PAP-453
DOE-79-01

BUREAU OF RECLAMATION
HYDRAULICS BRANCH

STANDARDIZED INTAKES AND

OUTLETS FOR LOW-HEAD

HYDROPOWER DEVELOPMENTS

BY

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DOE-79-01
PAP-453

ACKNOWLEDGMENTS

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ABSTRACT

This report presents the results of a state-of-the-art review
performed to investigate the present status of standardization of
flow passage design for low-head hydropower developments. The
current design practices regarding flow passages are summarized and
examined to determine if improvements could be made to reduce losses
or simplify present design without introducing significant additional
losses.

Several standard "off the shelf" units are currently available in
the low-head range (less than 20 m). Standard axial flow tube and
bulb units and cross flow turbines can be obtained in the range from
50 kW to 5000 kW. These predesigned units reduce equipment costs by
eliminating the need for site-specific engineering and by using
standardized manufacturing techniques. However, these savings are
often obtained at the expense of a slight loss in unit efficiency.
Manufacturer's drawings with standard flow passage dimensions for
bulb and rim-generator units are included.

Most designers and consultants feel that standardization of flow
passage design is not feasible. Structural and geological considera-
tions often affect the design. However, a manual to aid in the
planning and design of flow passages is considered to be worthwhile.
An alternative to standardization of flow passages in general would
be to develop a standardized design for a series of powerplants to
be installed at similar locations. An example of this approach is
the proposed harnessing of the Rhine River upstream of Lake Constance.
Research regarding the effect of intake shape, boundary layer
control in the draft tube, and rate of rise of the exit channel in
the tailrace may lead to simplified designs and reduced construction
costs. Fifty-two references on the design of flow passages and
the advantages of low-head hydropower are cited.

Key words and identifiers: flow passage design/ intakes/ draft
tubes/ vortices/ standardization/ low-head hydropower/ axial flow
turbines/ standardized designs/ forebay/ tailrace/ bulb turbines/
tube turbines/ rim generator turbines/ turbine setting

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I. PURPOSE AND SCOPE

A. Purpose

The objective of standardization is to reduce the cost of low-head hydropower. The main purpose of this phase of the study was to determine the present status of standardization of flow passages and to determine if standardization is a feasible means of reducing the cost of low-head structures. Another objective was to examine current design practices to identify areas where design changes or further research could result in less costly structures.

The need for additional energy production combined with diminishing reserves of fossil fuels has resulted in a need to develop the remaining hydropower capacity in the United States. There are many existing impoundments and completely undeveloped sites that could be developed with little effect on the environment. There are also many sites in need of updating. Additional generating units and replacement of obsolete equipment could significantly increase power generation from these sites. At the same time coal-fired and nuclear powerplants face an uncertain future because of environmental concerns and lengthy licensing procedures. Low-head hydropower could make a substantial contribution to help gap the predicted energy shortage in the next 10 years.

The main obstacle to development of low-head sites (heads below 20 m) has been economics. The cost per kWh for low-head hydropower is still slightly higher than is for fossil fuel plants. However, with the cost of fuels continuing to rise, the low-head hydropower alternative is becoming more favorable.

In low-head plants, the head losses could reduce the net effective head by a significant portion of the total available energy. Therefore, it would be desirable to streamline the flow passages as much as possible to minimize the head losses. At the same time, economics dictate that the flow passage shape be as simple as possible. With these conflicting interests in mind, the present design methods were examined to determine possible design improvements to further reduce losses or to simplify present designs without introducing significant additional losses.

The design of penstock entrances for high-head dams is one example that illustrates the value of taking a closer look at conventional design practices. It was conservatively estimated that $13 million was saved in construction costs on the penstock entrances for the Third Powerplant at Grand Coulee by reducing the size of the bellmouth entrances (reference 1). Penstock entrances had historically been designed using the same criteria used in designing high-velocity conduits

while the velocity in penstocks is much lower. Hydraulic model studies indicated that losses were actually lower in the smaller entrance. The same criteria used to design high-velocity conduits are used to design intake shapes for low-head plants. Therefore, savings may be possible by changing design methods in low-head structures.

