HYDRAULIC MODEL STUDIES
ON BULB TURBINE INTAKES

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Bureau of Reclamation
Intakes for bulb and rim generator turbines are very large in relation to their runner diameters. Because the water velocity is low in the intake area, the losses are small. This report describes research on the effect that simplifying the intake design has on energy losses and flow distribution. Using straight surfaces in place of curved bellmouth entrances and shortening the length of the intake could significantly reduce construction costs. In addition, reducing the intake size would reduce the size of trashracks, gates, and associated operating equipment.
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by
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GLOSSARY

\( A_r = \) flow area at the runner
\( C_p = \) pressure drop coefficient
\( C = \) coefficient of discharge
\( d = \) diameter
\( D = \) reference length
\( d/D = \) relative depth in intake
\( D1 = \) runner diameter
\( E = \) Euler number
\( E' = \) Velocity of approach factor
\( g = \) acceleration due to gravity
\( \Delta h = \) relative loss
\( H = \) gross head
\( h_w = \) pressure drop across orifice
\( K = \) flow coefficient
\( L/D1 = \) reference distance
\( p, p_1, p_2 = \) pressures
\( \Delta p = \) pressure drop
\( P_i = \) Inlet pressure
\( Q = \) flow rate, or discharge
\( R = \) Reynolds number
\( T_1 = \) ambient temperature
\( V = \) velocity
\( V_r = \) average intake velocity in the axial direction
\( V_\bar{V} = \) average velocity at the runner in the axial direction
\( Y = \) adiabatic expansion factor
\( \gamma = \) specific force = \( \rho g \)
\( \gamma_a = \) specific force of air
\( \theta = \) wicket gate angle
\( \nu = \) kinematic viscosity
\( \rho = \) density
PURPOSE

The purpose of this study was to investigate possible simplifications in the design of the intake flow passages for bulb turbines and to determine the head losses associated with these simplifications. Simplifying the design and construction of bulb turbine intakes could lead to cost savings in both material and labor.

INTRODUCTION

The main obstacle in the development of small hydroelectric powerplants has been economics; in most cases, the cost per installed kilowatt for small hydropower is still higher than for fossil fuel plants. For low-head hydroelectric installations, with head less than 20 m, the major costs are the initial investment in the civil works structure and the fluid machinery. If the cost of the structure can be reduced without introducing additional head losses, more small hydroelectric installations would be feasible.

Intakes for bulb and rim generator turbines are very large in relation to their runner diameters. Because the water velocity is low in the intake section, the losses are small. It was concluded in an earlier literature review [1] that savings could be achieved by replacing curved surfaces with straight surfaces and shortening the length of the intake. Reducing the intake size would result in additional savings in trashracks, bulkheads, entrances, gates, and the associated operating equipment.

Another small-hydropower reference [2] states that "Irregularities of flow as well as flow separations in the intake section have an unfavorable effect on the turbine's hydraulic behavior, and an optimum design for the intake portion is therefore essential for smooth, undisturbed turbine performance." However, the data presented herein demonstrate that simplifications in the intake section do not adversely affect the flow field leading to the guide vanes and runner.

CONCLUSIONS

- Significant simplifications and size reductions can be made in the intakes of bulb and rim generator turbines without increasing energy losses or adversely affecting flow distribution.
- Comparative energy losses and velocity distributions illustrate the advantages of using simplified intake designs.
- Structural costs for a bulb turbine structure using a simplified intake design (intake 4) would be about 10 percent less than the present standard design (intake 1).

SCOPE OF STUDY

A model of a typical bulb turbine installation was built in such a manner that the intake section could be readily removed to permit comparison of other intake shapes with the conventional shape. The model dimensions basically corresponded to standard flow passage dimensions used by a major manufacturer. After testing the original intake, three other intakes of various shapes and sizes were tested and the results compared. Dimensions are given in terms of the runner diameter, $D_1$.

