CALIFORNIA INSTITUTE OF TECHNOLOGY

Hydrodynamics Laboratories

PUMP PERFORMANCE
OF THE
MARK 40 PUMPJET

A REPORT ON RESEARCH CONDUCTED UNDER CONTRACT WITH
THE BUREAU OF ORDNANCE OF THE DEPARTMENT OF THE NAVY
AND OFFICE OF NAVAL RESEARCH

[redacted]

Department of the Navy
Bureau of Ordnance
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(15) [redacted]
Office of Naval Research
Contract N6-onr-244

(6) PUMP PERFORMANCE OF THE MARK 40 PUMPJET,

[redacted]

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[redacted]

(17) 2

(174250)

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Acknowledgment

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PUMP PERFORMANCE OF THE MARK 40 PUMPJET

CHARACTERISTIC CURVES FOR THE MARK 40 PUMPJET IN NORMAL PUMP AND REVERSE TURBINE REGIONS OF OPERATION

Summary

From the results of tests conducted in the Hydraulic Machinery Laboratory[1*] at the California Institute of Technology, the pump performance was analyzed and found to be satisfactory for the pumpjet installation in the Mark 40 Torpedo. The pump will operate at or very near the point of best efficiency, which is 84.5 ± 1.0 per cent at a projectile speed of 80 knots.

The propulsion unit was tested from zero flow rate point through zero head point into the reverse turbine range. The performance was found to be very similar to that of a conventional centrifugal pump fitted with a diffuser vane case or a volute case.

Introduction

The Mark 40 Torpedo is to be powered with a turbo-pumpjet. The turbo-pumpjet is a gas-turbine-powered centrifugal pump. Water enters the pump through a cylindrical duct in the torpedo nose and is discharged through eight nozzles in the form of high velocity jets. The reaction of the jets furnish the thrust necessary to overcome the projectile drag.

The complete hydraulic propulsion unit, namely, the entrance duct, the pump impeller, the diffuser casing and discharge nozzles, is referred to in this report as the "pumpjet."

Purpose

The object of the tests was to find the head, brake horsepower, and efficiency, vs. flow rate relationships for the Mark 40 pumpjet in the normal pumping, power dissipation, and the reverse turbine regions of operation. ~~(Figs. 1 - 5)~~. Under normal operating conditions the ram effect on the projectile nose assures high positive suction pressure and eliminates the possibility of cavitation. Thus, in this series of tests, cavitation studies were not made.

Test Setup

Hydraulically, the pumpjet unit used in these tests was an exact, full scale, duplicate of the Mark 40 installation. The U. S. Naval Ordnance Test Station, in Pasadena, California, furnished the Laboratory with a Mark 40 nose section and a pumpjet impeller. The nose section contains, in one casting, the entrance duct and the diffuser casing up to the straightener vanes which precede the discharge nozzles. The Mark 40 nozzle section was impractical to use in the test setup, hence, a set of eight duplicate nozzles and straightener vanes, more adaptable to test purposes, was made in the Laboratory shop and installed in the test unit.

* See bibliography at end of report.

The test setup is shown diagrammatically in Figs. 6 and 7. The pumpjet was placed in a closed hydraulic circuit. The entrance duct was fitted to a contraction nozzle designed to deliver a uniform energy flow to the impeller which is the condition that would be experienced by the prototype. Preceding the nozzle was a long, straight length of 12 in. pipe. The eight discharge nozzles were each followed by a needle type regulating valve which was used to equalize the flow in all nozzles. The regulating valves were adjusted with the pumpjet operating at the best efficiency point. This adjustment was necessary because the pressure in the discharge manifold, unlike that on the actual torpedo, was not everywhere equal. The torus-like discharge manifold merely afforded a convenient means of collecting the flow from the various jets.

The pumpjet was powered by the Laboratory dynamometer through a direct drive. Figs. 8 through 12 show the test unit in various stages of completion.

Measurements

The dynamometer standard torque mechanism was used to obtain the input torque for normal pump and power dissipation regions of operation and the output torque for the reverse turbine tests. The dynamometer speed was measured and controlled by the existing standard frequency speed control.

The rate of flow through the unit was measured by the appropriate size venturimeter permanently located in the Laboratory. The meters were located in the pump discharge line.

The differential head generated by the pump was measured by a differential pressure gage. On the suction side of the pump the pressure tap was located on a piezometer slot in the inlet duct. On the discharge side of the pump a pressure tap was located at a point just ahead of the grid straightener vanes in each of the discharge nozzle passages (Fig. 7). The discharge pressure lines from the eight nozzles were led to a common manifold (Fig. 11) and then to the gage. The differential head, so determined, did not include the losses incurred in the grids or the nozzles. In preliminary tests the discharge pressure was measured approximately 1/2 in. downstream from the grids in two nozzles only. The differential pressure across the pump, in this case, was of the order of 2 per cent less than that obtained when the discharge pressure was measured ahead of the grids.

Operation

The tests were conducted without any mechanical difficulties from the pumpjet with the exception of the outboard ball bearing. This bearing, exposed to fresh water, failed after an estimated operating time of 5-10 hours. It was run at speeds up to 4000 rpm, the highest test speed. It was found that packing the bearing in the test unit with commercial automobile water pump grease greatly extended its life. It is unlikely that this bearing will fail from this cause in the prototype since its time of operation is very short.

