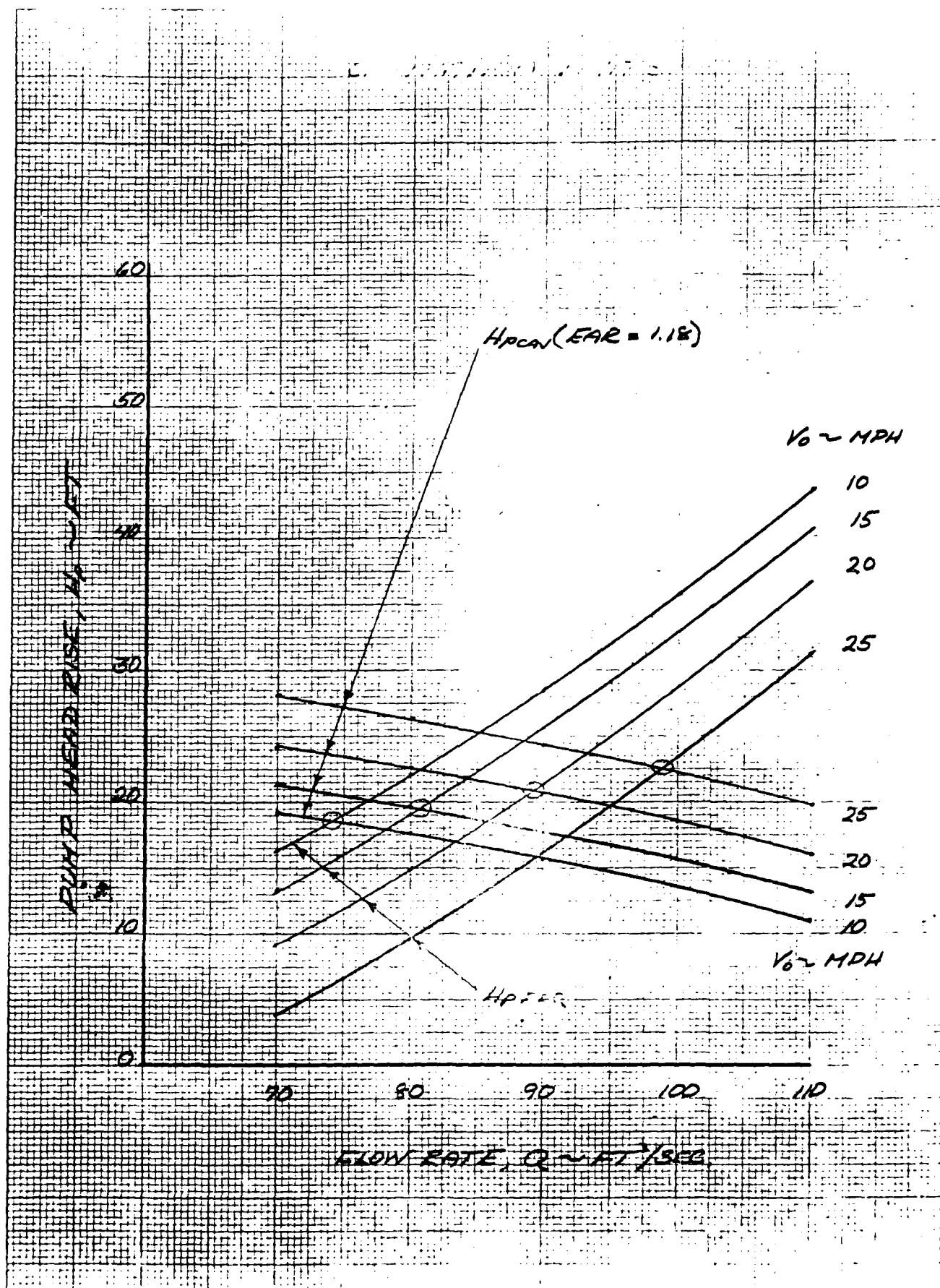
System Headcurve

CALCULATIONS

• REQUIRED PUMP HEAD

<u>V₀</u>	<u>Q</u>	<u>H_{0 REQ}</u>
14.70	70	16.31
	80	22.03
	90	28.51
	100	35.74
	110	43.75
22.05	70	13.37
	80	19.08
	90	25.56
	100	32.80
	110	40.80
29.40	70	9.25
	80	14.96
	90	21.44
	100	28.68
	110	36.68
36.75	70	3.95
	80	9.66
	90	16.14
	100	23.38
	110	31.38





SYSTEM F-105

CALCULATIONS (CONT.)

• ESTIMATED THRUST (POWER LIMIT)

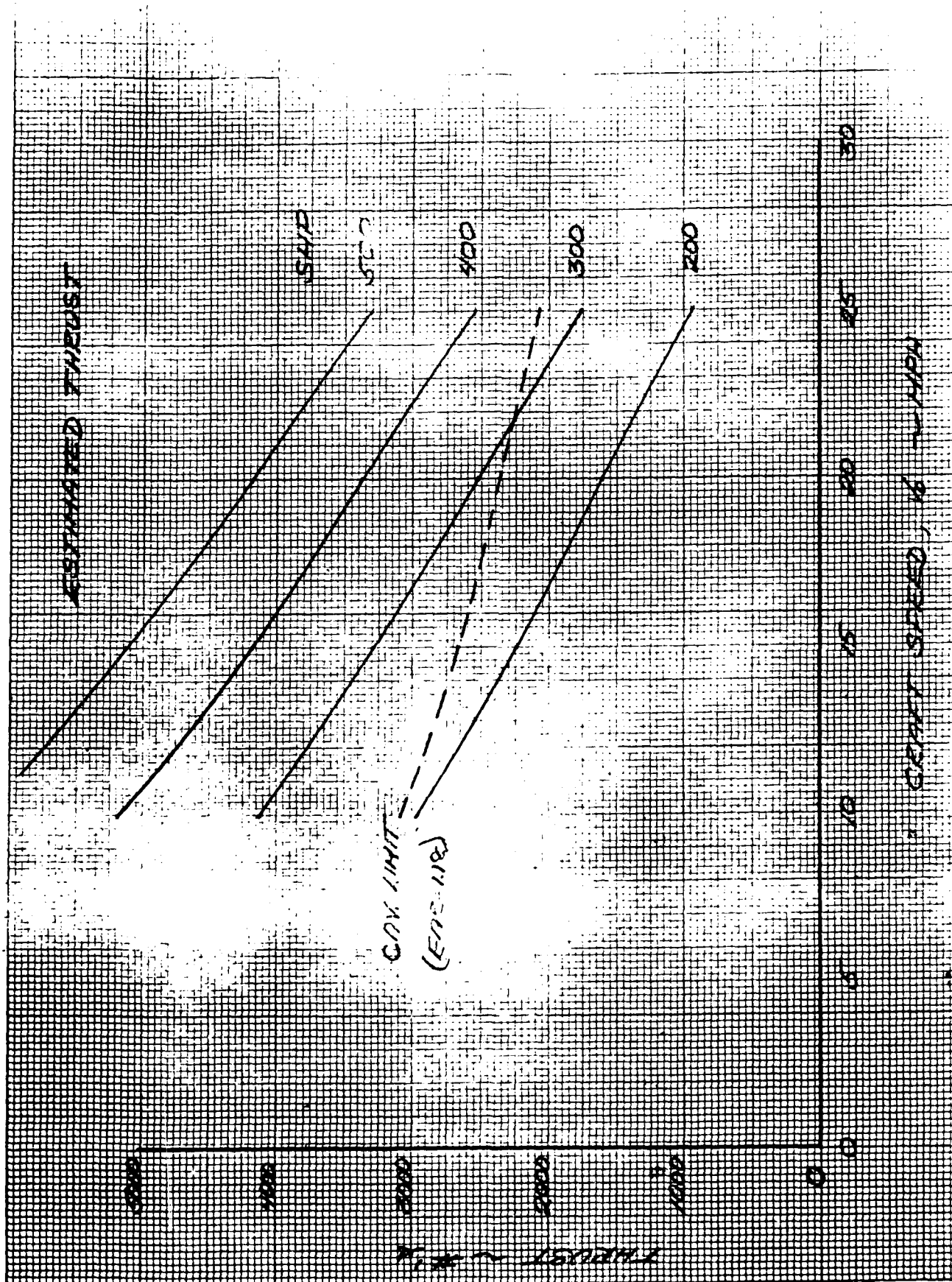
<u>V₀</u>	<u>SHF</u>	<u>Q₀</u>	<u>A₀</u>	<u>V₀</u>	<u>T</u>	<u>PC</u>
14.70	500	97.4	2.0942	46.51	6197	.3313
	400	90.7		43.31	5190	.3468
	300	83.0		39.63	4138	.3687
	200	73.2		34.95	2965	.3962
22.05	500	100.0		47.75	5140	.4121
	400	93.2		44.50	4185	.4195
	300	86.2		41.16	3295	.4403
	200	76.8		36.67	2246	.4502
29.40	500	103.8		49.57	4187	.4476
	400	97.7		46.65	3371	.4505
	300	90.7		43.31	2523	.4496
	200	82.0		39.16	1601	.4279
36.75	500	108.8		51.95	3308	.4421
	400	102.9		49.14	2550	.4260
	300	96.2		45.94	1768	.3938
	200	88.2		42.12	947	.3164

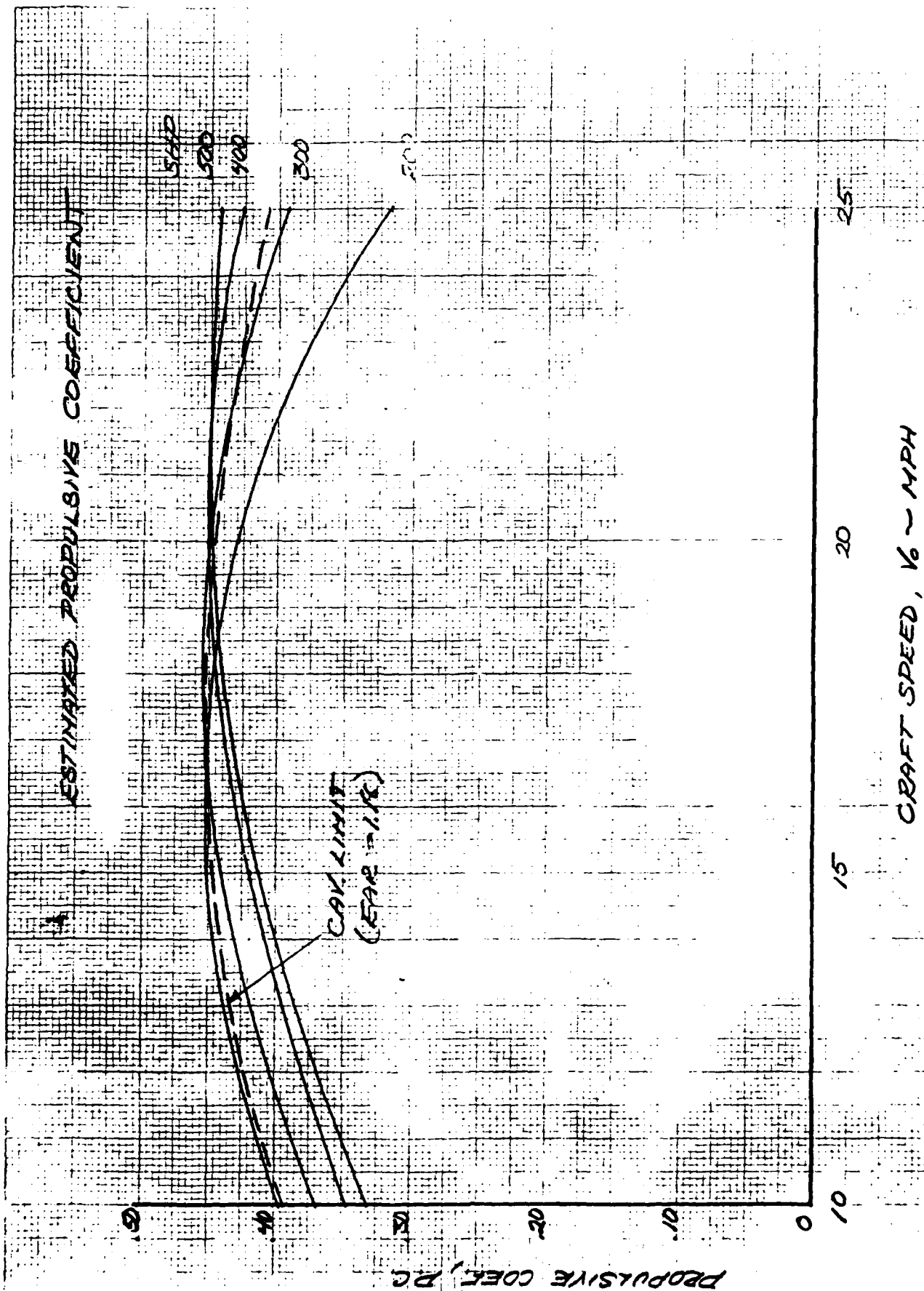
SYSTEM F-1

CALCULATIONS (CONT.)

• ESTIMATED THRUST (CALCULATED) (11117)

<u>V₀</u>	<u>Q_{eq}</u>	<u>A₀</u>	<u>V₀</u>	<u>T</u>	<u>H_{req}</u>	<u>λ_p</u>	<u>SH_P</u>	<u>PC</u>
14.70	74.3	2.0742	35.48	3088	18.6	.77	210	.3930
22.05	80.7		39.54	2661	19.5		239	.4464
29.40	89.3		42.64	2365	20.9		284	.4451
36.75	98.8		47.18	2061	22.5		338	.4074





Static Operation

REQUIRED PUMP HEAD

$$H_{PR1Q} = .00221Q^2$$

Q H_{PR1Q}

50 9.52
60 13.72
70 18.67

CAVITATION LIMIT (SEE CAVITATION LIMIT CALCS.)