Extensive testing was done on the original intake to determine the effect of a bulkhead slot, a pier in the intake, a draft tube, and various approach channel configurations.

MODEL DESIGN AND SIMILARITY

An air model was used in this study. Advantages of an air model over a water model include: (1) flexible, easy model construction, (2) little problem of leakage, and (3) quick model measurements and changes. The disadvantages are: (1) small levels of pressure differences, requiring delicate measuring apparatus; and (2) inability to simulate a free surface. Air models can be used to study hydraulic problems in which the flow is governed by inertia and viscosity effects [3]. Conditions of flow at an entry and flow through pressurized conduits fall within this category. The criterion of similarity for this type of flow and for transferring results to prototype conditions is well known to be the Reynolds model law (equation 1).

\[ Reynolds \text{ number} = R = \frac{VD}{\nu} \] (1)

where:

- $V = \text{velocity}$
- $D = \text{a characteristic length}$
- $\nu = \text{kinematic viscosity}$
Most prototype hydraulic structures have Reynolds numbers of the order of magnitude of $10^6$ to $10^8$. The achievement of these Reynolds numbers under laboratory conditions would require blowers with enormous capacity. It is possible, however, to attain approximate similarity at Reynolds numbers of about $10^4$ to $10^6$. At these Reynolds numbers, viscosity has little effect. It is also necessary to build the model large enough to avoid undesirable compressibility effects [3]. The model was designed to keep the Reynolds number as high as possible while limiting the air velocity to less than $50$ m/s to avoid compressibility effects.

The Euler number is a dimensionless ratio which relates inertia forces to pressure forces (equation 2).

$$ \text{Euler number} = E = \frac{\rho V^2}{\Delta p} $$

where:

- $\rho$ = fluid density
- $\Delta p$ = pressure drop
- $V$ = velocity

In incompressible fluids and in the absence of other forces (such as viscosity and gravity), the Euler number is exclusively a function of the geometry of the flow boundaries [3]. At Reynolds numbers high enough to attain similarity, the Euler number is a constant. Therefore, the Euler number will be the same for any prototype size as it is in the model if the geometry is similar. For this reason, the Euler number is also referred to as the geometrical flow number [3].

The first set of tests was conducted to determine the minimum discharge to obtain similarity. Tests were run at different discharges for the same model configuration. At Reynolds numbers greater than $10^5$, the Euler number is approximately constant (fig. 1). Therefore, all of the tests were conducted at Reynolds numbers greater than $10^5$. The Euler number is then used to scale results from the model to prototype conditions.

![Figure 1.—Euler number vs. Reynolds number.](image)

**THE MODEL**

Figure 2 is a schematic diagram of the test apparatus and figure 3 shows an overall view of the apparatus itself. Air was supplied by a blower through a supply line and orifice plate to a stilling chamber (plenum), where it entered the model intake section.

![Figure 2.—Schematic diagram of the test apparatus.](image)
Figure 4 shows the configuration of the bulb turbine model with the conventional intake section (intake 1). All four intake designs used in this study are shown on figure 5. The intake sections were made of sheet metal, and piezometer taps were included along the top, bottom, and sides of each intake to measure the pressure drop along these surfaces (fig. 5). A pressure tap was also included in the plenum to measure the total pressure required to produce a given flow through the model.

**Discharge Measurement**

The rate of flow, or discharge, was measured with a concentric, thin-plate orifice located in the supply line (figs. 2 and 6). Flange taps just upstream and downstream from the orifice plate were used to obtain the pressure differential across the plate.