B. Scope

The work during the first year of this study included a literature review and contacts with manufacturers, consultants, and operators of low-head turbines.

The flow passage studies included:

1. The forebay - which controls the flow approaching the intake.

2. The intake - which accelerates the flow to the turbine.

3. The draft tube - which decelerates the flow to recover the kinetic energy in the flow.

4. The tailrace - which carries the flow away from the turbines.

II. CONCLUSIONS AND RECOMMENDATIONS

A. Conclusions

1. Complete standardization of flow passages for low-head hydropower developments is not feasible. Structural and geological considerations may cause design variations from site to site. This would cause the flow passage designs to vary. The differences would be more evident in large installations.

2. "Off the shelf" units are presently available for small plants. These units have standard flow passages included. The low cost resulting from standardization is achieved at the expense of some loss in efficiency. However, the cost savings are significant, especially if several installations are considered as a group.

3. Standardized designs are feasible for similar site characteristics. For large installations, these standard designs should be as efficient as practicable, since in low-head plants the hydraulic losses represent a signficant portion of the total available head. Where geologic and structural considerations do not vary widely the same design may be used for a series of installations thus reducing the costs associated with site-specific engineering and design.

4. The hydraulic design of the draft tube is customarily determined by the manufacturer, since it is considered an integral part of the turbine in determining turbine performance. Results of model tests done by manufacturers optimizing flow passage dimensions are not publicly available. However, modern techniques of boundary layer control may be useful in shortening draft tubes, thus considerably reducing construction costs. The possibility of reducing draft tube lengths through boundary layer control has not been investigated.

5. Intake shape is typically determined using design criteria for high velocity conduit entrances. It is possible that intake shapes may be reduced in size or simplified without a significant loss in efficiency.

6. The effect on turbine efficiency of the rate of rise in the channel bottom in the tailrace is not known. A steep rise may cause flucuations in the tailwater level and reduce turbine efficiency.

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B. Recommendations

1. Investigate the effect that simplifying the intake shape and varying the rate of rise in the channel bottom might have on the hydraulic losses in low-head turbine flow passages.

2. Investigate the effect on the contraction coefficient of the shape of the flow separating pier between the powerhouse and weir and the pier between adjacent bays in the powerhouse.

3. Investigate the use of boundary layer control in draft tubes to shorten the length of draft tube required while maintaining hydraulic efficiency.

4. Evaluate the potential economic benefits which could result from design changes in the intake, draft tube, and tailrace.