Q H_{PR1Q}

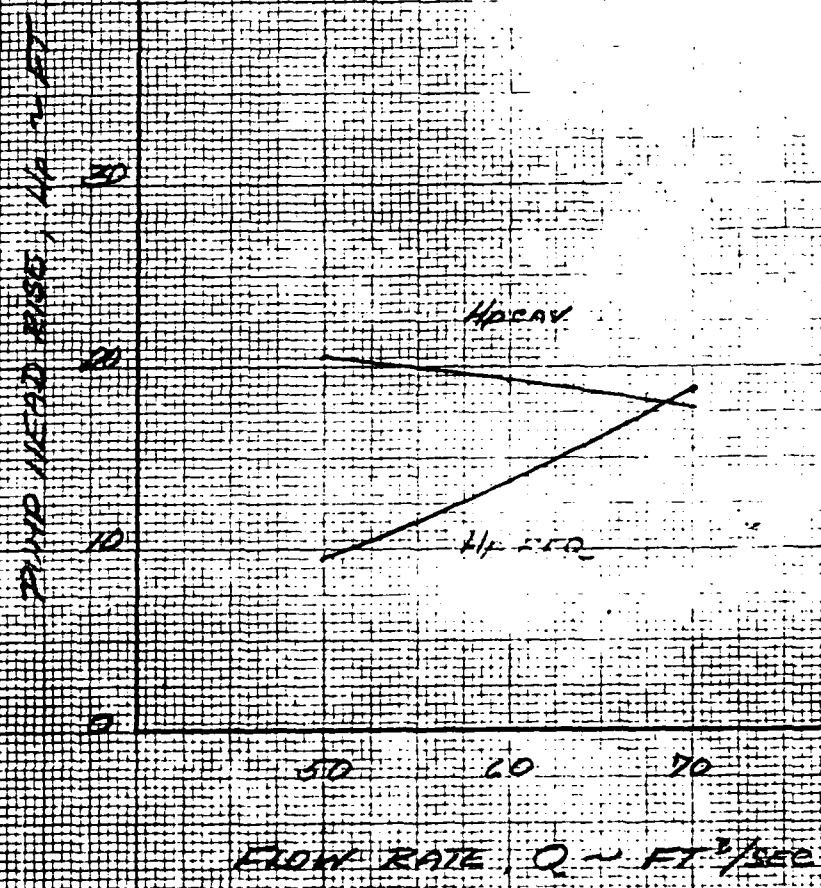
50 20.40
60 19.24
70 17.73

ESTIMATED THRUST

<u>Q_{REQ}</u>	<u>A₁</u>	<u>V₁</u>	<u>T_{TOTAL}</u>	<u>H_{PR1Q}</u>	<u>H₂</u>	<u>H_{L1}</u>	<u>Q₁</u>	<u>V₁</u>	<u>$\frac{V_1^2}{2g}$</u>	<u>H_{S3}</u>	<u>P_{S3}</u>
68.7	2.0942	32.80	4507	33.05	1	1.10	3.333	20.61	6.60	26.38	1699

USE 70 DATA FOR
EXCESS PRESSURE (1.10)

SECTION 1
CAVITATION LIMIT



ILLUSTRATIONS

REF. 1 TABLE

VON LAMEREN	FIG. 1	} MODEL TEST OF PUMP IN AXIAL CYL.
"	FIG. 2	
VACUZZI	FIG. 3	MODEL TEST IN INLET

DERIVATIONS

- #1 ESTIMATED INLET & CASING LOSSES
- #2 REQUIRED PUMP HEAD RISE
- #3 PROPELLER BLADE AREA

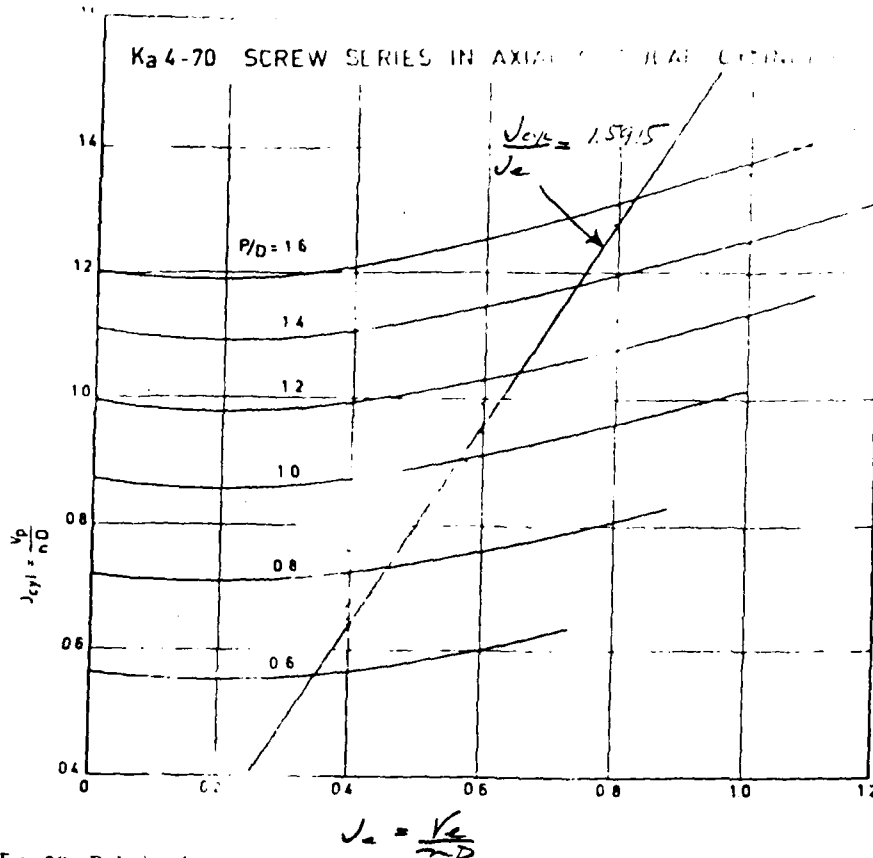


Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

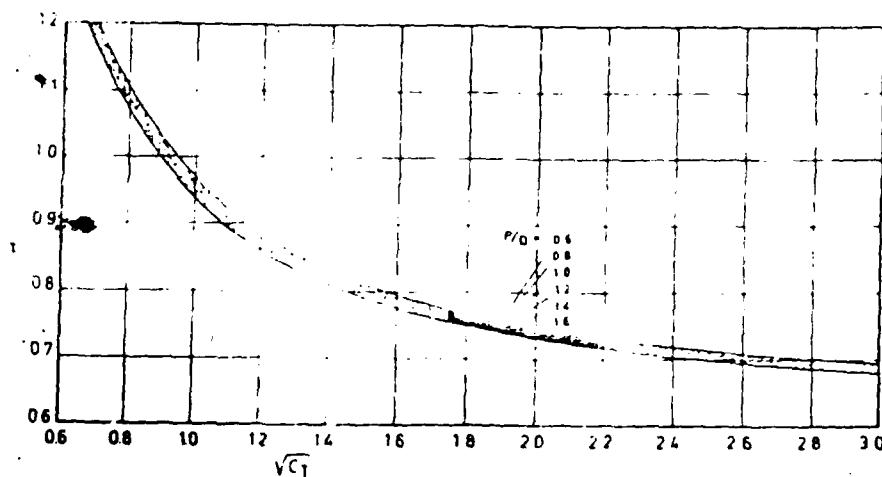


Fig. 30 Relation between thrust coefficient C_T and thrust ratio τ of nozzle no. 19a

figure have been obtained by substituting nozzles with different length-diameter ratios by systems of annular vortices and calculating the induced velocities in the screw disk.

If the radial displacement of the streamlines is small, we can consider the streamlines as lying approximately on cylindrical planes. If internal friction and turbulence are neglected, the radial

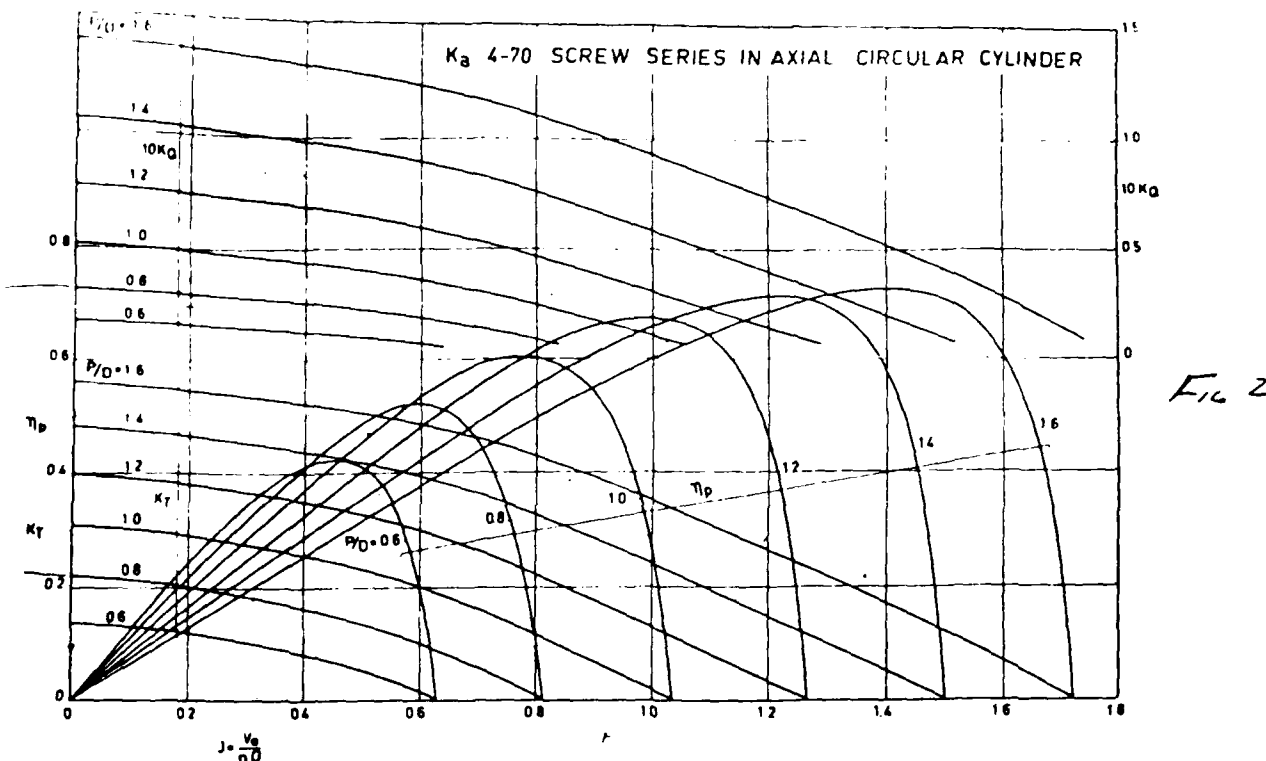


Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

been obtained from the experiments with the Ka 4-70 screw series in an axial circular cylinder and from the application of the momentum theorem.

From the comparison of the axial velocities obtained with these methods, we see that

1 The velocities agree reasonably well at high loadings of the ducted propeller system ($C_T > 1$).

2 The difference between the axial velocities becomes very large at low loadings ($C_T < 1$). In regard to the second conclusion, the following remark may be made. From Fig. 13 it can be seen that the nozzle drag due to friction becomes substantial at low loadings of the ducted-propeller system. Then, it is no longer permitted to neglect the effect of friction on the force action between nozzle and fluid.

The design of a screw in a nozzle may now be carried out as follows:

With given thrust T or power P , intake velocity V_0 , and number of revolutions n , the B_p and consequently the optimum diameter coefficient D can be determined with the aid of open-water test results of the nozzle considered, in combination with a systematic screw series (see, for instance, Fig. 24). In addition, the thrust coefficient C_T and the propeller thrust-total thrust ratio τ can be determined. With the aid of the experiments

of the systematic screw series in the axial circular cylinder or using the momentum theorem, the axial velocity V_p in the way of the screw can be found. In addition, the mean axial velocity in the vicinity of the screw due to the nozzle action, U_N , and due to the screw action U_p , can be calculated.

The pressure difference created by the screw becomes

$$\Delta p = \frac{T_p}{\frac{\pi}{4} (D^2 - d_h^2)}$$

In order to avoid an excessive loading of the inner radii of the screw blades, the usual assumption for axial pumps that the head is constant for all radii is abandoned. The following radial $\Delta p(r/R)$ distribution is suggested for the screws in nozzle no. 19a:

$$\Delta p(r/R) = [4.88 - 4r/R] \cdot [r/R - 0.133] \Delta p$$

The radial distribution of the axial and tangential velocities at the screw may be approximated as follows:

A reasonable radial distribution of the axial velocities due to the nozzle action can be determined from Fig. 32. The results given in this

COMPARISON OF RECTANGULAR AND ELLIPTICAL INLET RAM RECOVERY VARIATIONS WITH INLET VELOCITY RATIO

(Laboratory Water Channel Test of 2-inch Eye Diameter Waterjet Inlet Models)