Equations for computing actual rates of flow through an orifice plate are found in *Fluid Meters, Their Theory and Application* [4]. Flow rate, Q, was computed as follows:

\[
Q = 18.794 Kyd^2 \sqrt{\frac{T_1}{h_w P_1}}
\]

where:

- \( Q \) = flow rate (m³/s)
- \( Y \) = adiabatic expansion factor
- \( h_w \) = pressure drop across orifice (kPa)
- \( d \) = diameter of orifice (m)
- \( P_1 \) = inlet pressure (kPa - absolute)
- \( T_1 \) = temperature (kelvins)
- \( K \) = flow coefficient = \( CE' \)
- \( C \) = coefficient of discharge
- \( \beta \) = diameter of orifice/diameter of pipe
- \( E' \) = velocity of approach factor = \( \frac{1}{\sqrt{1 - \beta^4}} \)

For \( K = 0.622 \) (from equation 1-5-76 in [4]), \( d = 0.1222 \) m, \( Y = 0.991 \) (from equation 1-5-50 in [4]), and \( \beta = 0.483 \), equation 3 can be approximated by:

\[
Q = 0.1730 \sqrt{h_w P_1} \]

**TEST PROCEDURES AND INSTRUMENTATION**

Data collected during the tests included: (1) total discharge through the model; (2) velocity profiles immediately upstream from the bulb (at the bulkhead slot in fig. 4); (3) pressures along the top, bottom, and sides of the intake; and (4) plenum and atmospheric pressures.
SECTION THROUGH BULB TURBINE MODEL

SECTION A-A

18-WICKET GATES AT 20° SPACING

WICKET GATE OPENING DETAIL

SECTION C-C

Figure 4.—Bulb turbine model (with intake 1 shown).
Figure 5.—The four intakes tested. (D1=runner diameter=155 mm)
Figure 6.—Closeup views of the model.
3. Record the inlet pressure, $P_i$
4. Record differential pressure across the orifice plate, $h_w$
5. Compute the flow rate, $Q$
6. Record velocity or pressure data

**Velocity Measurement**

Velocity measurements were made for each intake at various wicket gate openings. Measurements were made at the bulkhead slot location immediately upstream from the bulb (fig. 5) using a hot-wire, constant-temperature anemometer (fig. 7). For each test, 117 velocity measurements were taken in a 9-by-13-point grid using a telescoping probe attached to the self-contained instrument, and the readings were recorded manually. The hot-wire anemometer readings were checked by measuring known velocities through an orifice in the side of the stilling chamber.

**Pressure Measurement**

Pressure differences in an air model are small. Therefore, sensitive and accurate instruments are needed to collect acceptable data.

Pressure taps from the orifice plate, the intake, and the plenum were all connected to a single scanning-type pressure sampling valve for measuring multiple pressures. With this system, one pressure transducer was used to measure all the pressures. A manually operated step drive was used to connect each pressure line sequentially to a center port in the valve (fig. 8), and the center port was connected to a ±6.895-kPa differential pressure transducer having a total error of 0.06 percent (combined linearity and hysteresis). The opposing side of the transducer was open to atmospheric pressure. Calibration pressures were applied to the transducer with a known height water column. Figure 9 shows the calibration curve. The lack of scatter in the data around the calibration curve demonstrates the accuracy of the transducer.

Transducer excitation was 12 V d.c., and the output was amplified 1000 times with a high-gain data amplifier. The amplified output was in the range of -5 to +8 volts. Output voltages, read with a high-speed DVM (digital voltmeter), were collected and stored using a data acquisition system. A microprocessor (fig. 3) was programmed to read average and fluctuating voltages from the DVM. The voltages were converted to pressures and, when the test was complete, the data were transferred from memory to cassette tape for later printing, plotting, and analysis. Atmospheric pressure was recorded before and after each test to ensure that the reference pressure did not vary substantially during the test.
RESULTS

Velocities

Velocity distributions for a given intake design were very similar when the wicket gate angles (θ) were less than 45°. For 45° and 60° wicket gate angles, the profiles were very erratic and unsymmetrical.

Velocity distributions were plotted for the four intakes for θ = 30° (fig. 10). The average velocity, \( \bar{V} \), was different for each intake; however, flow distribution is not affected by the actual values of velocity if the Euler number is constant. (See the section on similarity.)