Test Results

The results of the tests are presented graphically in Figs. 1 through 5

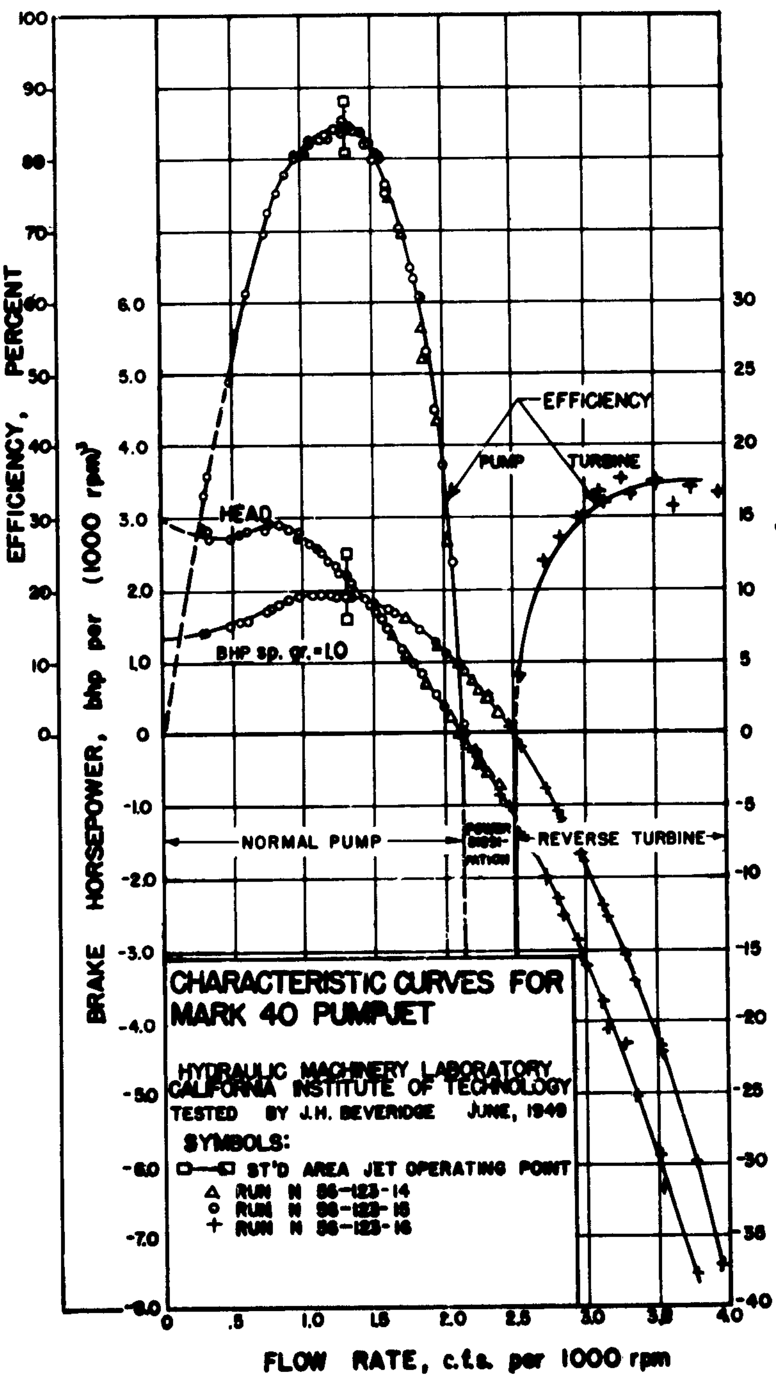

EFFICIENCY, PERCENT
BRAKE HORSEPOWER, bhp per (1000 rpm)³
HEAD, ft. per (1000 rpm)²
FLOW RATE, c.f.s. per 1000 rpm
EFFICIENCY
PUMP
TURBINE
HEAD
BHP sp. gr.=1.0
NORMAL PUMP
POWER DISSI-PATION
REVERSE TURBINE
CHARACTERISTIC CURVES FOR MARK 40 PUMPJET
HYDRAULIC MACHINERY LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
TESTED BY J.H. BEVERIDGE JUNE, 1949
SYMBOLS:
ST'D AREA JET OPERATING POINT
RUN N 56-123-14
RUN N 56-123-15
RUN N 56-123-16