SYM	CONFIGURATION
□	RECTANGULAR
○	ELLIPTICAL - 0.3 IN. AFT LIP RADIUS

----- Estimated Performance of Jacuzzi Inlet Configuration

$$\eta_i = 1 - \frac{(P_{T\infty} - \bar{P}_i)}{q_\infty}$$

◆ Measured Performance of 28HJ Inlet

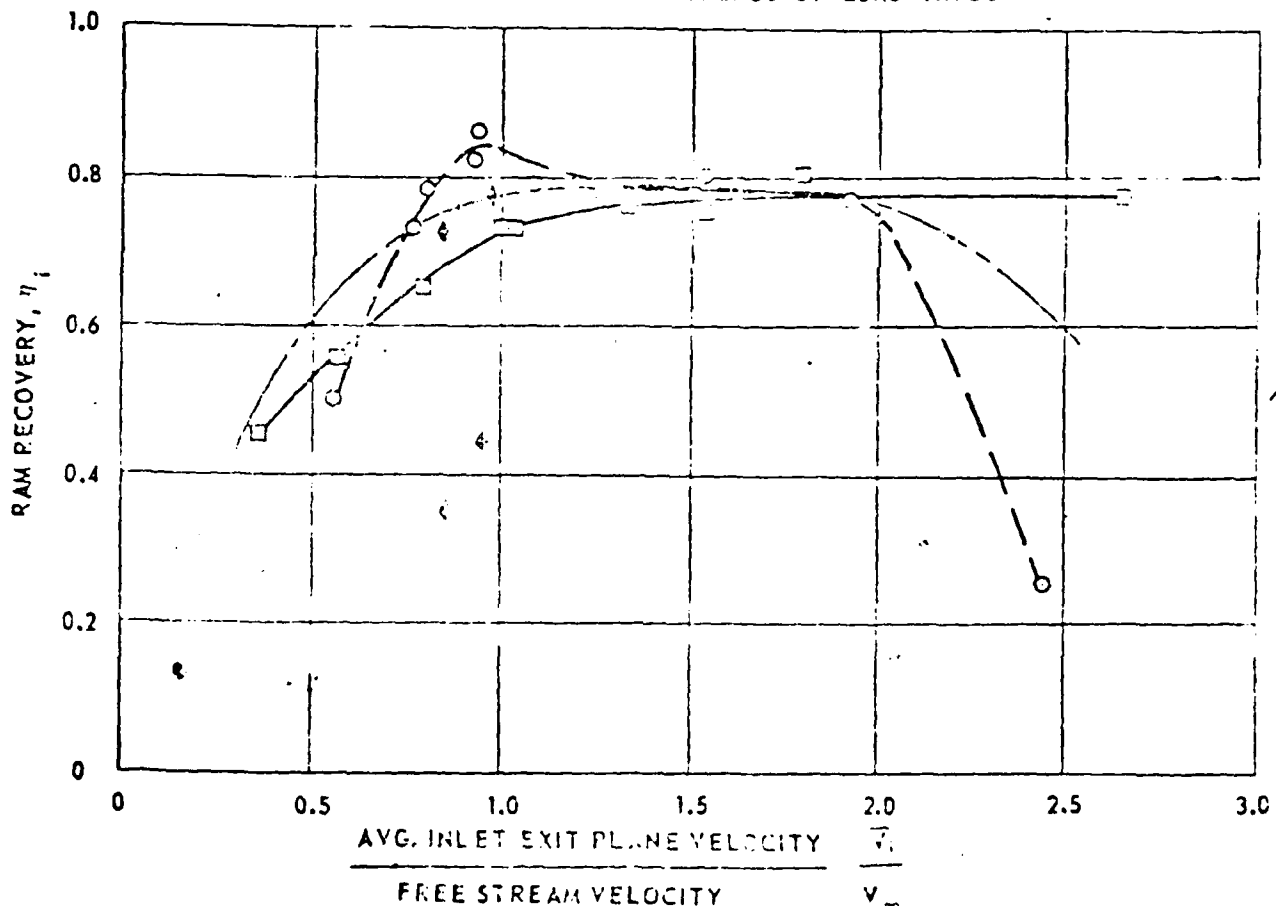


FIG. 3

B-22 Jacuzzi

INTAKE FRICION & BED

$$Q = \text{NOMINAL FLOW RATE} = 100 \text{ FT}^3/\text{SEC}$$

$$A_I = \text{INLET AREA} = (1.667)(7.00) = 3.333 \text{ FT}^2$$

$$V_I = \text{INLET VELOCITY} = Q/A_I = 100/3.333 = 30 \text{ FT/SEC}$$

$$d_e = \text{EQUIV. INLET DIAM.} = \frac{4A_I}{2(1.667) + 7.00} = \frac{4(3.333)}{2(1.667) + 7.00} = 1.8182'$$

$$Re = \frac{V_I d_e}{\mu} = \frac{(30)(1.8182)}{1.24 \times 10^{-5}} = 4.40 \times 10^6$$

$$e/d_e = \text{RELATIVE ROUGHNESS} = .000005/1.8182 = .00000275$$

$$f = \text{FRICTION FACTOR} = .00925$$

$$L_I = \text{INTAKE LENGTH} = \sqrt{21^2 + 12^2} = 17.23' = 1.44'$$

$$L_B = \text{EQUIV. LENGTH OF BEND} = \left(\frac{L}{D}\right)_B (d_e) \left(\frac{D}{90}\right) = (36)(1.8182) \left(\frac{35}{90}\right) = 25.45'$$

$$L = \text{TOTAL EQUIV. LENGTH} = L_I + L_B = 1.44 + 25.45 = 26.89'$$

$$HL = f \left(\frac{L}{d_e}\right) \left(\frac{V_I^2}{2g}\right) = (.00925) \left(\frac{26.89}{1.8182}\right) \frac{(30)^2}{2(32.2)} = 1.9118'$$

SHAFT

$$C_D = \text{SHAFT DRAG COEFF.} = 1.1 \sin^3\left(\frac{\alpha}{2}\right) = 1.1 \sin^3\left(\frac{35}{2}\right) = .0299$$

$$V_I = 30 \text{ FT/SEC}$$

$$L = \text{SHAFT LENGTH} = 2.667'$$

$$d = \text{SHAFT DIAM.} = 2'' = .1667'$$

$$D = \text{SHAFT DRAG} = C_D A_I V_I^2 L d = (.0299) \left(\frac{3}{2}\right) (30)^2 (2.667) (.1667) = 11.96'$$

$$HL = \frac{D}{\rho g A_I} = \frac{11.96}{(2)(32.2)(3.333)} = .0557'$$

(22. 14711)

TRANSITION

$$A_T = \text{CROSS SECTION AREA} = \frac{3.333 + 2.0942}{2} = 2.7138 \text{ ft}^2$$

$$V_T = \text{VELOCITY} = Q/A_T = 100/2.7138 = 36.85 \text{ FT/SEC}$$

$$d_e = \text{EQUIV. DIAM.} = \sqrt{\frac{2.7138}{.785}} = 1.8593'$$

$$Re = \frac{V_T d_e}{\mu} = \frac{(36.85)(1.8593)}{1.24 \times 10^{-5}} = 5.53 \times 10^6$$

$$e/d_e = \text{RELATIVE ROUGHNESS} = .000005/1.8593 = .00000269$$

$$f = \text{FRICTION FACTOR} = .009$$

$$L_T = \text{TRANSITION LENGTH} = 10'' = .833'$$

$$HL = f \left(\frac{L}{d_e} \right) \left(\frac{V_T^2}{2g} \right) = (.009) \left(\frac{.833}{1.8593} \right) \frac{(36.85)^2}{2(32.2)} = .0850'$$

BEARING TUBE

$$A_P = \text{CROSS SECTION AREA} = 2.0942 \text{ ft}^2$$

$$V_P = \text{VELOCITY} = Q/A_P = 100/2.0942 = 47.75 \text{ FT/SEC}$$

$$L = \text{PIPE LENGTH} = 1.66'$$

$$Re = \frac{V_P L}{\mu} = \frac{(47.75)(1.66)}{1.24 \times 10^{-5}} = 6.42 \times 10^6$$

$$C_f = .00315$$

$$S = \text{PIPE SURFACE AREA} = (1.66) \pi (.375) = 1.94 \text{ ft}^2$$

$$D = \text{TUBE DRAG} = (C_f + .0005)(S) \left(\frac{V_P^2}{2} \right) = (.00315 + .0005)(1.94) \left(\frac{47.75^2}{2} \right) = 15.62'$$

$$HL = \frac{D}{\rho g A} = \frac{15.62}{2(32.2)(2.0942)} = .1162'$$

Calculation

$$A_p = \text{CROSS SECTION AREA} = 2.0942 \text{ FT}^2$$

$$V_p = \text{VELOCITY} = Q/A_p = 100/2.0942 = 47.75 \text{ FT/SEC}$$

$$C = \text{STRUT CHORD} = 4'' = .333'$$

$$Re = \frac{V_p C}{\nu} = \frac{(47.75)(.333)}{1.24 \times 10^{-5}} = 1.28 \times 10^6$$

$$C_f = .00419$$

$$t/c = \text{STRUT THICKNESS RATIO} = .375/4 = .0938$$

$$C_D = 2(C_f + .0008)(1 + 1.2 t/c) = 2(.00419 + .0008)(1 + 1.2 \times .0938) = .0111$$

$$S = \text{STRUT PLANFORM AREA} = 2(1.667 - .333)(.333) = .8884 \text{ FT}^2$$

$$D = \text{STRUT DRAG} = C_D S A_p V_p^2 = (.0111)(.8884)(3/2)(47.75)^2 = 22.48^{\#}$$

$$HL = \frac{D}{Q} = \frac{22.48}{100} = .2248$$

TOTAL HEAD LOSS

INTAKE FRICTION + BEND	1.9118
ORIFICE	.0557
TRANSITION	.0850
BEARING LOSS	.1162
STRUT DRAG	.2248

$$H_{L_T} = 2.3354'$$

$$k = \frac{H_{L_T}}{Q_{nom}^2} = \frac{2.3354}{(100)^2} = .000234$$

$$H_{L_T} = .000234 Q^2$$

Example 12.12 (continued)

Casing

$$Q = \text{NOMINAL FLOW RATE} = 100 \text{ FT}^3/\text{SEC}$$

$$A_p = 2.0942 \text{ FT}^2$$

$$V_p = Q/A_p = 100/2.0942 = 47.75 \text{ FT/SEC}$$

$$d = \text{CASING DIAM.} = 1.667'$$

$$Re = \frac{V_p d}{\nu} = \frac{(47.75)(1.667)}{1.214 \times 10^{-5}} = 6.42 \times 10^6$$

$$e/d = \text{RELATIVE ROUGHNESS} = .000005/1.667 = .000003$$

$$f = \text{FRICTION FACTOR} = .0088$$

$$L_c = \text{CASING LENGTH} = 1.667'$$

$$HL = f \left(\frac{L_c}{\pi} \right) \left(\frac{V_p^2}{2g} \right) = (.0088) \left(\frac{1.667}{\pi} \right) \frac{(47.75)^2}{2(32.2)} = .3116'$$

$$k = \frac{HL_c}{Q_{avg}^2} = \frac{(.3116)}{(100)^2} = .0000312$$

$$HL_c = .0000312 Q^2$$

$$H_{f, \text{net}} = H_{L_1} + H_{L_2} + H_{L_3} - H_0$$

$$H_{L_1} = \frac{V_1^2}{2g}$$

$$V_1 = Q/A_1$$

$$A_1 = A_2 = 2.0942 \text{ ft}^2$$

$$H_{L_1} = \frac{Q^2}{(2.0942)^2 (2)(32.2)} = .00354 Q^2$$

$$H_{L_2} = .000234 Q^2$$

$$H_{L_3} = .0000312 Q^2$$

} DERIVATION #1

$$H_0 = (RPR) \frac{V_0^2}{2g}$$

$$RPR = .70$$

$$H_0 = \frac{(.70) V_0^2}{2(32.2)} = .0109 V_0^2$$

$$\begin{aligned} H_{f, \text{net}} &= .00354 Q^2 + .000234 Q^2 + .0000312 Q^2 - .0109 V_0^2 \\ &= .00381 Q^2 - .0109 V_0^2 \end{aligned}$$

Expanded Area

<u>n</u>	<u>C</u>	<u>T.H.</u>	<u>f(A_x)</u>
2	10.70	1/2	5.35
3	12.25	1	12.25
4	13.70	1	13.70
5	14.94	1	14.94
6	16.00	1	16.00
7	16.90	1	16.90
8	17.50	1	17.50
9	17.85	1	17.55
10	18.00	1/2	9.00
			<u>123.49</u>

$$A_x = (3)(1)(123.49) = 370.47 \text{ in}^2$$

$$A_0 = (.785)(20)^2 = 314 \text{ in}^2$$

$$E.A.R. = \frac{370.47}{314} = 1.18$$

Projected Area

<u>n</u>	<u>C</u>	<u>α</u>	<u>P</u>	<u>T.H.</u>	<u>f(A_x)</u>
2	10.70	57.86	5.69	1/2	2.84
3	12.25	46.70	8.40	1	8.40
4	13.70	38.51	10.72	1	10.72
5	14.94	32.18	12.60	1	12.60
6	16.00	27.95	14.13	1	14.13
7	16.90	24.45	15.35	1	15.35
8	17.50	21.70	16.26	1	16.26
9	17.85	19.48	16.83	1	16.83
10	18.00	17.66	17.15	1/2	8.58
					<u>105.74</u>

$$\begin{aligned} \alpha &= \arctan\left(\frac{P}{C}\right) / r \\ &= \arctan\left(\frac{20}{2.7}\right) / r \\ &= \arctan 3.1831 / r \end{aligned}$$

$$A_p = (3)(1)(105.74) = 317.22 \text{ in}^2$$

$$A_0 = 314 \text{ in}^2$$

$$P.A.R. = \frac{317.22}{314} = 1.01$$

STRUCTURAL ANALYSIS

- PROPELLER
- SHAFT
- CASING

PROPELLER STRESS ANALYSIS

$$R = \text{PROP. RADIUS} = 10''$$

$$Z = \text{NO. OF BLADES} = 3$$

$$D = \text{PITCH} = 20''$$

$$T_p = \text{PROP. THRUST} = \rho g H_p A_p = (64.4)(20.9)(2.0942) = 2819''$$

$$N = \text{PROP. SPEED} = \frac{60 Q_{12}}{A_{\text{dev}} D} = \frac{(60)(893)}{(2.0942)(.905)(1.667)} = 1696 \text{ RPM}$$

$$Q' = \text{PROP TORQUE} = 63024 \frac{\text{SHP}}{N} = \frac{(63024)(264)}{1696} = 10554 \text{ IN}^2$$

20 MPH
Cav. Limit

$$a = \frac{2\pi R}{P} = \frac{2\pi \cdot 10}{20} = 3.1416$$

$$x = r/R$$

$$K = f(x)$$

TABLE I

CONALLY

$$A_1 = f(a, x)$$

" I

"

$$A_2 = f(a, x)$$

" II

"

$$B_1 = f(a, x)$$

III

"

$$B_2 = f(a, x)$$

IV

"

$$C_1 = f(a, x)$$

"

$$C_2 = f(a, x)$$

"

$$f = \text{MAX. SHEAR STRESS (FACE)}$$

$$\sigma_r = \text{SPANWISE BENDING STRESS}$$

$$= \frac{RK}{Ect^2} \left[A_1 \left(\frac{2\pi RT}{P} \right) + A_2 \left(\frac{Q'}{R} \right) \right]$$

$$\sigma_\theta = \text{CHORDWISE BENDING STRESS}$$

$$= \frac{RK}{Ect^2} \left[B_1 \left(\frac{2\pi RT}{P} \right) + B_2 \left(\frac{Q'}{R} \right) \right]$$

$$\sigma_c = \text{CENTRIFUGAL STRESS} = \frac{2240 N^2 R^2 C}{10^6}$$

$$\sigma_{T \text{ MAX}} = \text{MAX. TENSION STRESS (FACE)} = \sigma_r + \sigma_c$$

$$\sigma_{S \text{ MAX}} = \text{MAX. SHEAR STRESS (FACE)} = \frac{\sigma_r - \sigma_\theta}{2}$$

PROPELLER STRESS CALCULATION

BENDING STRESSES

R	Z	P	T	Q'	a	X	K	A ₁	A ₂	C	T	σ _R	B ₁	B ₂	σ
10.00	3	20.00	2819	10551	2.1116	.20	.1203	6.50	57.95	10.20	.125	8654	2226	1930	2820
						.30	.1207	6.94	39.26	12.25	.638	8303	3.13	1619	3820
						.40	.1007	7.27	31.20	13.20	.550	7882	3.97	1925	4495
						.50	.0720	7.66	27.63	14.94	.463	7417	4.79	1998	5750
						.60	.0437	8.24	22.00	16.00	.325	6856	5.63	2093	7150
						.70	.0214	10.00	22.95	16.90	.288	6008	6.58	2220	8450
						.80	.0075	11.61	30.10	17.50	.201	4759	7.65	2360	9750
						.90	.0013	12.53	30.32	17.85	.113	2718	8.44	2474	11100

COMBINED STRESSES

R	N	X	C ₁	σ _C	X	σ _R	σ _B	σ _C	A _{Tmax}	A _{max}
10.00	1696	.20	16.6	1070	.20	8654	2879	1070	9724	3422
		.30	12.0	773	.30	8303	3828	773	9076	2939
		.40	9.5	612	.40	7882	4493	612	8494	2000
		.50	7.7	496	.50	7417	4759	496	7913	1577
		.60	6.2	399	.60	6856	4658	399	7255	1298
		.70	4.8	309	.70	6008	4158	309	6317	1080
		.80	3.4	219	.80	4759	3264	219	4978	847
		.90	1.9	122	.90	2718	1917	122	2840	422

PROBLEM 5. CONT.