It is obvious from comparing the velocity contours that the intake shape has a significant effect on velocity distribution. (See fig. 5 for intake shapes.) The velocity distributions for all four intakes show the flow stagnating in front of the bulb and flowing around it to the sides.

In intake 1, the velocities were high near the top, due to the smooth, bellmouth-type top curve, and low near the bottom, with a steep transition from top to bottom.

In intake 2, the velocities were fairly uniform throughout, and the local velocities did not vary greatly from the average velocity.

Intake 3 did not have entrance curves and the corners were square. The effect of the square corners is apparent in the velocity profiles. Velocities were high through the center portion of the profile and very low at the edges and corners.

Flow distribution in intake 4 was fairly uniform. The pattern did not indicate separation as in intake 3, nor a steep gradient from top to bottom as in intake 1. The overall velocity distribution was the most uniform in intake 4.

Velocity profiles from top to bottom along data column 3 in figure 10 are plotted in figure 11. This graph illustrates the differences in flow distribution among the four intakes in the zone where most of the flow passes around the bulb.

It should be noted that the average velocity head, \( \bar{V}^2/2g \), in the intake section is usually only about 1 percent of the total prototype head. Therefore, flow irregularities in the intake section should have relatively minor effect on losses through the structure.

Pressures

Euler number accuracy. — An indication of the accuracy of the Euler numbers is needed in order to determine if the variations are due to geometry changes or data scatter. Therefore, several runs were made for the same geometry at different discharges (runs 101 through 116). The Euler numbers varied by ±0.015; therefore, changes greater than 0.015 are due to geometry effects rather than data scatter.

Overall losses through the model. — Pressure drops through the entire model as well as pressures along the flow surfaces in the intake section were measured for each test run. Table 1 lists the Euler numbers for the different configurations tested. These data can be used to assess the effect of changes in geometry on pressure drops (losses) through the model. The table also defines the model configuration for each run number.

The reference velocity used to calculate the Euler numbers in table 1 was the average axial velocity at the runner, \( \bar{V}_r \):

\[
\bar{V}_r = \frac{Q}{A_r} \tag{5}
\]
Figure 10.—Velocity distributions for the four intakes ($\theta=30^\circ$).
Equation 7 shows that the relative losses are proportional to velocity head. Therefore, actual losses will be more for higher discharges, if the Euler number is the same.

**Example calculation of relative losses.** — For a hypothetical prototype installation the following values are given:

Runner diameter — \( D_1 = 3.51 \text{ m} \)

Area at the runner — \( A_r = 8.03 \text{ m}^2 \)

Gross prototype head — \( H = 13.52 \text{ m} \)

Discharge relationship—

\[
\begin{align*}
\text{at } \theta = 0^\circ, & \quad Q = 127 \text{ m}^3/\text{s} \\
\text{at } \theta = 15^\circ, & \quad Q = 85 \text{ m}^3/\text{s} \\
\text{at } \theta = 30^\circ, & \quad Q = 43 \text{ m}^3/\text{s}
\end{align*}
\]

Since \( V_r = Q/A_r \) (equation 5),

\[
\begin{align*}
\text{at } \theta = 0^\circ, & \quad V_r = 15.82 \text{ m/s} \\
\text{at } \theta = 15^\circ, & \quad V_r = 10.58 \text{ m/s} \\
\text{at } \theta = 30^\circ, & \quad V_r = 5.36 \text{ m/s}
\end{align*}
\]

The Euler numbers in table 1 can be used to compute the relative head losses—using equation (7). Table 2 gives relative head losses for this example using intake 1 as a reference.

Table 2 shows that, for this example, intake 1 has about 1.5 percent less loss than the other intakes when the wicket gates are fully open (\( \theta = 0^\circ \)). However, when the gates are partially closed (\( \theta = 15^\circ, 30^\circ \)), intake 4 has less loss than intake 1. Intake 2 has less loss at \( \theta = 30^\circ \). This is consistent with the velocity comparisons (figs. 10 and 11) showing that intakes 2 and 4 have a more uniform velocity distribution than intake 1.