Fig. 1

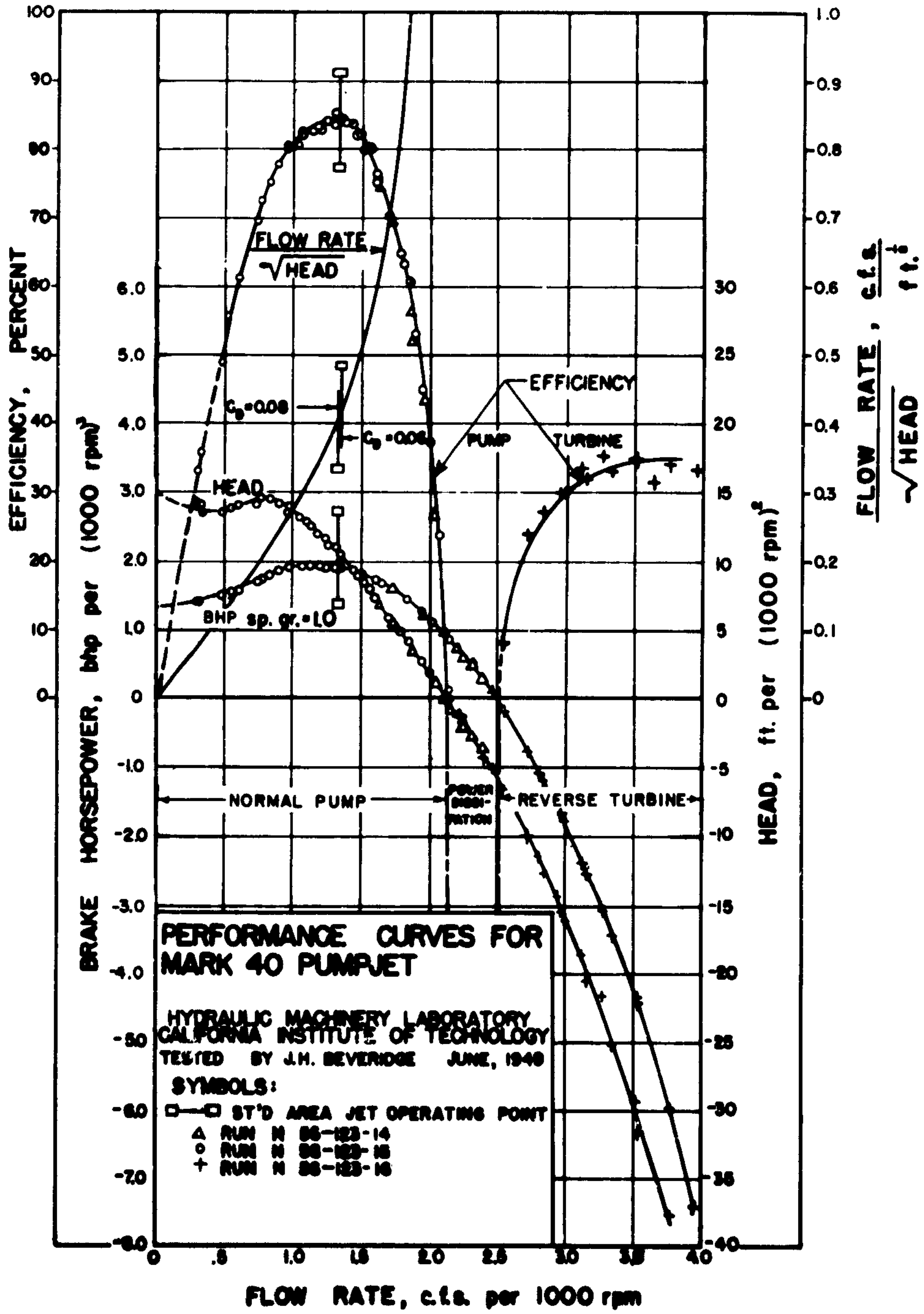

PERFORMANCE CURVES FOR MARK 40 PUMPJET
HYDRAULIC MACHINERY LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
TESTED BY J.H. BEVERIDGE JUNE, 1948
SYMBOLS:
ST'D AREA JET OPERATING POINT
RUN N 86-123-14
RUN N 86-123-15
RUN N 86-123-16
EFFICIENCY, PERCENT
BRAKE HORSEPOWER, bhp per (1000 rpm)^3
HEAD, ft. per (1000 rpm)^2
FLOW RATE / $\sqrt{\text{HEAD}}$, c.f.s. / ft.$^{1/2}$
FLOW RATE, c.f.s. per 1000 rpm
EFFICIENCY
PUMP
TURBINE
HEAD
BHP sp. gr. = 1.0
C_p = 0.08
C_p = 0.06
NORMAL PUMP
POWER DISSIPATION
REVERSE TURBINE

Fig. 2

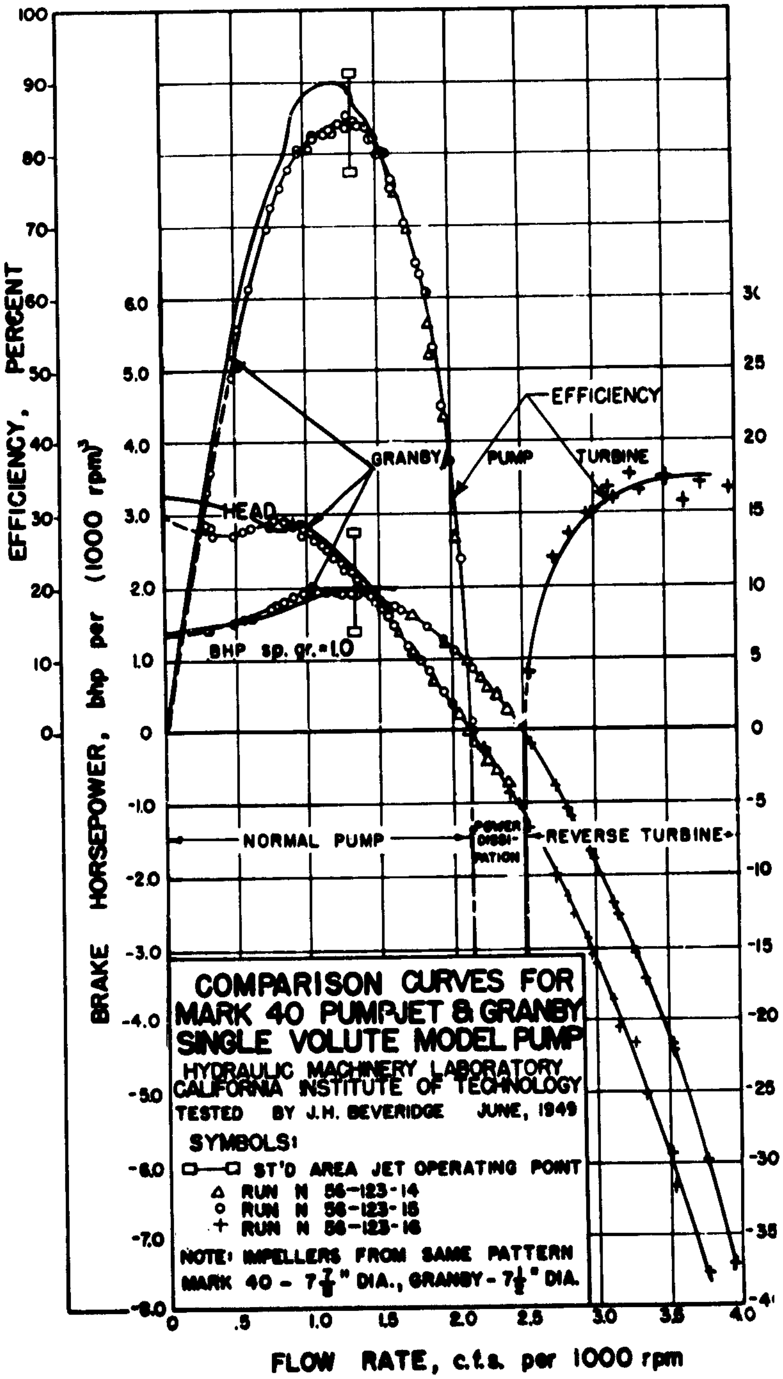

COMPARISON CURVES FOR MARK 40 PUMPJET & GRANBY SINGLE VOLUTE MODEL PUMP
HYDRAULIC MACHINERY LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
TESTED BY J.H. BEVERIDGE JUNE, 1949
SYMBOLS:
ST'D AREA JET OPERATING POINT
RUN N 56-123-14
RUN N 56-123-15
RUN N 56-123-16
NOTE: IMPELLERS FROM SAME PATTERN
MARK 40 - 7 7/8" DIA., GRANBY - 7 1/2" DIA.
EFFICIENCY, PERCENT
BRAKE HORSEPOWER, bhp per (1000 rpm)³
HEAD, ft. per (1000 rpm)²
FLOW RATE, c.f.s. per 1000 rpm
EFFICIENCY
GRANBY
PUMP
TURBINE
HEAD
BHP sp. gr. = 1.0
NORMAL PUMP
POWER DISSI-PATION
REVERSE TURBINE