TORSION STRESS

$$Q' = \text{TORSION MOM.} = 10534 \text{ IN}^2$$

$$d = \text{SHAFT DIAM} = 2", \quad r = 1.00"$$

$$J = \text{POLAR MOM. OF INERTIA} = \left(\frac{\pi}{2}\right) r^4 = \left(\frac{\pi}{2}\right) (1)^4 = 1.5708$$

$$s_s = \text{TORSIONAL STRESS} = \frac{Q' r}{J} = \frac{(10534)(1)}{1.5708} = 6719 \text{ psi}$$

$$\text{FACTOR OF SAFETY} = \frac{20000}{6719} = 2.98 \text{ ON SHEAR YIELD (6061-T6)}$$

WHIRLING FREQUENCY

$$W = \text{WEIGHT PER UNIT LENGTH} = (.785)(2)^2(1)(.098) = .3014 \text{ #/IN}$$

$$L = \text{DISTANCE BETWEEN SUPPORTS} = 36"$$

$$I = \text{MOM. OF INERTIA} = .049 d^4 = (.049)(2)^4 = .784 \text{ IN}^4$$

$$D = \text{STATIC DEF. DUE TO OWN WT.}$$

$$= .00542 \frac{W L^4}{EI} = \frac{(.00542)(.3014)(36)^4}{(10,200,000)(.784)} = .000343" \quad \text{FREE-FIXED}$$

$$f = \text{WHIRLING FREQ.} = \frac{3.57}{2 \sqrt{D}} = \frac{3.57}{2 \sqrt{.000343}} = 192 \text{ CPS.}$$

$$= 11,499 \text{ RPM}$$

$$N_{DES} = 169,000$$

INLET CASING DESIGN

INLET CASING DESIGN PRESSURE

$$\begin{aligned} P &= \text{DESIGN PRESSURE} \\ &= \text{EXTERNAL PRESSURE} - \text{INTERNAL PRESSURE} \\ &= (H_{ATM} + H_L - H_{IS}) \frac{64}{144} \\ &= (33.08 + 1 - 26.38) \left(\frac{64}{144} \right) = 3.42 \text{ psi} \end{aligned}$$

NOTE: MINIMUM H_{IS} OCCURS DURING
STATIC OPERATION AT CAV. LIMIT

INLET CASING STRESS

$$\begin{aligned} P &= \text{DESIGN PRESSURE} = 3.42 \text{ psi} \\ L &= \text{SPAN} = 24" \\ W &= \text{UNIT WIDTH} = 1" \\ M &= \text{BENDING MOM. IN PLATE} = \frac{P \cdot L^2 \cdot W}{12} = \frac{(3.42)(24)^2(1)}{12} = 164.16 \text{ in}^2 \\ t &= \text{PLATE THICKNESS} = .375" \\ Z &= \text{SECTION MODULUS OF PLATE} = \frac{W \cdot t^2}{6} = \frac{(1)(.375)^2}{6} = .0234 \text{ in}^3/\text{in width} \\ S &= \text{BENDING STRESS IN PLATE} = \frac{M}{Z} = \frac{164.16}{.0234} = 7015 \text{ psi} \end{aligned}$$

ESTIMATED WEIGHTS

INLET CASING

$$A = \left[\frac{2(34.5)(24)}{2} + (20)(34.5) + 2 \left(\frac{35}{360} \right) (365)(44)^2 + \left(\frac{35}{360} \right) \pi (48)(20) \right] \frac{1}{144} = 15.02$$

$$f = .375", \quad w = (.375)(144)(.096) = 5.18 \text{ #/ft}^2$$

$$W = (15.02)(5.18) = 77.80 \text{ #}$$

TRANSITION

$$A = \left[\frac{2(24+20)}{2} + 20 \right] \frac{10}{144} = 5.24 \text{ ft}^2$$

$$f = .375", \quad w = 5.18 \text{ #/ft}^2$$

$$W = (5.24)(5.18) = 27.14 \text{ #}$$

PROPELLER CASING

$$A = 20\pi(20)/144 = 8.73 \text{ ft}^2$$

$$f = .375", \quad w = 5.18 \text{ #/ft}^2$$

$$W = (8.73)(5.18) = 45.20 \text{ #}$$

OUTLET

$$A = 4(4)(375)(.77)(8) = 34.08 \text{ ft}^2$$

$$f = .096 \text{ #/in}^2$$

$$W = (34.08)(.096) = 3.27 \text{ #}$$

INLET TUBES

S.E.	d	a	T.H.	f(Δ)
1	4.00	12.56	1/2	6.28
2	4.00	12.56	1	12.56
3	3.50	9.62	1	9.62
4	2.25	5.94	1/2	2.97
				31.43

$$\Delta = 4(31.43) - 8(.785)(2.25)^2 - 4(.785)(2.25)^2 = 66.55 \text{ ft}^2$$

$$w = .096 \text{ #/in}^2$$

$$W = (66.55)(.096) = 6.39 \text{ #}$$

CASING TOTAL

$$W = 77.80 + 27.14 + 45.20 + 3.27 + 6.39 = 160 \text{ # CONV. CONST.}$$

$$= (160) \left(\frac{1.50}{2.66} \right) =$$

90# composite

B-35

Flow

BLADE

x	c	f	a	T.M.	$f(\nabla)$
.20	10.70	.725	5.57	$\frac{1}{2}$	2.76
.30	12.25	.638	5.35	1	5.35
.40	13.20	.550	5.35	1	5.35
.50	14.44	.463	4.91	1	4.91
.60	16.00	.375	4.26	1	4.26
.70	16.90	.288	3.46	1	3.46
.80	17.50	.201	2.50	1	2.50
.90	17.85	.113	1.43	1	1.43
1.00	18.00	.060	1.02	$\frac{1}{2}$	.51
					30.73

$$\nabla = 3(1)(30.73) = 92.19, \text{ in}^3$$

$$w = .28 \text{ in}^3/\text{in}^3$$

$$W = (92.19)(.28) = 25.81 \text{ in}^3$$

HUB

$$\nabla = (10)(.765)(4)^2 - (6)(.765)(1.75)^2 - (4)(.765)(3.25)^2 = 78.01, \text{ in}^3$$

$$w = .28 \text{ in}^3/\text{in}^3$$

$$W = (78.01)(.28) = 21.84 \text{ in}^3$$

Total

$$W = \frac{25.81}{21.84}$$

$$47.65 \text{ in}^3$$

SHAFTING WEIGHT

SHAFT

$$L = 68/12 = 5.67'$$

$$W = (.215)(2)(12)(.092) = 3.62 \text{ #/ft}$$

$$W = (5.67)(3.62) = 21 \text{ #}$$

Misc. (BEARINGS, HOUSINGS, SEALS)

$$W = 10 \text{ #}$$

TOTALS

$$W = 21 + 10 = 31 \text{ #}$$

INLET CASING

$$V = \frac{20(34.5)(24)}{2} + \left(\frac{35}{34.5}\right)(.785)(48)^2(20) = 11797 \text{ in}^3$$

TRANSITION

$$V = \left[\frac{(24)(20) + (.785)(20)^2}{2} \right] 10 = 3970 \text{ in}^3$$

PROPELLER CASING

$$V = 20(.785)(20)^2 = 6280 \text{ in}^3$$

TOTALS

$$V_{\text{tot}} = 11797 + 3970 + 6280 = 22047 \text{ in}^3 = 12.76 \text{ ft}^3$$

$$W = 64 \text{ ft}^3 \text{ min}^{-1}$$

$$V = (12.76 \text{ ft}^3) = 817 \text{ #}$$

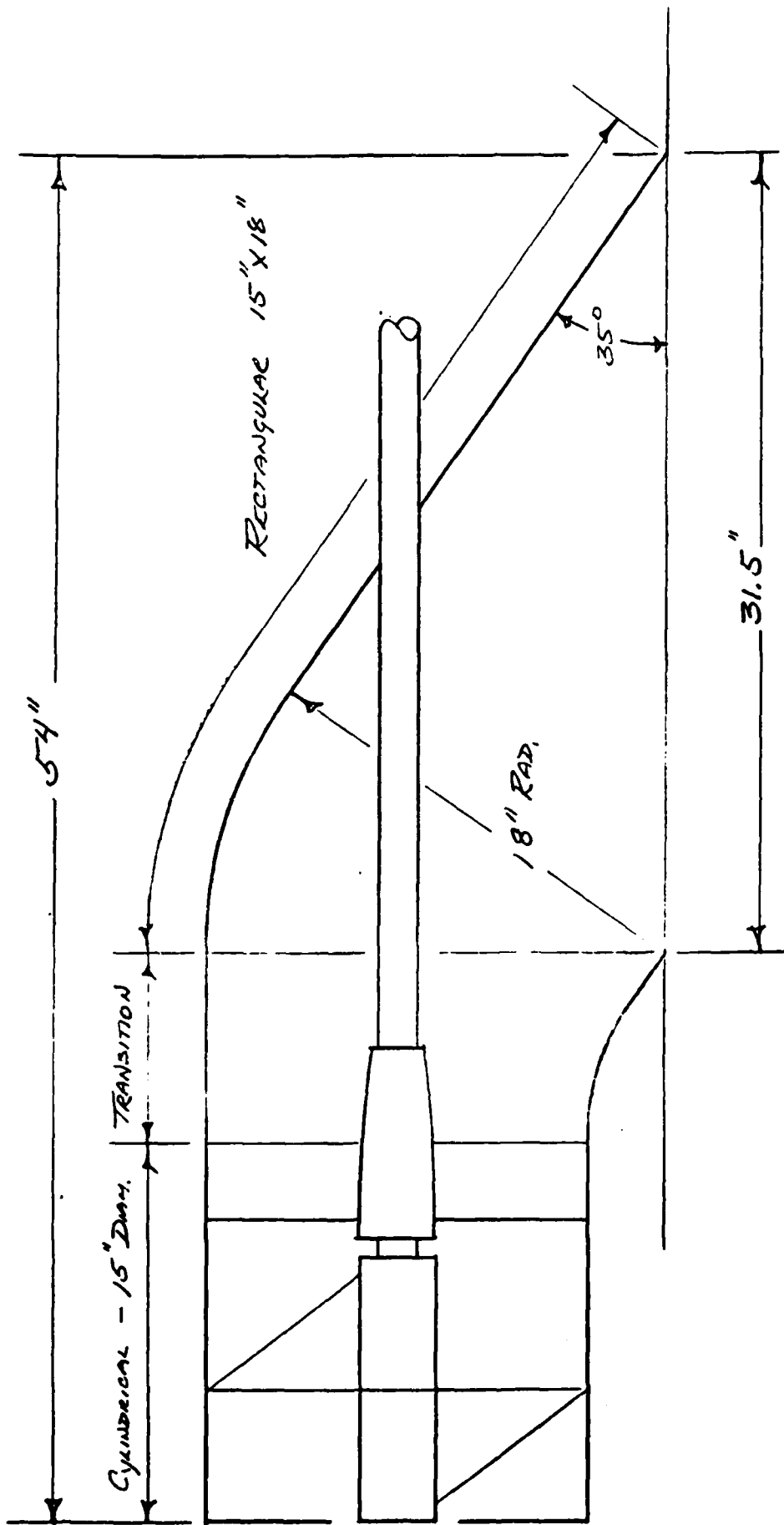
WEIGHT SUMMARY

	COMP. CONST.	COMP. CONST.
	WT	WT
CASING	90 Polyester - Glass Cloth	160 Alum. (6061-T6)
PROPELLER	48 Nuc. Ac. Fib.	
SHAFTING	31 Alum. (6061-T6)	21 Alum. (6061-T6)
Dry Wt.	169 #	239 #
Water	812	
Wet Wt.	981 #	

APPENDIX C

OBJECTIVES:

- Determine performance, weight and dimensional characteristics of a propulsion pump suitable for a multiple unit "tailgate" installation, using a 15-inch diameter impeller, in a high-speed (20 mph) amphibian.
- Use simple "propeller-in-tube" approach.
- Examine higher blade area ratios than available in existing propeller series data.



SKETCH OF 15 INCH DIAMETER PUMP

PERFORMANCE CHARACTERISTICS

- POWER LIMITS
- CAVITATION LIMITS
- SYSTEM PERFORMANCE
- STATIC OPERATION
- DERIVATIONS & REF. MATL.