**Pressure drop coefficients.** — Another form of the Euler number is the pressure drop coefficient,
Table 1. — Pressure drop through the model (Euler numbers)

<table>
<thead>
<tr>
<th>Wicket gate angle (degrees)</th>
<th>Intake 1</th>
<th>Intake 2</th>
<th>Intake 3</th>
<th>Intake 4</th>
<th>Intake 1 with pier</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>4.321</td>
<td>149</td>
<td>4.178</td>
<td>166</td>
<td>4.195</td>
</tr>
<tr>
<td>20</td>
<td>2.360</td>
<td>162</td>
<td>2.349</td>
<td>171</td>
<td>—</td>
</tr>
<tr>
<td>25</td>
<td>1.728</td>
<td>163</td>
<td>1.754</td>
<td>172</td>
<td>—</td>
</tr>
<tr>
<td>30</td>
<td>1.297</td>
<td>151</td>
<td>1.322</td>
<td>168</td>
<td>1.290</td>
</tr>
<tr>
<td>35</td>
<td>.925</td>
<td>164</td>
<td>.967</td>
<td>173</td>
<td>—</td>
</tr>
<tr>
<td>40</td>
<td>.683</td>
<td>165</td>
<td>.709</td>
<td>174</td>
<td>—</td>
</tr>
<tr>
<td>45</td>
<td>.499</td>
<td>152</td>
<td>.500</td>
<td>169</td>
<td>.506</td>
</tr>
<tr>
<td>60</td>
<td>.153</td>
<td>153</td>
<td>.168</td>
<td>170</td>
<td>—</td>
</tr>
</tbody>
</table>

Intake 1 without draft tube

<table>
<thead>
<tr>
<th>Wicket gate angle (degrees)</th>
<th>With both sides</th>
<th>With both sides (lowered)</th>
<th>With one side</th>
<th>With bottom and no sides</th>
<th>With bulkhead slot and both sides</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>E</td>
<td>Run No.</td>
<td>E</td>
<td>Run No.</td>
<td>E</td>
</tr>
<tr>
<td>0</td>
<td>1.810</td>
<td>120</td>
<td>1.813</td>
<td>129</td>
<td>1.814</td>
</tr>
<tr>
<td>15</td>
<td>1.648</td>
<td>121</td>
<td>1.650</td>
<td>130</td>
<td>1.644</td>
</tr>
<tr>
<td>20</td>
<td>1.465</td>
<td>125</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>25</td>
<td>1.210</td>
<td>128</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>30</td>
<td>.984</td>
<td>122</td>
<td>.985</td>
<td>131</td>
<td>.997</td>
</tr>
<tr>
<td>35</td>
<td>.768</td>
<td>127</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>40</td>
<td>.567</td>
<td>128</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>45</td>
<td>.411</td>
<td>123</td>
<td>.413</td>
<td>132</td>
<td>—</td>
</tr>
<tr>
<td>60</td>
<td>.143</td>
<td>124</td>
<td>.140</td>
<td>133</td>
<td>—</td>
</tr>
</tbody>
</table>

Model with draft tube

<table>
<thead>
<tr>
<th>Wicket gate angle (degrees)</th>
<th>Intake 3</th>
<th>Intake 4 with no approach channel</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>E</td>
<td>Run No.</td>
</tr>
<tr>
<td>0</td>
<td>4.137</td>
<td>175</td>
</tr>
<tr>
<td>15</td>
<td>3.709</td>
<td>176</td>
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<tr>
<td>30</td>
<td>1.384</td>
<td>177</td>
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<tr>
<td>45</td>
<td>.519</td>
<td>178</td>
</tr>
<tr>
<td>60</td>
<td>.163</td>
<td>179</td>
</tr>
</tbody>
</table>