Fig. 3

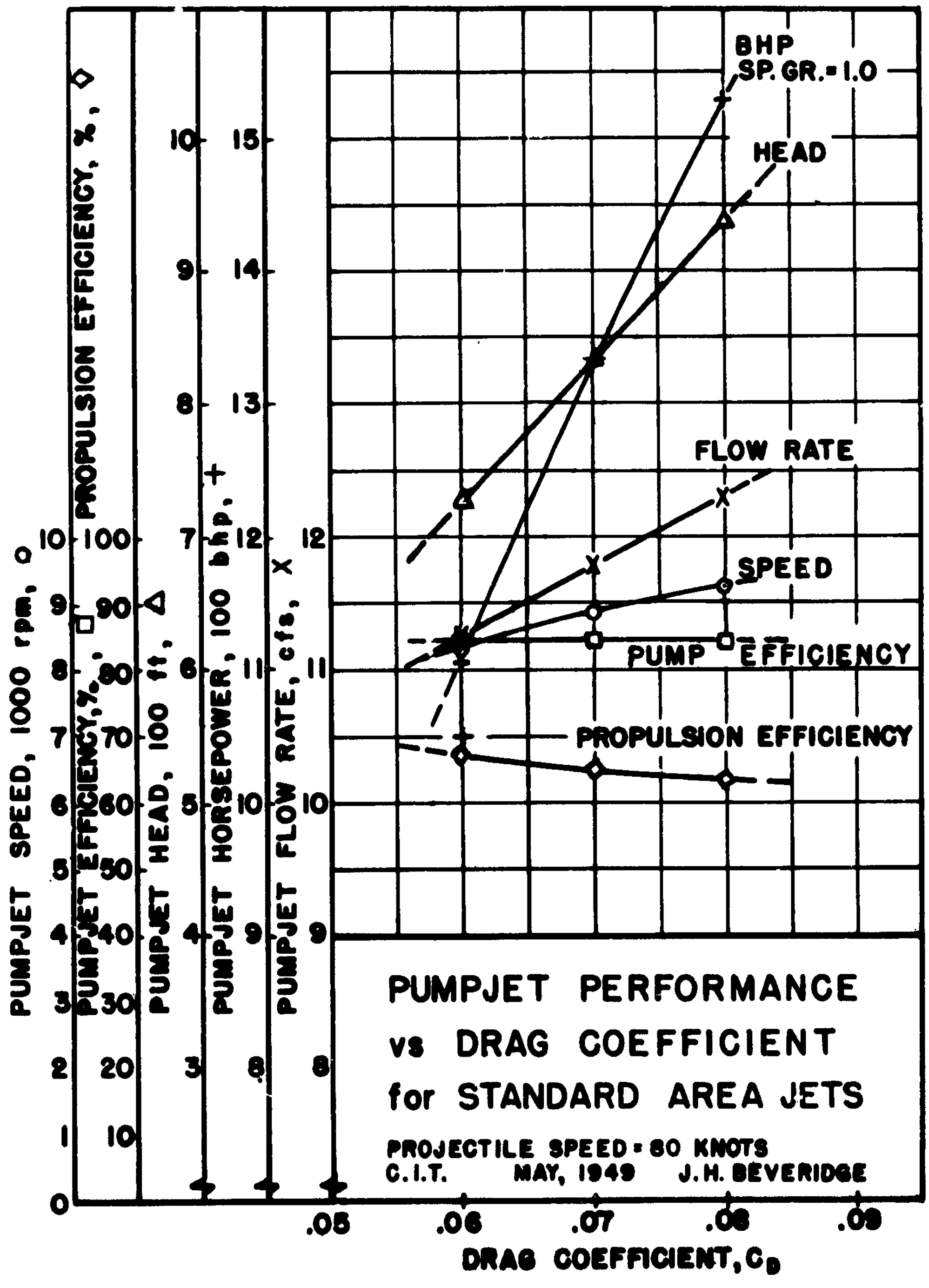

BHP
SP. GR. = 1.0
HEAD
FLOW RATE
SPEED
PUMP EFFICIENCY
PROPULSION EFFICIENCY
PUMPJET PERFORMANCE
vs DRAG COEFFICIENT
for STANDARD AREA JETS
PROJECTILE SPEED = 60 KNOTS
C.I.T. MAY, 1949 J.H. BEVERIDGE
PUMPJET SPEED, 1000 rpm, ○
PUMPJET EFFICIENCY, %, □
PROPULSION EFFICIENCY, %, ◇
PUMPJET HEAD, 100 ft, △
PUMPJET HORSEPOWER, 100 bhp, +
PUMPJET FLOW RATE, cfs, X
DRAG COEFFICIENT, C_D
.05
.06
.07
.08
.09