POWER LIMIT

CALCULATION NOTES

$$D = \text{PROP. DIAM.} = 15' = 1.25'$$

$$A_p = \text{PROP. DISC. AREA} = .785 \left[(1.25')^2 - (.25')^2 \right] = 1.1775 \text{ ft}^2$$

$$A_E = \text{INLET AREA} = (1.50)(1.25) = 1.875 \text{ ft}^2$$

$$J_{\text{cyl}} = \text{ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE}$$

$$J_e = \text{ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE}$$

$$\frac{J_{\text{cyl}}}{J_e} = \frac{A_E}{A_p} = \frac{1.875}{1.1775} = 1.5924$$

$$P/D = \text{PROP. PITCH-DIAM. RATIO}$$

$$J_{\text{cyl}} = f(P/D, \frac{J_{\text{cyl}}}{J_e}) = .905 \quad (\text{VON KARMAN - FIG. 1})$$

$$J_e = J_{\text{cyl}} / \left(\frac{J_{\text{cyl}}}{J_e} \right) = .905 / 1.5924 = .5683$$

$$\lambda_D = \text{PROPULSION EFFICIENCY OF PROP.-TUBE COMB.} = f(J_e, P/D) \quad (\text{FIG. 2})$$

$$\lambda_{RR} = \text{PROP. RELATIVE ROTATIVE EFFICIENCY} = .95 \quad (\text{GUESS - TUNING PARAM.})$$

$$\lambda_P = \text{PUMP EFFICIENCY} = \frac{J_{\text{cyl}} \lambda_D \lambda_{RR}}{\lambda_D}$$

$$\text{SHP} = \text{PUMP SHP}$$

$$Q = \text{FLOW RATE (GPM)} = \text{FT}^3/\text{SEC}$$

$$H_{\text{PAIL}} = \text{PUMP HEAD RISE AVAILABLE} = \frac{550 \text{ SHP } \lambda_P}{\rho g Q}$$

Power Limit

Calculation

<u>D</u>	<u>A_p</u>	<u>A_E</u>	<u>$\frac{V_{avg}}{V_a}$</u>	<u>$\frac{P}{D}$</u>	<u>$\frac{V_{avg}}{V_a}$</u>	<u>$\frac{I_{avg}}{I_a}$</u>	<u>$\frac{I_{avg}}{I_a}$</u>	<u>SHP</u>	<u>Q</u>	<u>Answer</u>
1.25	1.175	1.95	1.924	1.00	.905	.5683	.51	.95	.77	500
										40 42.70
										60 57.10
										80 41.10
										400
										40 65.26
										60 43.84
										80 32.88
										300
										40 49.32
										60 32.88
										80 24.66
										200
										40 32.88
										60 21.92
										80 16.44
										100
										40 16.44
										60 10.96
										80 8.22

CAVITATION LIMIT

CALCULATION NOTES

$$D = \text{PROP. DIAM.} = 15'' = 1.25'$$

$$A_p = \text{PROP. DISC AREA} = .785 [(1.25)^2 - (.25)^2] = 1.1775 \text{ FT}^2$$

$$A_E = \text{INLET AREA} = (1.75)(1.50) = 1.875 \text{ FT}^2$$

$$V_{CYL} = \text{ADVANCE RATIO BASED ON VELOCITY INSIDE TUBE}$$

$$V_R = \text{ADVANCE RATIO BASED ON VELOCITY OUTSIDE TUBE}$$

$$\frac{V_{CYL}}{V_R} = \frac{A_E}{A_p} = \frac{1.875}{1.1775} = 1.5924$$

$$P/D = \text{PROP. PITCH-DIAM. RATIO}$$

$$V_{CYL} = f(P/D, \frac{V_{CYL}}{V_R}) \quad (\text{VAN LAMEREN - FIG. 1})$$

$$Q = \text{FLOW RATE} \sim \text{FT}^3/\text{SEC}$$

$$N = \text{PROP. SPEED} = \frac{Q}{A_p V_{CYL} D}$$

$$H_{LI} = \text{INLET HEAD LOSS} = .000766 Q^2 \quad (\text{DERIVATION \#1})$$

$$V_I = \text{INLET VELOCITY} = \frac{Q}{A_E}$$

$$V_{ITAN} = \text{TANGENTIAL VELOCITY, AT .25' FROM INLET} = \frac{V_I D}{2}$$

$$V_{IR} = \text{TOTAL VELOCITY, AT .75' FROM INLET} = \sqrt{V_I^2 + V_{ITAN}^2}$$

$$H_{ATN} = \text{HEAD DUE TO TANGENTIAL VELOCITY}$$

$$H_2 = \text{HEAD DUE TO ELEVATION}$$

$$H_V = \text{HEAD DUE TO VAP. PRESS.}$$

CAVITATION LIMIT

CALCULATION NOTI - (CONT.)

V_0 = FREE STREAM VELOCITY = CRAFT SPEED

RPR = RAM PRESS. RECOVERY RATIO = .70 (JACUZZI - FIG. 3)

H_0 = RAM HEAD RECOVERED = $(RPR) \frac{V_0^2}{2g}$

H_{IS} = INLET STATIC HEAD (ABOVE VAP. PRESS.) = $H_{atm} + H_2 - H_V + H_0 - H_{L2} - \frac{V_2^2}{2g}$

P_{IS} = INLET STATIC PRESS. (ABOVE VAP. PRESS.) = $\rho g H_{IS}$

σ_{ix} = LOCAL CAV. NO. AT I. LEAD = $\frac{P_{IS}}{\rho/2 V_{ix}^2}$

T_{cmax} = PROP. LOAD COEFF. AT CAV. LIMIT = .7 σ_{ix} (GAWN)

PAR = PROP. PROJECTED AREA RATIO

H_{pcav} = PUMP HEADRISE AT CAV. LIMIT = $\frac{(T_{cmax})(PAR)(V_{ix})^2(P_L)}{\rho g}$

Travel Notes

Introduction

[illegible]

CALCULATION NOTE

• REQUIRED PUMP HEAD RISE

$$V_0 = \text{CRAFT SPEED}$$

$$Q = \text{FLOW RATE} \sim \text{FT}^3/\text{SEC}$$

$$H_{\text{REQ}} = .0121 Q^2 - .0109 V_0^2 \quad (\text{DERIVATION \# 2})$$

• ESTIMATED THRUST (POWER LIMIT)

$$V_0 = \text{CRAFT SPEED}$$

$$\text{SHP} = \text{PUMP INPUT POWER}$$

$$Q_{\text{EQ}} = \text{EQUILIBRIUM FLOW RATE @ } H_{\text{PAVAIL}} = H_{\text{REQ.}}$$

$$A_0 = \text{JET AREA} = A_p = 1.1775 \text{ FT}^2$$

$$V_0 = \text{JET VELOCITY} = Q_{\text{EQ}}/A_0$$

$$T = \text{THRUST} = \rho Q_{\text{EQ}}(V_0 - V_0)$$

$$\text{P.C.} = \text{PROPULSIVE COEF.} = \frac{T V_0}{550 \text{ SHP}}$$

• ESTIMATED THRUST (CAVITATION LIMIT)

$$V_0 = \text{CRAFT SPEED}$$

$$Q_{\text{EQ}} = \text{EQUILIBRIUM FLOW RATE @ } H_{\text{CAV}} = H_{\text{REQ.}}$$

$$A_0 = \text{JET AREA} = A_p = 1.1775 \text{ FT}^2$$

$$V_0 = \text{JET VELOCITY} = Q_{\text{EQ}}/A_0$$

$$T = \text{THRUST} = \rho Q_{\text{EQ}}(V_0 - V_0)$$

$$H_{\text{REQ}} = \text{EQUILIBRIUM HEAD RISE @ } H_{\text{CAV}} = H_{\text{REQ.}}$$

$$\eta_p = \text{PUMP EFFICIENCY} = .77 \quad (\text{POWER LIMIT CALC.})$$

$$\text{SHP} = \text{PUMP INPUT POWER} = \frac{\rho g H_{\text{REQ}} Q_{\text{EQ}}}{550 \eta_p}$$