*E = Euler number

\[E = \rho V^2 / \Delta p; \text{ where } \rho = \text{density}; \ V = \text{velocity at the runner}; \ \Delta p = \text{pressure drop through the model (plenum pressure minus atmospheric pressure)}\]

Table 2. — Comparison of intake losses

<table>
<thead>
<tr>
<th>(\theta = 0^\circ)</th>
<th>(\theta = 15^\circ)</th>
<th>(\theta = 30^\circ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\Delta h(m))</td>
<td>(\Delta h/H(%))</td>
<td>(\Delta h(m))</td>
</tr>
<tr>
<td>Intake 2</td>
<td>0.202</td>
<td>1.5</td>
</tr>
<tr>
<td>Intake 3</td>
<td>.178</td>
<td>1.3</td>
</tr>
<tr>
<td>Intake 4</td>
<td>.210</td>
<td>1.6</td>
</tr>
</tbody>
</table>

\(\theta = \text{wicket gate angle} (0^\circ = \text{fully open})\)

\(\Delta h = \text{difference in head loss from intake 1 (relative loss)}\)

\(H = \text{gross head} = 13.52 \text{ m}\)
The pressure drop coefficient is the ratio of drop in pressure head to a reference velocity head (equation 8).

\[ C_p = \frac{\Delta p}{\frac{1}{2} \rho V^2} \]  \hfill (8)

where \( g \) = gravitational acceleration
\( \gamma \) = specific force = \( \rho g \)

It can be shown by combining equations (2) and (8) that:

\[ C_p = \frac{2}{E} \]

and equation (7) becomes

\[ \Delta h = \frac{V^2}{2g} (C_{p2} - C_{p1}) \]  \hfill (9)

Plotting \( C_p \) allows observation of losses in terms of a reference velocity head. Figure 12 is a comparison of pressure drop coefficients for the four intakes. The head required to move the flow through the model (for \( \theta = 0^\circ \)) is about 48 percent of the velocity head at the runner (\( C_p = 0.48 \)). The plot illustrates the relative importance of intake losses to the overall losses. Intake 3, with no entrance curves, has the highest losses. However, even with no attempt to streamline the entrance, the losses in intake 3 are not significantly higher than in the other intakes. Intakes 2 and 4, with simplified and shortened entrances, show lower losses for partial gate openings than the traditional bellmouth-type design (intake 1), and they have a more uniform velocity distribution.

Additional testing. — Extensive testing of intake 1 was performed to determine the effect of other geometric features, including:

- A bulkhead slot
- A center pier in the intake
- The approach channel configuration
- A draft tube

Figure 13 shows intake 1 with a center pier and the approach channel with a bottom and no sides. Figures 14 through 17 illustrate the effect of these geometric features. Figure 14 is a comparison plot showing that the bulkhead slot has essentially no effect on losses. This figure also illustrates the repeatability of the data. A pier in the intake (fig. 15) and the configuration of the approach channel leading to the intake (fig. 16) have little effect on the overall losses. However, the approach channel did have a significant effect in one case: The Euler numbers for intake 4 without an approach channel are lower than intake 4 with an approach channel bottom, except for \( \theta = 60^\circ \). (See table 1, runs No. 192-196 vs. 197-201.) Tests on intake 1 did not show a significant effect due to the approach channel (runs No. 101-139). This difference can be explained by referring to figure 11. The relative velocity at the bottom of intake 1 is very low compared to intake 4. Therefore, the approach channel (which guides the flow into the bottom of the intake, preventing flow separation along the bottom) would be more important in intake 4 than in intake 1.

![Pressure drop through the model](image)

Figure 12.—Pressure drop through the model—comparison of four intakes.

Although the draft tube does not affect intake losses, tests were run with and without a draft
prototype situation the intake velocity head is about 1 percent of the gross head.

The reference distance \((L/D_1)\) is the distance from the front face of the intake to the piezometer (fig. 2).