Fig. 4

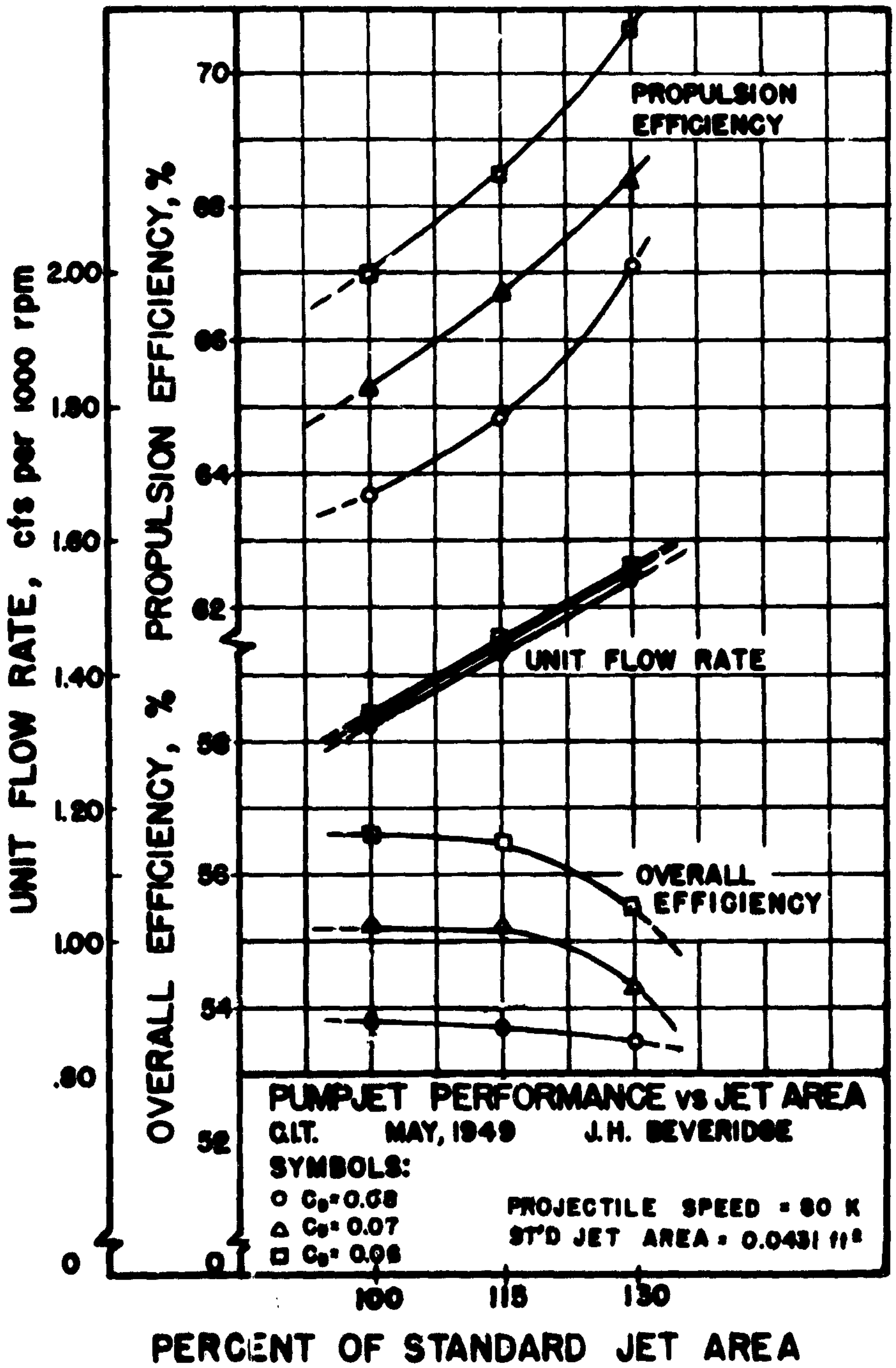

PROPULSION EFFICIENCY
UNIT FLOW RATE
OVERALL EFFICIENCY
UNIT FLOW RATE, cfs per 1000 rpm
OVERALL EFFICIENCY, % PROPULSION EFFICIENCY, %
PUMPJET PERFORMANCE vs JET AREA
C.I.T. MAY, 1949 J.H. BEVERIDGE
SYMBOLS:
PROJECTILE SPEED = 80 K
ST'D JET AREA = 0.0431 ft²
100
115
130
PERCENT OF STANDARD JET AREA

Fig. 5

Discussion of Test Results

The characteristic curves were obtained for three contiguous regions of operation, namely, normal pump, power dissipation, and reverse turbine. All three flows were in the same direction, that is, from the suction nozzle to the discharge. The normal pump region of characteristic curves was used to obtain the steady state operating conditions of the pumpjet for the proposed projectile velocity of 80 knots. (See Appendix II). The steady state operating point is marked on the characteristic curves in Figs. 1 through 3. The steady state operating point indicated in Fig. 1 is based on an assumed drag coefficient of 0.08. It is seen that the operating point is at or very near the point of best efficiency.

The position of the unit operating point of the pumpjet is dependent upon the drag coefficient of the projectile. The term "unit operating point" refers to the operating point on the characteristic curves having coordinates of flow rate per 1000 rpm, head per $(1000 \text{ rpm})^2$ and brake horsepower per $(1000 \text{ rpm})^3$. A series of drag coefficients from 0.06 to 0.08 was assumed in the calculations and the pumpjet unit operating point corresponding to each drag coefficient was found and plotted in Fig. 2. It is to be noted in Fig. 2 that the position of the unit operating point does not vary greatly over the range of drag coefficients chosen. However, Fig. 4 shows that the pumpjet speed, flow rate, head, and brake horsepower do vary widely over the range of drag coefficients chosen. Thus the difficult and somewhat speculative determination of the correct drag coefficient is not as critical a problem in locating the pumpjet unit operating point as might at first be anticipated.

A change in the throat area of the discharge nozzles, and hence the area of the jets, also changes the pumpjet unit operating point. This effect is indicated in Fig. 5 for the same projectile speed of 80 knots and for various drag coefficients. The total throat area of the nozzles tested, denoted in this report as the standard jet area, was 0.0431 sq. ft on a diameter of 0.994 in. It can be concluded from Fig. 5 that the nozzles used were very close to the optimum for the given set of operating conditions.

It is of interest to compare the characteristic curves of the unit tested, which uses a slightly modified Granby model pump impeller, and the Byron Jackson Granby model pump. Comparison shows (Fig. 3) that the conversion from a normal single volute case to the diffuser jet case does not materially alter the general character of the characteristic curves. The fact that the test unit does not show as high a peak efficiency as the Granby model pump is not too surprising if it is noted that the surface area, hence the skin friction loss, of the diffuser jet case is considerably greater than that found in the single volute case.

Conclusions

1. The characteristic curves of the Mark 40 Pumpjet are very similar to the characteristic curves of a modern centrifugal pump.

2. The Mark 40 Pumpjet operating point is at or very near the point of maximum efficiency. There are no instabilities in the characteristic curves near the anticipated operating point.

3. The Mark 40 Pumpjet pump efficiency and propulsion efficiency are relatively insensitive to variation of projectile drag over the range of drag coefficients from 0.06 to 0.08.

4. The present total jet area (0.0431 sq. ft) is very satisfactory as far as the overall efficiency of the Mark 40 Pumpjet is concerned.