$$\text{P.C.} = \text{PROPULSIVE COEF.} = \frac{T V_0}{550 \text{ SHP}}$$

SYSTEM PERFORMANCE

CALCULATIONS

• REQUIRED PUMP HEAD RISE

V_0 (ft/sec)	Q	H_{pump}
14.70	40	17.00
	60	41.20
	80	75.08

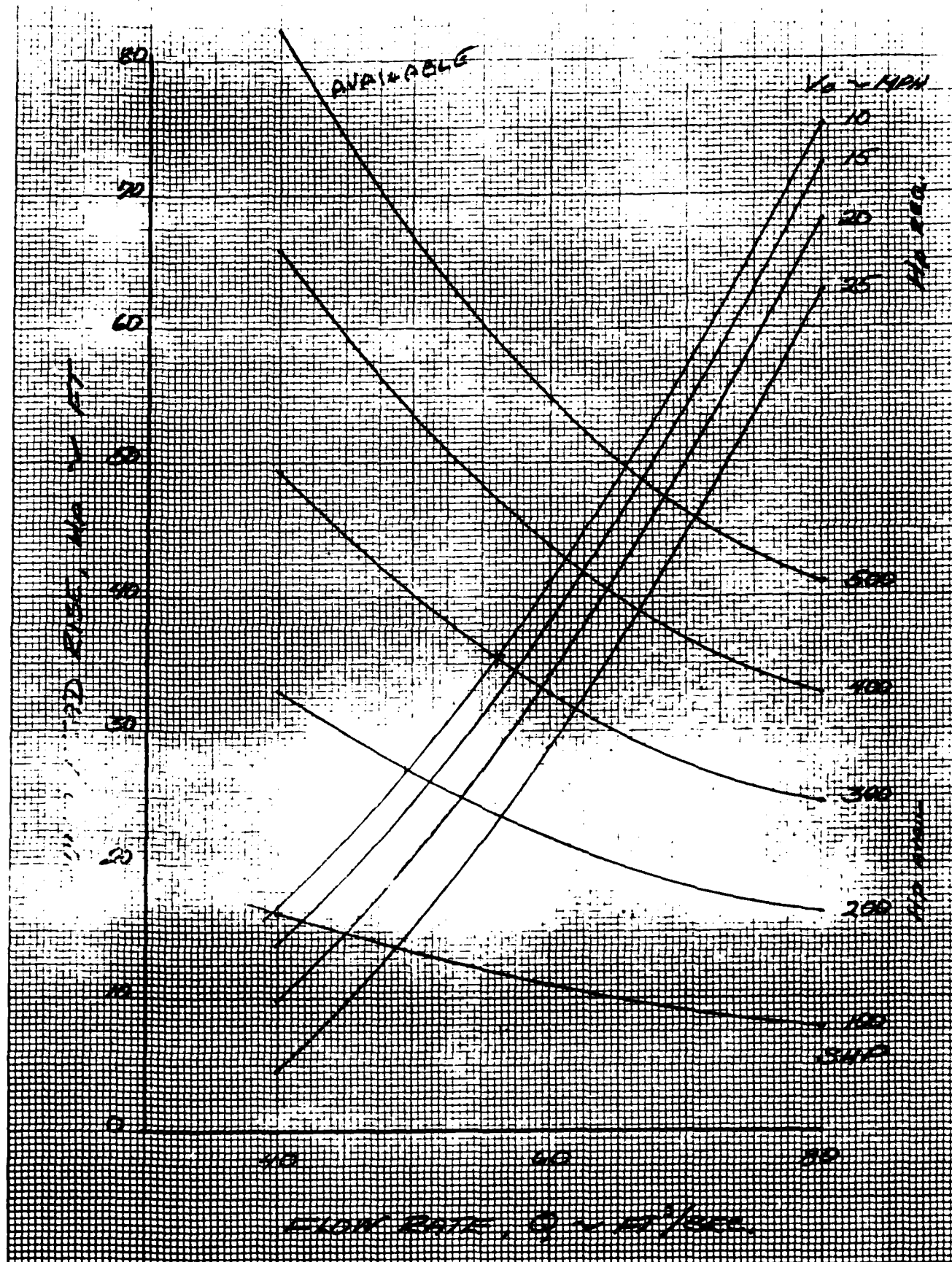
22.05
22.05
22.05

22.05	40	14.06
	60	38.26
	80	72.14

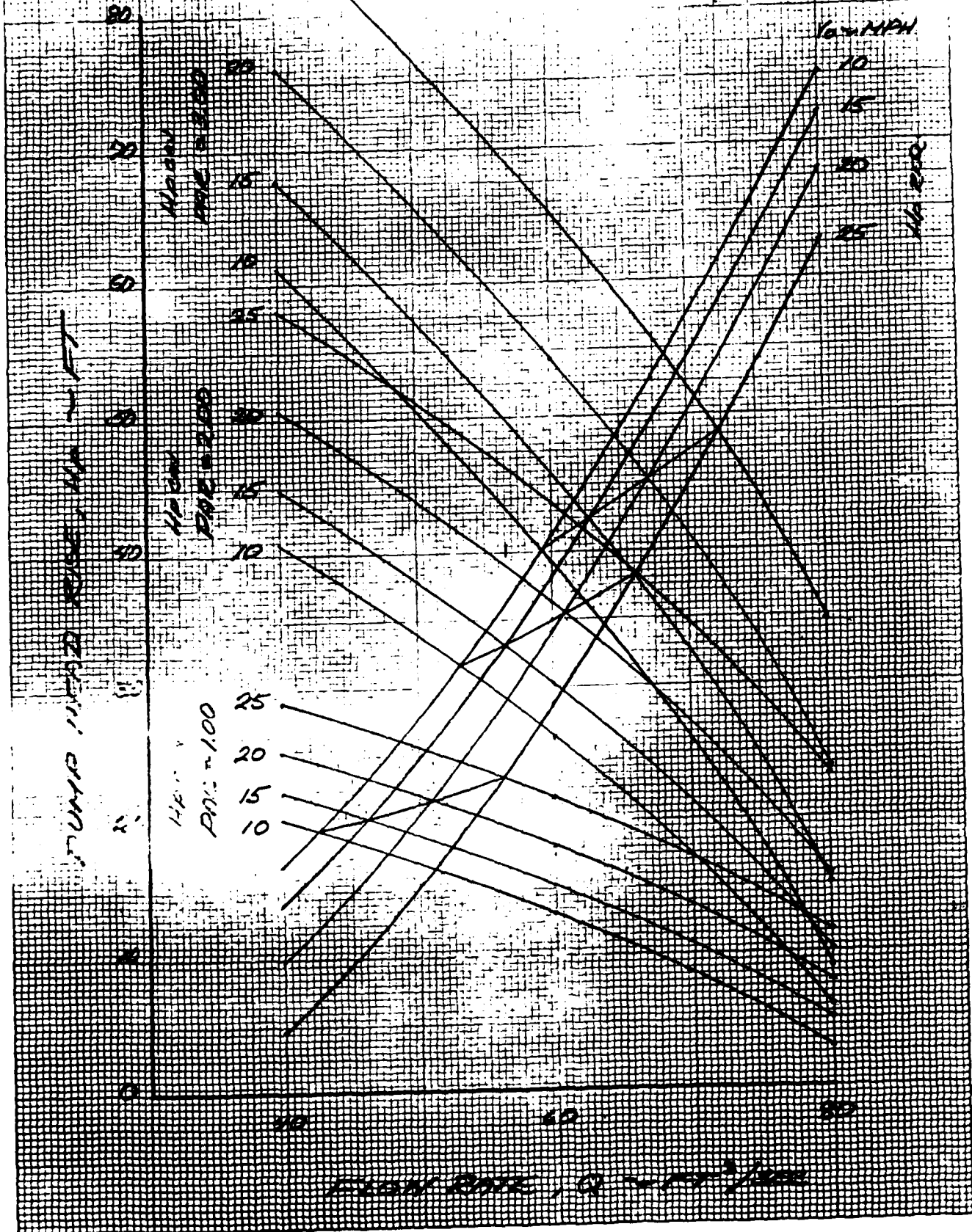
29.40	40	9.94
	60	34.14
	80	68.02

36.75	40	4.64
	60	28.84
	80	62.72

POWER LIMITS



CRITICAL LIMITS

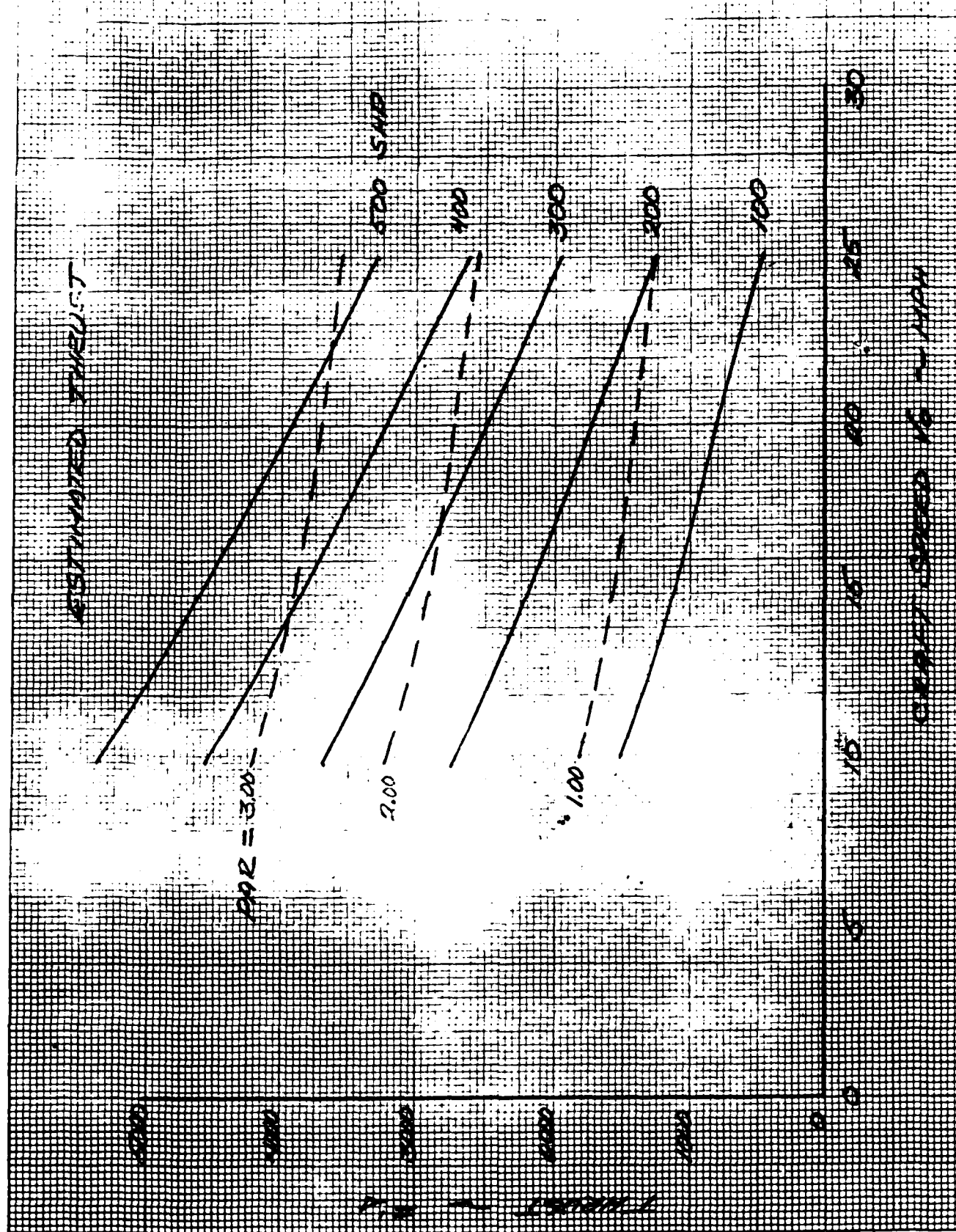


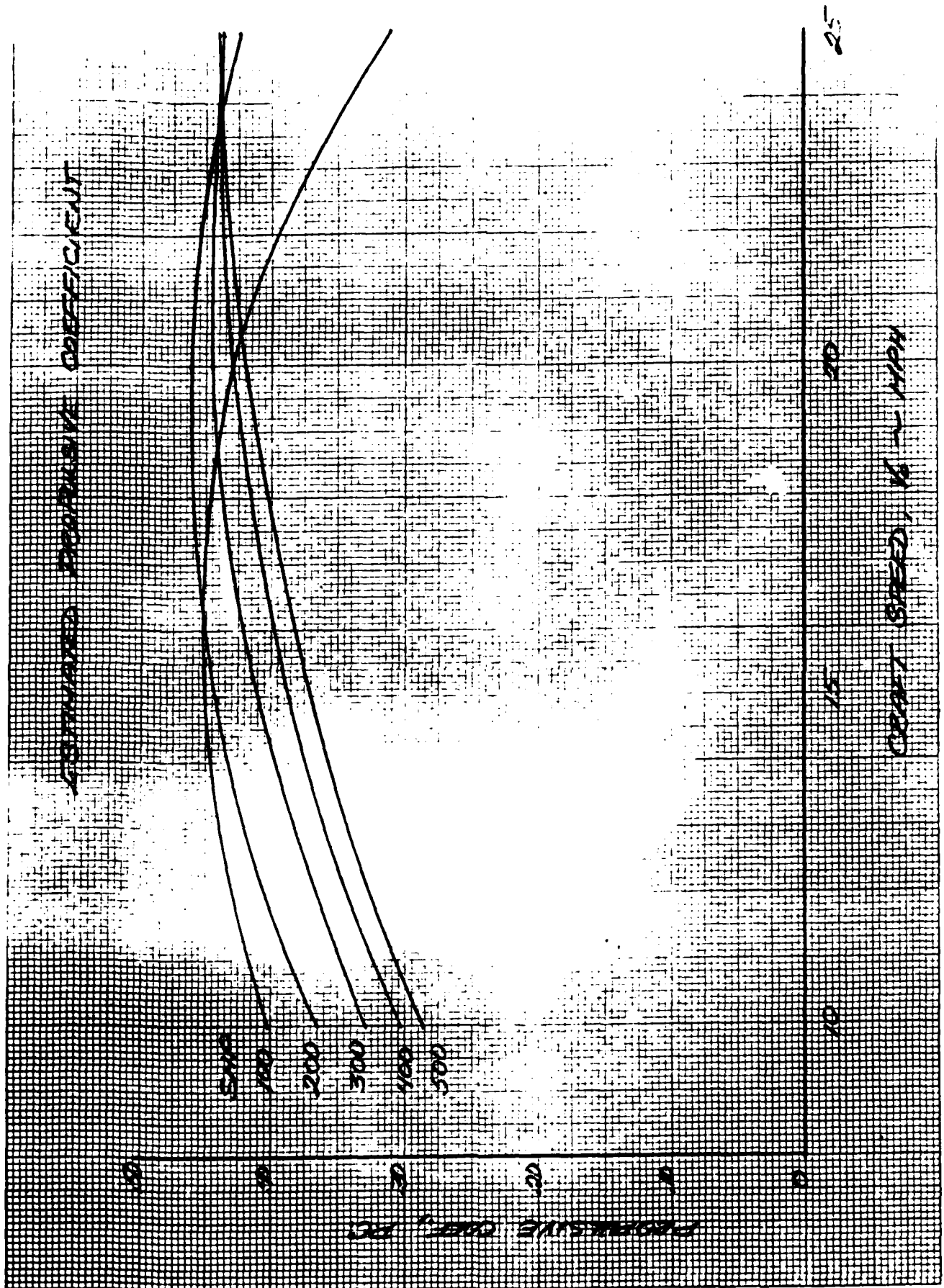
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CALCULATIONS (CONT.)

• ESTIMATED THRUST (POWER LIMIT)

V_0 ft/sec	SHP	Q_{REQ}	A_0	V_0	T	P.C.
14.70	500	65.6	1.1775	55.71	5381	.2876
	400	61.2		51.97	4562	.3048
	300	56.1		47.64	3696	.3293
	200	49.7		42.21	2734	.3654
	100	39.7		33.72	1510	.4036
22.05	500	66.7		56.65	4616	.3701
	400	62.4		52.99	3861	.3871
	300	57.4		48.25	3065	.4096
	200	51.3		43.57	2208	.4426
	100	41.8		35.50	1124	.4506
29.40	500	68.4		58.09	3925	.4196
	400	64.2		54.52	3225	.4310
	300	59.4		50.45	2501	.4456
	200	53.6		45.52	1728	.4618
	100	45.1		38.30	863	.4292
36.25	500	70.7		60.04	3293	.4401
	400	66.5		56.48	2624	.4323
	300	61.9		52.57	1954	.4363
	200	56.5		47.98	1269	.4240
	100	46.9		41.53	667	.4120





STATIC OIL LIFT

REQUIRED PUMP HEAD RISE

$$H_{REQ} = .0121 Q^2$$

Q H_{REQ}

40 19.36

60 43.56

80 77.44

CAVITATION LIMIT

(SEE CAVITATION LIMIT CALCS.)

Q H_{CAV}

40 56.25

60 34.43

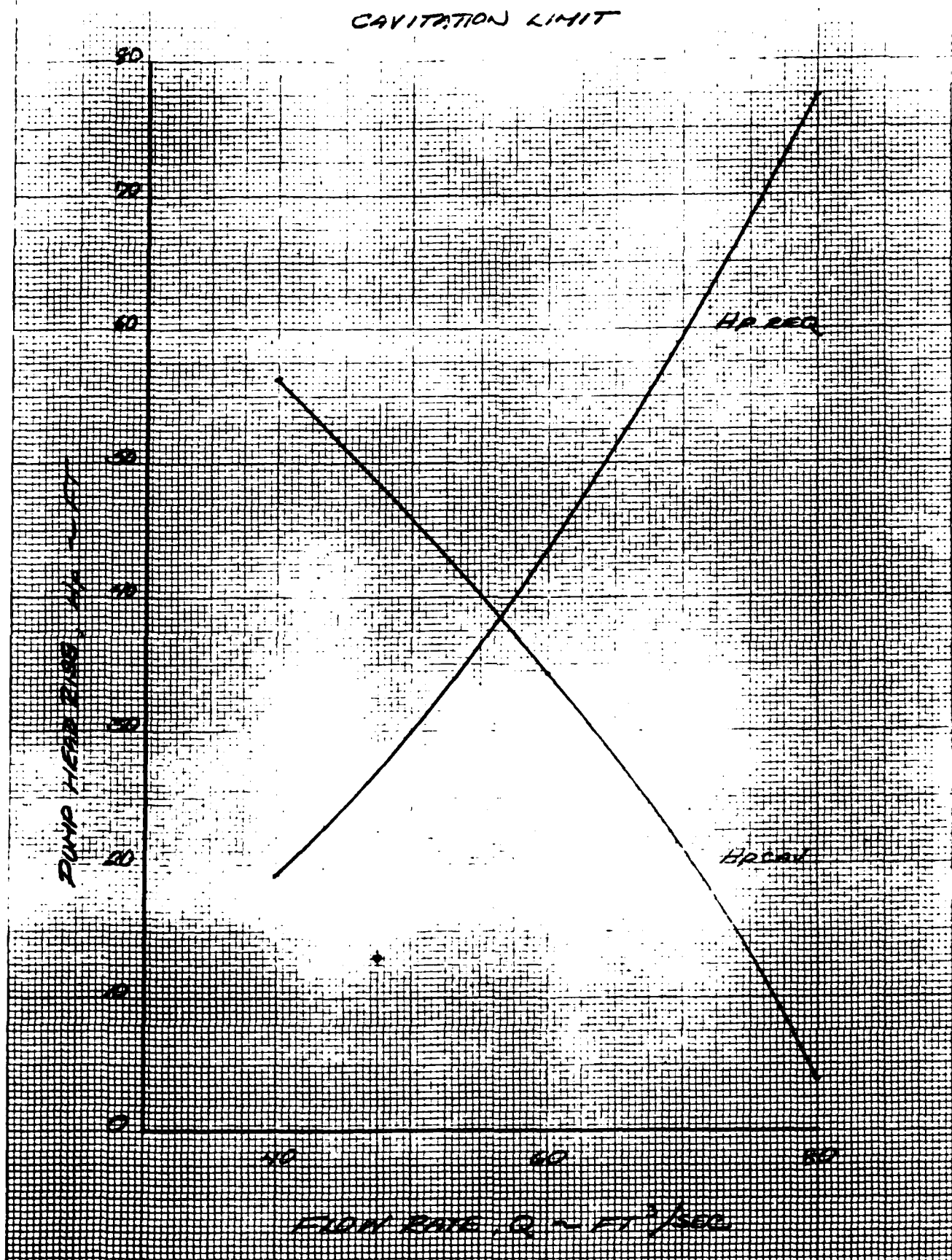
80 3.93

ESTIMATED THRUST

<u>Q_{REQ}</u>	<u>A₂</u>	<u>V₂</u>	<u>T_{STAT}</u>	<u>H_{ATMOS}</u>	<u>H₂</u>	<u>1/H₂</u>	<u>P₂</u>	<u>V₂</u>	<u>$\frac{V_2^2}{2g}$</u>	<u>H_{LS}</u>	<u>P₂</u>
56.4	1.1775	47.43	5403	33.07	3.67	0.272	1.875	30.08	14.05	19.59	1262

USE TO DETERMINE INLET
DESIGN PRESS. (STRUCT.)