The average pressure at the end of the intake section is an indication of the losses to that point. The average pressure drop coefficients at the end of the intakes were \(C_{P_1} = 1.216, C_{P_2} = 1.200, C_{P_3} = 1.497, \) and \(C_{P_4} = 1.183,\) for intakes 1, 2, 3, and 4 (from figs. 18 through 21). After subtracting the velocity head from the pressure drops, the energy losses—in terms of intake velocity heads—were 0.216, 0.200, 0.497, and 0.183 for intakes 1 through 4, respectively. This indicates that intake 3 has more than twice as much loss as the others and intake 4 has the lowest losses. It should be noted that the intake velocity head is only 2 percent of the runner velocity head. This puts the intake losses in perspective with the overall losses discussed previously.

Figures 22 through 25 compare pressures along the same surface for the four intakes. These figures illustrate that intake 3 has the highest local pressure drops, and intake 4 brings the pressure drop down in about one-half the distance of the other intakes, without high local pressure drops.

**Intake pressure data accuracy.** — The intake section pressures shown on figures 12 and 14 through 25 are average pressures. Figure 26 shows pressures recorded during one test, illustrating a typical range of pressure fluctuations for intake 1. Figure 27 shows the average pressures for three separate tests with similar conditions (see table 1 for test conditions). This figure illustrates the repeatability of the average pressure data in the intake section.

**Wicket Gate Angle vs. Percent Gate Opening**

Figure 28 is a cross reference for wicket gate angle, \(\theta\), vs. percent gate opening, where percent gate opening is defined as: open area/open area when gates are fully open. This figure should be useful in computing losses when the discharge relationship is given in terms of percent gate opening. If percent gate opening is defined as percentage of the full range of \(\theta\), the gate opening (in percent) is \((1-\theta/75)\) 100.
Figure 14.—Pressure drop through the model—intake 1, with and without a bulkhead slot.

Figure 15.—Pressure drop through the model—intake 1, with and without a center pier in the intake.

Figure 16.—Pressure drop through the model—intake 1, approach channel comparison.

Figure 17.—Pressure drop through the model—intake 1, with and without a draft tube.
Figure 18.—Pressure drop along intake surfaces—intake 1.

Figure 19.—Pressure drop along intake surfaces—intake 2.

Figure 20.—Pressure drop along intake surfaces—intake 3.

Figure 21.—Pressure drop along intake surfaces—intake 4.
PRESSURE DROP COMPARISON TOP CENTERLINE
WICKET GATE OPENING=15 DEGREES

○ = INTAKE #1-RUN #158
△ = INTAKE #3-RUN #187
□ = INTAKE #4-RUN #198

Figure 22.—Pressure drop comparison, top centerline—four intakes.

PRESSURE DROP COMPARISON RIGHT SIDE CENTERLINE
WICKET GATE OPENING=15 DEGREES

○ = INTAKE #1-RUN #158
△ = INTAKE #3-RUN #187
□ = INTAKE #4-RUN #198

Figure 24.—Pressure drop comparison, right side centerline—four intakes.

PRESSURE DROP COMPARISON LEFT SIDE CENTERLINE
WICKET GATE OPENING=15 DEGREES

○ = INTAKE #1-RUN #158
△ = INTAKE #3-RUN #187
□ = INTAKE #4-RUN #198

Figure 23.—Pressure drop comparison, left side centerline—four intakes.

PRESSURE DROP COMPARISON BOTTOM CENTERLINE
WICKET GATE OPENING=15 DEGREES

○ = INTAKE #1-RUN #158
△ = INTAKE #3-RUN #187
□ = INTAKE #4-RUN #198

Figure 25.—Pressure drop comparison, bottom centerline—four intakes.
Figure 26.—Pressure fluctuations, intake 1, top centerline.

Figure 27.—Pressure drop comparison, intake 1, data repeatability.

Figure 28.—Percent gate opening vs. wicket gate angle.
BIBLIOGRAPHY


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