For detailed information as to the procedure employed in calculating the pump head, flow rate, brake horsepower, and efficiency from the test data, reference should be made to Appendix I which presents, in outline form, complete sample calculations of these quantities. In Appendix II is outlined the method used in estimating the pumpjet operating point for the proposed projectile speed of 80 knots.

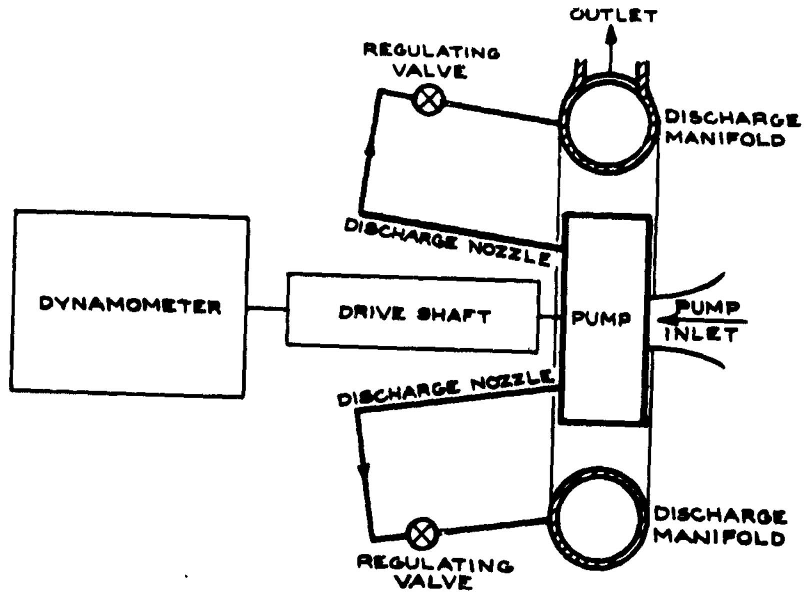

OUTLET
REGULATING VALVE
DISCHARGE MANIFOLD
DISCHARGE NOZZLE
DYNAMOMETER
DRIVE SHAFT
PUMP
PUMP INLET
DISCHARGE NOZZLE
DISCHARGE MANIFOLD
REGULATING VALVE

Fig. 6

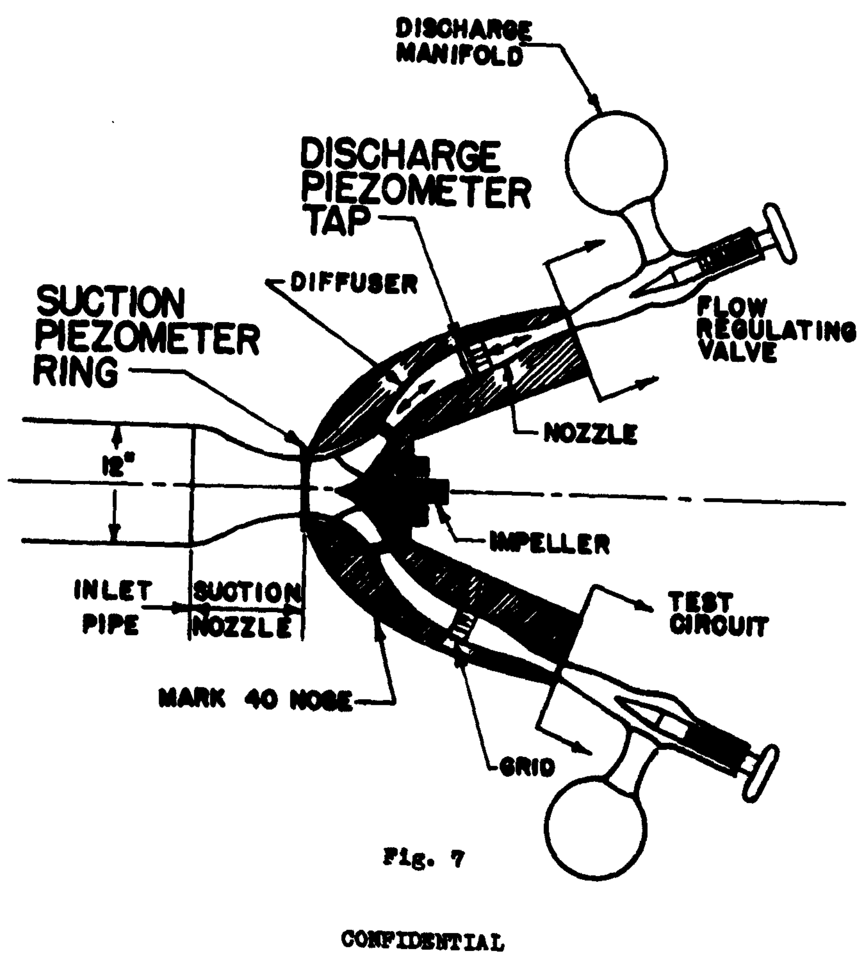

DISCHARGE MANIFOLD
DISCHARGE PIEZOMETER TAP
DIFFUSER
SUCTION PIEZOMETER RING
FLOW REGULATING VALVE
NOZZLE
12"
IMPELLER
INLET PIPE
SUCTION NOZZLE
TEST CIRCUIT
MARK 40 NOSE
GRID

Fig. 7

APPENDIX I

SAMPLE CALCULATIONS OF PUMPJET CHARACTERISTICS FROM TEST DATA

Run N-56-123-15, Point Number 27

The test data taken from the above run is introduced in the step of the calculations to which it pertains.

I. Pump Speed, N, rpm

Speed preset at 2200 rpm

Note: The speed is controlled by setting the speed control gear box to the nearest 0.5 rpm.

II. Torque

Torque = dynamometer lower torque reading	=	00.00
+ " upper " "	=	54.99
+ zero + windage + friction	=	-05.70
Total torque	=	49.29 ft lbs

Note: Windage and friction losses include pumpjet no-load bearing friction but not the John Crane seal friction.