STATIC OPTIMIZATION
CAVITATION LIMIT



DERIVATIONS

Ref. 11, 12

VON LAMEREN	FIG. 1	} MODEL TEST OF PROP. IN AXIAL CYL.
"	FIG. 2	
JACUZZI	FIG. 3	MODEL TEST OF INLET

DERIVATIONS

- #1 ESTIMATED INLET & CASING LOSSES
- #2 REQUIRED PUMP HEAD RISE

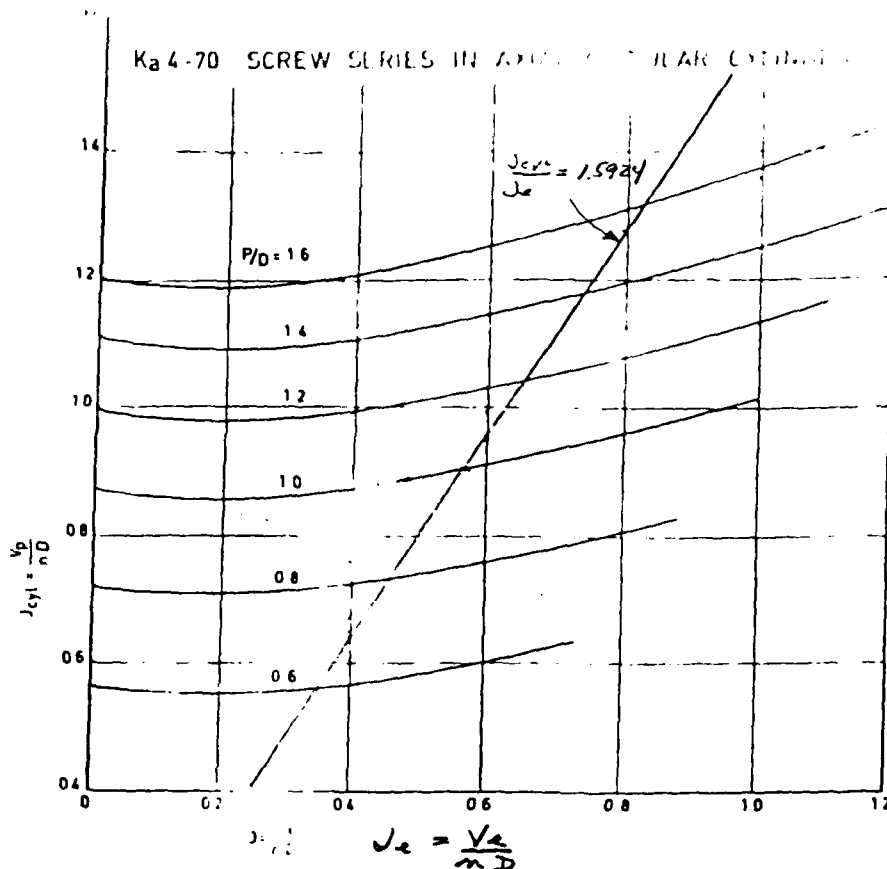


Fig. 29 Relation between velocity of "screw + cylinder" combination and velocity in cylinder

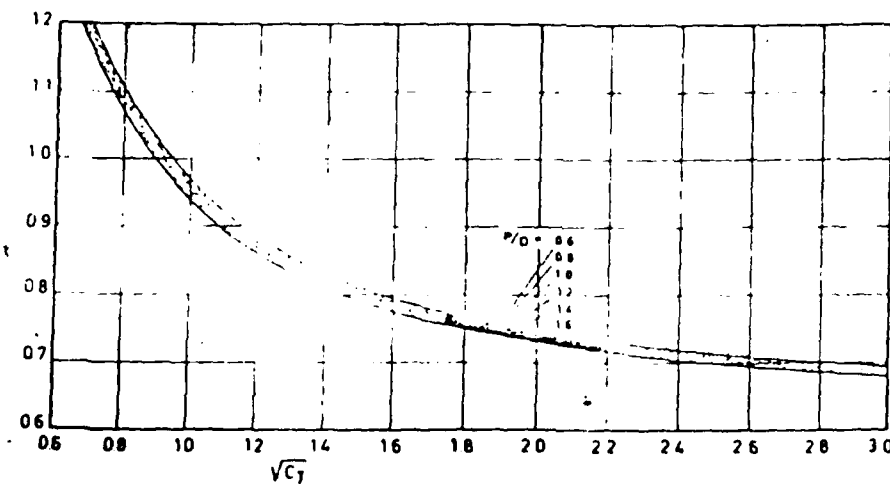


Fig. 30 Relation between thrust coefficient C_T and thrust ratio τ of nozzle no. 19a

figure have been obtained by substituting nozzles with different length-diameter ratios by systems of annular vortexes and calculating the induced velocities in the screw disk.

If the radial displacement of the streamlines is small, we can consider the streamlines as lying approximately on cylindrical planes. If internal friction and turbulence are neglected, the radial

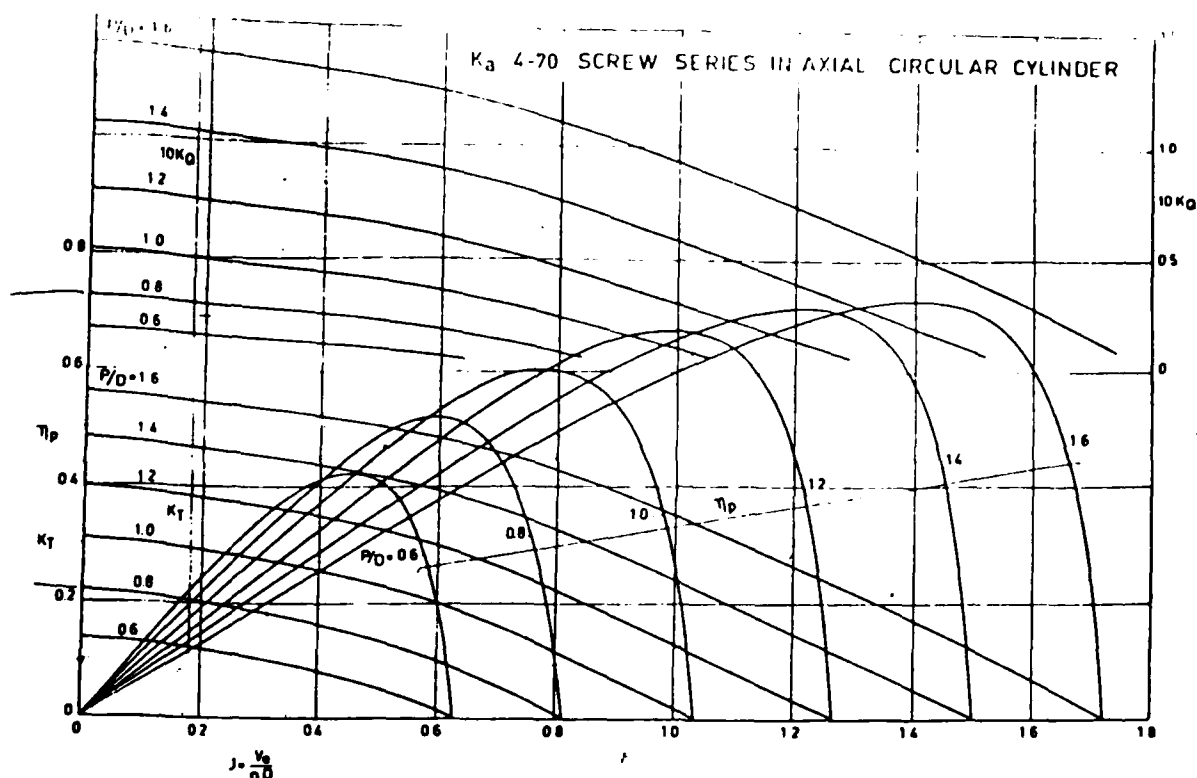


Fig. 28 Results of open-water tests with Ka 4-70 screw series in an axial cylinder

been obtained from the experiments with the Ka 4-70 screw series in an axial circular cylinder and from the application of the momentum theorem.

From the comparison of the axial velocities obtained with these methods, we see that

1 The velocities agree reasonably well at high loadings of the ducted propeller system ($C_T > 1$).

2 The difference between the axial velocities becomes very large at low loadings ($C_T < 1$).

In regard to the second conclusion, the following remark may be made. From Fig. 13 it can be seen that the nozzle drag due to friction becomes substantial at low loadings of the ducted-propeller system. Then, it is no longer permitted to neglect the effect of friction on the force action between nozzle and fluid.

The design of a screw in a nozzle may now be carried out as follows:

With given thrust T or power P , intake velocity V_0 , and number of revolutions n , the B_p and consequently the optimum diameter coefficient D can be determined with the aid of open-water test results of the nozzle considered, in combination with a systematic screw series (see, for instance, Fig. 24). In addition, the thrust coefficient C_T and the propeller thrust-total thrust ratio τ can be determined. With the aid of the experiments

of the systematic screw series in the axial circular cylinder or using the momentum theorem, the axial velocity V_p in the way of the screw can be found. In addition, the mean axial velocity in the vicinity of the screw due to the nozzle action, U_w , and due to the screw action U_p , can be calculated.

The pressure difference created by the screw becomes

$$\Delta p = \frac{T_p}{\frac{\pi}{4} (D^2 - d_s^2)}$$

In order to avoid an excessive loading of the inner radii of the screw blades, the usual assumption for axial pumps that the head is constant for all radii is abandoned. The following radial $\Delta p(r/R)$ distribution is suggested for the screws in nozzle no. 19a:

$$\Delta p(r/R) = [4.88 - 4r/R] \cdot [r/R - 0.133] \Delta p$$

The radial distribution of the axial and tangential velocities at the screw may be approximated as follows:

A reasonable radial distribution of the axial velocities due to the nozzle action can be determined from Fig. 32. The results given in this

COMPARISON OF RECTANGULAR AND ELLIPTICAL INLET RAM RECOVERY VARIATIONS WITH INLET VELOCITY RATIO

(Laboratory Water Channel Test of 2-inch Eye Diameter Waterjet Inlet Models)

SYM	CONFIGURATION
□	RECTANGULAR
○	ELLIPTICAL - 0.3 IN. AFT LIP RADIUS

----- Estimated Performance of Jacuzzi Inlet Configuration

$$\eta_i = 1 - \frac{(P_{T\infty} - \bar{P}_i)}{q_\infty}$$

◆ Measured Performance of 28HJ Inlet

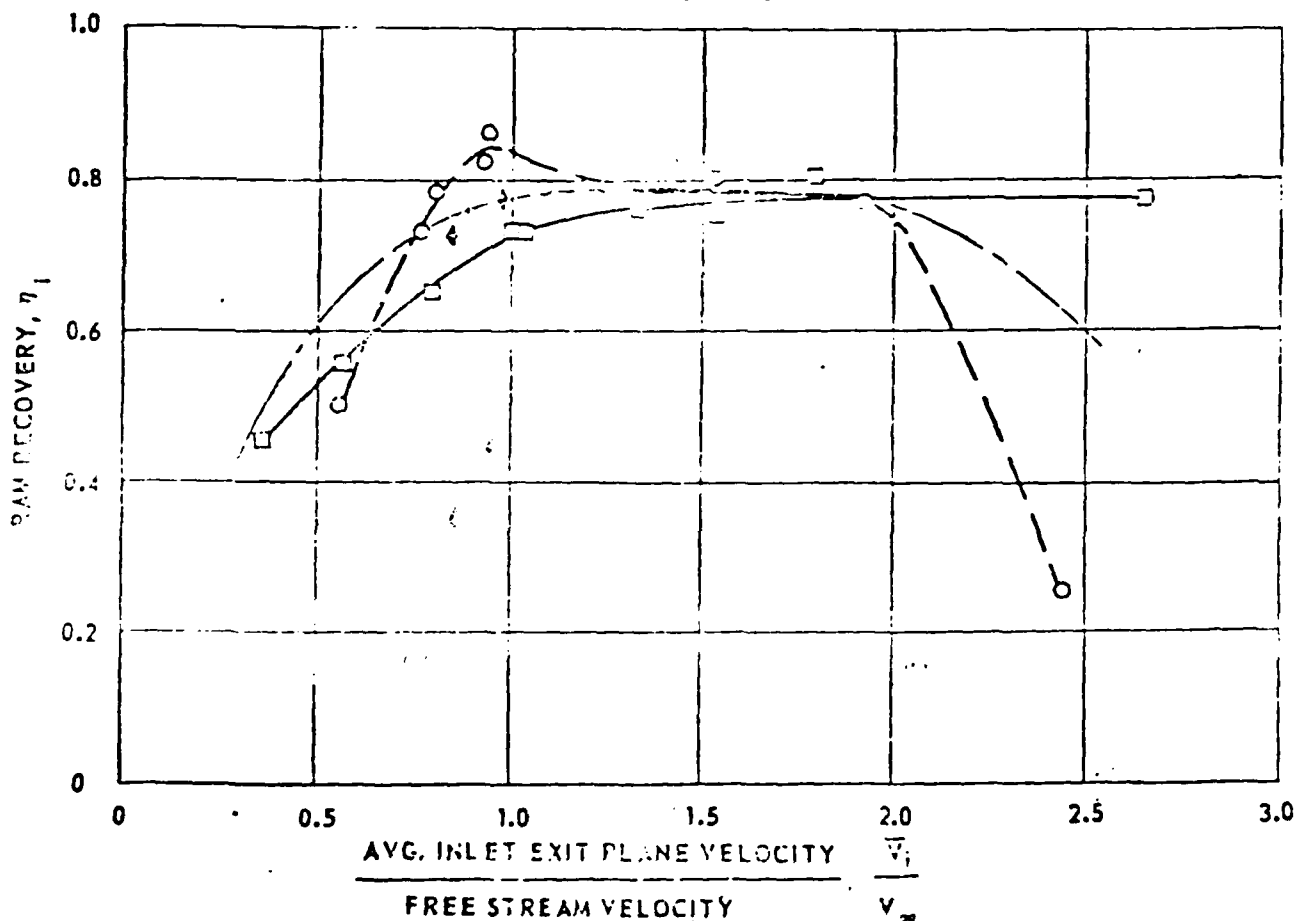


Fig. 3

Jacuzzi C-22

PIPE FRICTION & HEAD

$$Q = \text{NOMINAL FLOW RATE} = 60 \text{ ft}^3/\text{SEC}$$

$$A_I = \text{INLET AREA} = (1.25)(1.50) = 1.875 \text{ ft}^2$$

$$V_I = \text{INLET VELOCITY} = Q/A_I = 60/1.875 = 32.00 \text{ ft/sec}$$

$$d_e = \text{EQUIV. INLET DIAM} = \frac{4A_I}{2(1.25+1.50)} = \frac{4(1.875)}{2(1.25+1.50)} = 1.36'$$

$$R_e = \frac{V_I d_e}{\nu} = \frac{(32)(1.36)}{1.24 \times 10^{-5}} = 3.51 \times 10^6$$

$$e/D = \text{RELATIVE ROUGHNESS} = .000004$$

$$f = \text{FRICTION FACTOR} = .0096$$

$$L_I = \text{INTAKE LENGTH} = \sqrt{15^2 - 9^2} = 12.98' = 1.08'$$

$$L_B = \text{EQUIV. LENGTH OF BEND} = \left(\frac{L}{D}\right) \left(d_e\right) \left(\frac{K}{90}\right) = (36)(1.36)\left(\frac{35}{90}\right) = 19.04'$$

$$L = \text{TOTAL EQUIV. LENGTH} = L_I + L_B = 1.08 + 19.04 = 20.12'$$

$$HL = f \left(\frac{L}{d_e}\right) \left(\frac{V_I^2}{2g}\right) = (.0096) \left(\frac{20.12}{1.36}\right) \frac{(32)^2}{2(32.2)} = 2.258'$$