III. Brake Horsepower, BHP

Total BHP input to pump

BHP = speed x torque x constant

$$BHP = 2200 \times 49.29 \times \frac{2\pi}{60 \times 550}$$

BHP = 20.64

Unit brake horsepower, bhp_{1000}, bhp per $(1000 \text{ rpm})^3$

$$bhp_{1000} = \frac{20.64}{\left(\frac{2200}{1000}\right)^3} = 1.94 \text{ bhp}/(1000 \text{ rpm})^3$$

BHP for speed n is

$$BHP_n = 1.94 \left(\frac{n}{1000}\right)^3$$

IV. Flow Rate, Q, cfs

Total Q

$$Q = \frac{C_d A_L}{144} \sqrt{\frac{2g \times \Delta q \times \left(\frac{\delta Hg}{\delta H_2O} - 1\right)}{\left(\frac{A_L}{A_T}\right)^2 - 1}}$$

Q = flow rate, cfs

C_d = coefficient of discharge of meter = 1.0

A_L = area of entrance end of meter = 26.0 in.^{2}

A_T = throat area of meter = 11.23 in.^{2}

Δq = differential head across meter, ft Hg

$\frac{\gamma_{Hg}}{\gamma_{H_2O}}$ = ratio sp. wt. Hg to sp. wt. H_2O = 13.6

g = gravitational acceleration = 32.2 ft/sec^2

For the above test

Δq = 1.008 ft Hg

$Q = \sqrt{6.05 \times \Delta q}$

Q = 2.47 cfs

Unit flow rate, q_{1000}, cfs per 1000 rpm

$$q_{1000} = \frac{2.47}{\frac{2200}{1000}} = 1.12 \text{ cfs/1000 rpm}$$

Flow rate at speed, n,

$$Q_n = 1.12\left(\frac{n}{1000}\right) \text{cfs}$$

V. Head Generated by Pump, H, ft of water

Total H

$$H = \frac{144(P_3 - P_2)}{\gamma} + \frac{V_3^2 - V_2^2}{2g} \quad \text{or, in terms of Q,}$$

$$H = \frac{144(P_3 - P_2)}{\gamma} + \frac{Q^2}{2g}\left[\frac{1}{A_3^2} - \frac{1}{A_2^2}\right]$$

where subscript 2 refers to the pump suction and 3 to a point just ahead of the straightener vanes in the nozzle passages. (See Fig. 7 and page 14.)

$P_3 - P_2 = \Delta P$, differential pressure across pump + gage correction, psi.

γ = sp. wt. of fresh water = 62.4 lbs/ft^3

For the above example

$$H = \frac{144(26.1 + 0.6)}{62.4} - (0.80)$$

H = 60.8 ft

Unit head, h_{1000}, ft per $(1000\ \text{rpm})^2$

$$h_{1000} = \frac{60.8}{\left(\frac{2200}{1000}\right)^2} = 12.56\ \text{ft}/(1000\ \text{rpm})^2$$

Head at speed, n,

$$H_n = 12.56 \left(\frac{n}{1000}\right)^2$$

VI. Pump Efficiency, η_p, per cent

$$\eta_p = \frac{WHP}{BHP}$$

WHP = water horsepower

$$\eta_p = \frac{\gamma QH \times 100}{550 \times BHP}$$

$$\eta_p = \frac{(62.4)(2.47)(60.8)(100)}{(550)(20.64)}$$

$$\eta_p = 82.7 \text{ per cent}$$

VII. Results

BHP = 20.64

bhp_{1000} = 1.94 bhp/$(1000\ \text{rpm})^3$

Q = 2.47 cfs

q_{1000} = 1.12 cfs/(1000 rpm)

H = 60.8 ft

h_{1000} = 12.56 ft/$(1000\ \text{rpm})^2$

η_p = 82.7 per cent

N = 2200 rpm

APPENDIX II

DETERMINATION OF THE MARK 40 PUMPJET OPERATING POINT[2]

I. Data

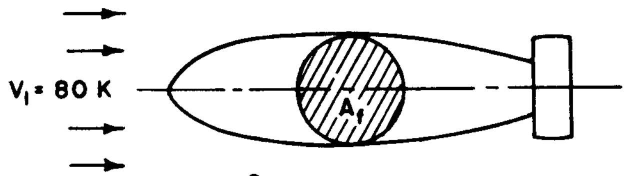
d = projectile dia. = 21 in.

d_t = discharge nozzle throat dia. = jet dia. = 0.994 in.

α = inclination of jets to axis of projectile = 15° .

V_1 = projectile velocity = 80 knots or 135 ft/sec.

II. Projectile Drag, D, lbs



$$D = C_D \times A_f \times \frac{\rho V_1^2}{2} \quad \text{where}$$

A_f = frontal cross sectional area of projectile

$= 0.785 \left(\frac{21}{12}\right)^2 = 2.40\ ft^2.$

ρ = mass density of fluid (fresh water)

$= \frac{62.4}{32.2} \frac{lb\text{-}sec^2}{ft^4}$

V_1 = projectile velocity = 135 ft/sec

C_D = drag coefficient. Assumed values are: 0.06, 0.07, 0.08.

These drag coefficients were chosen in lieu of drag studies in progress in the Hydrodynamics Laboratory water tunnel. In Ref. (2) C_D = 0.049 was used in preliminary calculations.

$$D = (0.08)(2.40)\left(\frac{62.4}{32.2}\right)\frac{(135)^2}{2}$$

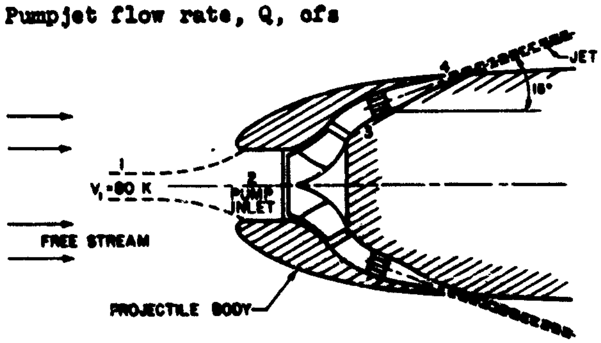
D = 3390 lbs for C_D = 0.08

D = 2970 lbs for C_D = 0.07

D = 2540 lbs for C_D = 0.06

C_D = 0.08 is used in the following calculations.

III. Pumpjet flow rate, Q, cfs



For steady state operation the axial thrust of the jets, T, is equal to the drag of the projectile, D.