SHAFT

$$C_D = \text{DRAG COEFF. DUE TO CROSSFLOW} = 1.1 \sin^3\left(\frac{\theta}{2}\right) = (1.1) \sin^3\left(\frac{35}{2}\right) = .0299$$

$$V_I = 32 \text{ ft/sec}$$

$$L = \text{SHAFT LENGTH} = 2'$$

$$d = \text{SHAFT DIAM} = 1.50" = .125'$$

$$D = \text{SHAFT "DRAG"} = C_D R_e V_I^2 L d = (.0299) (36) (32)^2 (2) (.125) = 2.65$$

$$HL = \frac{D}{\rho g A_I} = \frac{2.65}{2(32.2)(1.875)} = .0634'$$

(DEVIATION #1)

TRANSITION

$$A_T = \text{CROSS SECTION AREA} = \frac{1.875 + 1.125}{2} = 1.5263 \text{ FT}^2$$

$$V_T = \text{VELOCITY} = Q/A_T = 60/1.5263 = 39.31 \text{ FT/SEC}$$

$$d_e = \text{EQUIV. DIAM.} = \sqrt{\frac{1.5263}{.785}} = 1.39'$$

$$Re = \frac{V_T d_e}{\mu} = \frac{(39.31)(1.39)}{1.24 \times 10^{-5}} = 4.41 \times 10^6$$

$$e/D = \text{RELATIVE ROUGHNESS} = .0000035$$

$$f = \text{FRICTION FACTOR} = .0093$$

$$L_T = \text{TRANSITION LENGTH} = 2.5'' = .625'$$

$$HL = f \left(\frac{L}{d_e} \right) \frac{V_T^2}{2(32.2)} = (.0093) \left(\frac{.625}{1.39} \right) \frac{(39.31)^2}{2(32.2)} = .100'$$

BEARING TUBE

$$A_D = \text{CROSS SECTION AREA} = (.96)(.785)(1.25)^2 = 1.1775 \text{ FT}^2$$

$$V_D = \text{VELOCITY} = Q/A_D = 60/1.1775 = 50.96 \text{ FT/SEC}$$

$$L = \text{TUBE LENGTH} = 1.25'$$

$$Re = \frac{V_D L}{\mu} = \frac{(50.96)(1.25)}{1.24 \times 10^{-5}} = 5.14 \times 10^6$$

$$C_f = .00329$$

$$S = \text{TUBE WALL THICKNESS} = (1.25) \pi (1.25) = .98 \text{ FT}^2$$

$$D = \text{TUBE DRAG} = (C_f + .0008)(S) \left(\frac{V_D}{2} \right)^2 = (.00329 + .0008)(.98) \left(\frac{50.96}{2} \right)^2 = 10.36$$

$$HL = \frac{D}{\rho g A_D} = \frac{10.36}{2(32.2)(1.1775)} = .137'$$

Example 1 - Struts (20 in dia)

Struts

$$A_p = \text{CROSS SECTION AREA} = 1.1075 \text{ ft}^2$$

$$V_p = \text{VELOCITY} = Q/A_p = 60/1.1075 = 53.96 \text{ ft/sec}$$

$$c = \text{STRUT CHORD} = 3" = .25'$$

$$Re = \frac{V_p c}{\nu} = \frac{(53.96)(.25)}{1.24 \times 10^{-5}} = 1.03 \times 10^6$$

$$C_f = .00437$$

$$t/c = \text{STRUT THICKNESS RATIO} = .3125/3 = .1042$$

$$C_D = 2(C_f + .0006)(1 + 1.2 t/c) = 2(.00437 + .0006)(1 + 1.2 \times .1042) = .0116$$

$$S = \text{STRUT PLAN FORM AREA} = 4(.50)(.25) = .50 \text{ ft}^2$$

$$D = \text{STRUT DRAG} = C_D S \rho/2 V_p^2 = (.0116)(.50)(3/2)(53.96)^2 = 15.06 \text{ lb}$$

$$HL = \frac{D}{\rho g A_p} = \frac{15.06}{2(62.4)(1.1075)} = .199'$$

TOTAL INLET LOSS

INTAKE FRICTION + END	2.258
SHAFT	.063
WHEEL	.100
VALVE	.137
	<u>.199</u>

$$HL = 2.757'$$

$$K = \frac{HLE}{Q_{nom}} = \frac{2.752}{(60)^2} = .000766$$

$$HLE = .000766 Q^2$$

CASING

$$Q = \text{NOMINAL FLOW RATE} = 60 \text{ FT}^3/\text{SEC}$$

$$A_p = 1.1775 \text{ FT}^2$$

$$V_p = Q/A_p = 60/1.1775 = 50.96 \text{ FT/SEC}$$

$$d = \text{CASING DIAM} = 1.25'$$

$$\epsilon = \frac{V_p d}{1.24 \times 10^{-5}} = \frac{(50.96)(1.25)}{1.24 \times 10^{-5}} = 5.14 \times 10^6$$

$$e/d = \text{RELATIVE ROUGHNESS} = .000004$$

$$f = \text{FRICTION FACTOR} = .009$$

$$L_c = \text{CASING LENGTH} = 1.25'$$

$$h/L = f \left(\frac{L_c}{d} \right) \left(\frac{V_p^2}{2g} \right) = (.009) \left(\frac{1.25}{1.25} \right) \frac{(50.96)^2}{2(32.2)} = .363'$$

$$L = \frac{h/L_c}{(Q_{100})^2} = \frac{.363}{(60)^2} = .000101$$

$$h_c = .000101 Q^2$$

$$H_{PRE} = H_{L1} + H_{L2} + H_{L3} - H_0$$

$$H_{L1} = \frac{V_0^2}{2g}$$

$$V_0 = \frac{Q}{A_0}$$

$$A_0 = A_T = 1.1775 \text{ ft}^2$$

$$H_{L1} = \frac{Q^2}{(1.1775)^2 (2)(32.2)} = .0112 Q^2$$

$$H_{L2} = .000766 Q^2$$

$$H_{L3} = .000101 Q^2$$

(DERIVATION #1)

$$H_0 = \frac{(RPR) V_0^2}{2g}$$

$$RPR = .70$$

$$H_0 = \frac{(.70) V_0^2}{2(32.2)} = .0109 V_0^2$$

$$H_{PRE} = .0112 Q^2 + .000766 Q^2 + .000101 Q^2 - .0109 V_0^2$$

$$= .0121 Q^2 - .0109 V_0^2$$

STRUCTURAL ANALYSES

- SNAFT
- CASIN;

PROPS. SHAFT

TORSIONAL STRESS

$$N = \text{PROP. SPEED} = \frac{60 Q_{eq}}{A_p J_{c, - D}} = \frac{(60)(67.1)}{(1.1775)(.905)(.125)} = 3022 \text{ RPM}$$

$$SHP = 462$$

$$Q' = \text{PROP TORQUE} = \frac{63024 \text{ SHP}}{N} = \frac{(63024)(462)}{3022} = 9635 \text{ IN} \cdot \text{IN}$$

$$d = \text{SHAFT DIAM.} = 1.50 \text{ IN}, r = .75 \text{ IN}$$

$$J = \text{POLAR MOM. OF INERTIA} = \frac{\pi}{2} r^4 = \left(\frac{\pi}{2}\right)(.75)^4 = .4970$$

$$S_s = \text{TORSIONAL STRESS} = \frac{Q' r}{J} = \frac{(9635)(.75)}{(.4970)} = 14,540 \text{ PSI}$$

$$\text{FACTOR OF SAFETY} = \frac{20,000}{14,540} = 1.38 \text{ ON SHEAR YIELD (ASMET 22)}$$

20 MPH
CR. LIMIT
FAC = 3.00

WHIRLING FREQUENCY

$$W = \text{WEIGHT PER UNIT LENGTH} = (.285)(1.5)(1)(.28) = .4946 \text{ LB/IN}$$

$$L = \text{DISTANCE BETWEEN SUPPORTS} = 27 \text{ IN}$$

$$I = \text{MOM. OF INERTIA} = .049 L^4 = (.049)(1.5)^4 = .2481 \text{ IN}^4$$

$$D = \text{STATIC DEFLECTION TO OWN WT}$$

$$= .00542 \frac{W L^4}{EI} = \frac{(0.00542)(.4946)(1.5)^4}{(30,000)(.2481)} = .000205 \text{ IN FREE-FIXED}$$

$$f = \text{WHIRLING FREQ.} = \frac{3.55}{D^{1/2}} = \frac{3.55}{.000205^{1/2}} = 248 \text{ CPS}$$

$$= 14,874 \text{ RPM}$$

$$N_{DES} = 3022 \text{ RPM}$$

CASING STRUCTURE

INLET CASING DESIGN PRESSURE

P = DESIGN PRESSURE

EXTERNAL PRESSURE - INTERNAL PRESSURE

$$\begin{aligned} & (H_{ATH.} + H_2) - H_{I_2} \left(\frac{64}{144} \right) \\ & = (33.08 + 3.00) - 19.57 \left(\frac{64}{144} \right) = 7.33 \text{ psi} \end{aligned}$$

NOTE: MINIMUM H_{I_2} OCCURS DURING
STATIC OPERATION AT CAV. UNIT

INLET CASING STRESS

P = DESIGN PRESSURE = 7.33 psi

l = SPAN = 18"

W = UNIT WIDTH = 1"

M = BENDING MOM. IN PLATE = $\frac{Pl^2}{12} = \frac{(7.33)(18)^2(1)}{12} = 192.91 \text{ in}^2/\text{IN WIDTH}$

t = PLATE THICKNESS = .125"

Z = SECTION MODULUS = $\frac{bt^3}{12} = \frac{(1)(.125)^3}{12} = .00158 \text{ in}^3/\text{IN WIDTH}$

σ = BENDING STRESS = $\frac{M}{Z} = \frac{192.91}{.00158} = 121,462 \text{ psi}$

ESTIMATED WEIGHTS

Casing Weight

1.117 Casing

$$A = \left[\frac{2(25.85)(18)}{2} + (15)(25.85) + 2 \left(\frac{35}{360} \right) (285)(36)^2 + \left(\frac{35}{360} \right) \pi (36)(18) \right] \frac{1}{144} = 8.44 \text{ ft}^2$$

$$t = .3125", \quad w = (.3125)(144)(.096) = 4.32 \text{ #/ft}^2$$

$$W = (8.44)(4.32) = 36.46 \text{ #}$$

TRANSITION

$$A = \left[\frac{2(15)(18)}{2} + 15 \pi \right] \frac{2.5}{144} = 2.95 \text{ ft}^2$$

$$t = .3125", \quad w = 4.32 \text{ #/ft}^2$$

$$W = (2.95)(4.32) = 12.73 \text{ #}$$

PROPELLER CANNY

$$A = 15 \pi (15) / 144 = 4.91 \text{ ft}^2$$

$$t = .3125", \quad w = 4.32 \text{ #/ft}^2$$

$$W = (4.91)(4.32) = 21.21 \text{ #}$$

STUPTS

$$V = 4(3)(.3125)(.71)(6) = 15.96 \text{ in}^3$$

$$w = .096 \text{ #/in}^3$$

$$W = (15.96)(.096) = 1.53 \text{ #}$$

BEARING TUBE

S/A	d	a	T.M.	f(V)
1	3.00	2.86	1/2	3.53
2	3.00	2.06	1	2.02
3	2.625	5.41	1	5.41
4	2.06	3.34	1/2	1.62
				12.67

$$V = 3(12.67) - 6(2.02)(2) = 3(12.67)(1.15) = 26.96 \text{ in}^3$$

$$w = .096 \text{ #/in}^3$$

$$W = (26.96)(.096) = 2.59 \text{ #}$$

CASING TOTAL

$$W = 36.46 + 12.73 + 21.21 + 1.53 + 2.59 = 75 \text{ # COMB. CONST.}$$

$$= (65) \left(\frac{1.15}{2.66} \right) =$$

$$42 \text{ # COMPOSITE CONST.}$$

PROBLEM 11

$$W = (47.65) \left(\frac{15}{20} \right)^2 \left(\frac{PAR}{1.01} \right) = 19.90(PAR)$$

$$= (19.90)(3) \cdot 60^{\#} \text{ CONV. CONST.} \quad PAR = 3.00$$

$$= (60) \left(\frac{4.50}{2.66} \right) = 34^{\#} \text{ COMPOSITE CONST.}$$

SHAFTING WEIGHT

SHEET

$$\begin{aligned} L &= 54" = 4.50 \text{ FT} \\ W &= (.785)(1.50)^2(12)(.28) = 5.95 \text{ #/FT.} \\ W &= (4.50)(5.95) = 26.71 \text{ #} \end{aligned}$$

MISC. (BEARINGS, SEALS, HOUSING:)

$$W = 5 \text{ #}$$

SHAFTING TOTAL

$$W = 26.71 + 5 = 32 \text{ #}$$

WATER WEIGHT

INLET CASING

$$V = (16) \frac{(25.45)(16)}{2} + \left(\frac{35}{240} \right) (25.45)(24)^2 (15) = 4974 \text{ in}^3$$

TRANSITION

$$V = \left[\frac{(18)(20) + 25.45(15)^2}{2} \right] (7.5) = 2012 \text{ in}^3$$

PROPELLER CASING

$$V = (15)(25.45)(15)^2 = 2649 \text{ in}^3$$

TOTALS

$$V = 4974 + 2012 + 2649 = 9635 \text{ in}^3 = 5.58 \text{ FT}^3$$

$$W = 64 \text{ #/FT}^3$$

$$W = (5.58)(64) = 357 \text{ #}$$

WEIGHT SUMMARY

	<u>CONCRETE CONST.</u>		<u>COMPOSITE CONST.</u>	
	<u>WT.</u>	<u>MATL.</u>	<u>WT.</u>	<u>MATL.</u>
CASING	75	ALUM. (50K-4116)	42	POLYESTER - GRASS CORD
PROPANE	60	NICKEL AL. BR.	34	POLYCARB. - GLASS
SHAFING	32	AQUINER 22	32	AQUINER 22
DRY WEIGHT	167	#	108	#
WATER	35		35	
WET WEIGHT	524	#	465	#

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