From momentum considerations

$T = \rho Q \left[V_4 \cos\alpha - V_1\right]$ where

T = thrust, lbs

ρ = mass density, fresh water

Q = flow rate, cfs

V_4 = jet velocity, ft per sec

α = inclination of nozzles to projectile axis

V_1 = free stream velocity, ft per sec

In terms of Q

$$T = \rho Q \left[\frac{Q}{A_4} \cos\alpha - V_1\right] \text{ , or}$$

$$T = \frac{62.4}{32.2} \left[\frac{0.966}{0.0431} Q^2 - 135\ Q\right]$$

$$T = 43.5\ Q^2 - 262\ Q$$

For $T = D = 3390$ lbs

$Q = 12.3$ cfs

IV. Pumpjet head, H, ft

$$H = \left[\frac{P_3}{\gamma} + \frac{V_3^2}{2g} + Z_3\right] - \left[\frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2\right]$$

where $\frac{P}{\gamma}$ = static pressure, ft

$\frac{V^2}{2g}$ = velocity head, ft

Z = elevation, ft

Assume

Ram efficiency = 100 per cent

Discharge nozzle efficiency = 100 per cent

$$H = \left(\frac{P_4 - P_1}{\gamma}\right) + \left(\frac{V_4^2}{2g} - \frac{V_1^2}{2g}\right)$$

To estimate P_4, the pressure about the circumferential surface of the projectile in the vicinity of the discharge nozzle ports, reference was made to the experimental work by Lyons[5] in which the ratio of the pressure drop between a point in the

free stream, P_1, and a point on the body surface, P_4, to the free stream velocity head, $\rho V_1^2/2$,

$$\frac{P_4 - P_1}{\rho V_1^2/2} = \frac{p}{q} \quad \text{(Lyons' designation)}$$

was determined for various points along the profile of a body denoted as "Model A". The geometrical shape of the "Model A" nose section matches the profile of the Mark 40 torpedo nose section. From tabulated data in Table 8, on page 26, of the above reference, the average value of the ratio may be taken as -0.16. It is realized that this absolute value may be somewhat in error. The relatively undisturbed flow pattern about the test "Model A" and the pattern about the Mark 40 torpedo with jets in operation, is not exactly similar. However, as may be seen from the expression in the next step, a 50 per cent variation (-0.08 or -0.24 instead of -0.16) in the selected value introduces only a $\pm$ 2.5 per cent variation in the pump head under the given conditions. The above procedure is introduced as an indication of the proper procedure and is subject to modification as more applicable experimental data are made available.

Inserting in the previous expression for H,

$$\frac{P_4 - P_1}{\rho V_1^2/2} = -0.16 \quad \text{gives,}$$

$$H = \frac{V_4^2}{2g} - 1.16\frac{V_1^2}{2g} \quad \text{or, in terms of Q,}$$

$$H = \frac{Q^2}{2gA_4^2} - 1.16\frac{(135)^2}{2g} = 8.36Q^2 - 328$$

For Q = 12.3 cfs

H = 937 ft

V. Water horsepower, WHP

$$WHP = \frac{\gamma QH}{550}$$

$$WHP = \frac{(62.4)(12.3)(937)}{550}$$

WHP = 1308

VI. Thrust horsepower, THP

$$THP = \frac{\text{drag x projectile velocity}}{550}$$

$$THP = \frac{(3390)(135)}{550}$$

$$THP = 832$$

VII. Pump speed to satisfy items III and IV, N, rpm

$$\frac{\text{unit flow rate}}{\sqrt{\text{unit head}}} = \frac{\text{flow rate, cfs}}{\sqrt{\text{head, ft}}} = \frac{12.3}{\sqrt{937}} = 0.401$$

On the performance curves, Fig. 2, the ratio

$\frac{\text{unit flow rate}}{\sqrt{\text{unit head}}}$ is plotted against unit capacity for various points on the pump HQ curve. Entering this plot at $\frac{\text{unit flow rate}}{\sqrt{\text{unit head}}} = 0.401$ indicates the corresponding unit capacity

$$q_{1000} = 1.325 \text{ cfs/1000 rpm}$$

Thus

$$N = \frac{12.3}{\frac{1.325}{1000}} = 9283 \text{ rpm}$$

VIII. Pump operating point on unit HQ curve

Step VII above automatically locates the desired point.

Thus from performance curves

$$h_{1000} = 10.85 \text{ ft/(1000 rpm)}^2$$

$$bhp_{1000} = 1.91 \text{ bhp/(1000 rpm)}^3$$

$$\eta_p = 84.5 \text{ per cent}$$

It is interesting to check h_{1000} and bhp_{1000} against the H and BHP values previously determined.

$$H = (10.85)\left(\frac{9283}{1000}\right)^2 = 935 \text{ ft}$$

$$BHP = (1.91)\left(\frac{9283}{1000}\right)^3 = 1528$$

IX. Propulsion efficiency η_j, per cent

$$\eta_j = \frac{THP \times 100}{WHP} = \frac{(832)(100)}{1308}$$

$$\eta_j = 63.7 \text{ per cent}$$

X. Overall efficiency, η, per cent

$$\eta = \frac{WHP}{BHP} \times \frac{THP}{WHP} = \eta_p \times \eta_j = (84.5)(63.7)$$

$$\eta = 53.8 \text{ per cent}$$

XI. Specific speed of pump

$$n_s = \frac{N\sqrt{Q}}{H^{3/4}} = \frac{9180\sqrt{12.3}}{(937)^{3/4}}$$

$$n_s = 190 \text{ (Q in cfs)}$$

$$n_s = 4030 \text{ (Q in gpm)}$$

Fig. 8 - In the center of the discharge manifold, right foreground, may be seen the nozzle ring which contains the grid straightener vanes and affords a mounting for the jet nozzles. Some of the flow regulating valves are in place on the discharge manifold. The dynamometer is in the center background.

Fig. 9 - The Mark 40 diffuser jet pump case secured to the nozzle ring.

Fig. 10 - View from the drive end of the pumpjet. The nozzle ring with the discharge nozzles and their throat sections are visible behind the flow regulating valves.

Fig. 11 - Note the drive shaft in place
and the pressure line installation.

Fig. 12 - Laboratory setup for testing full scale pump for the pumpjet propulsion unit for the Mk 40 Torpedo.

BIBLIOGRAPHY

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