5. Prepare a design manual to serve as a guide in the design of intake and exit structures for various types of low-head turbines under various operating conditions. This guide would promote the use of low-head hydropower and possibly present alternate design methods which could reduce the cost of the civil works. Some manufacturers and designers may resist a publication presenting design methods that conflict with their present practices. However, if economically marginal projects are made feasible by cost reductions in the civil works the turbine manufacturers and designers could benefit from the additional business that would be generated.
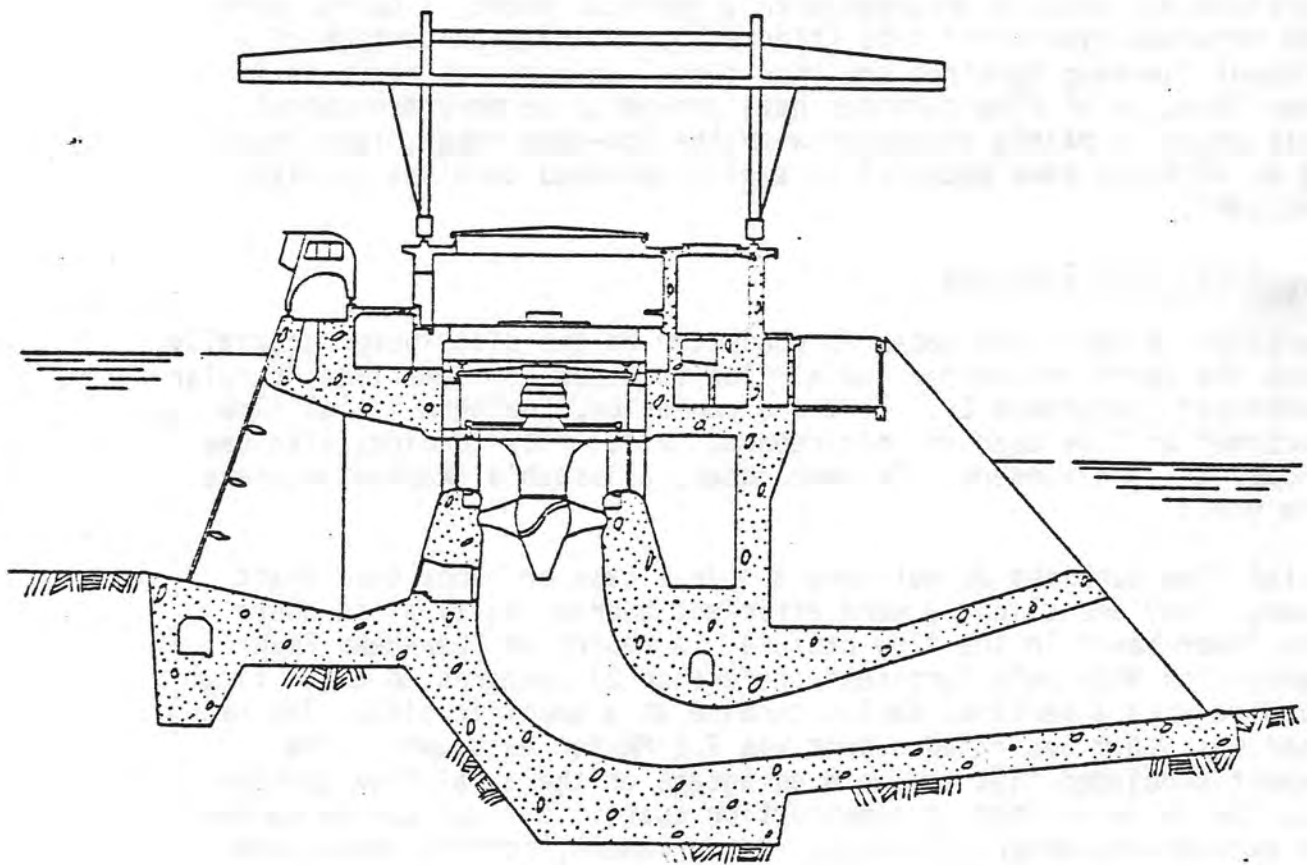
VERTICAL KAPLAN TURBINE

ESCHER WYSS
FIGURE I

III.   INTRODUCTION

A. Kaplan Turbines

In the head range between 3 and 45 m, turbines with propeller-type
runners are typically used (reference 2). If the runners are
adjustable, the turbines are called "Kaplan turbines." These
turbines are usually arranged with a vertical shaft, a spiral case,
and an elbow-type draft tube (figure 1). A large percentage of
present low-head turbines are this type. However, at heads less
than 20 m, axial flow turbines have proved to be more economical.
This study is mainly concerned with the low-head range, less than
20 m, although some material on small high-head turbines is also
included.

B. Axial Flow Turbines

Turbines in which the water is conducted to the distributor coaxially
with the shaft are called "axial flow turbines" or sometimes "tubular
turbines" (reference 2). To avoid confusion, the term "axial flow
turbine" will be used in this report. Axial flow turbines also use
propeller-type runners. In some cases, adjustable (Kaplan) runners
are used.

Axial flow turbines do not have a spiral case or elbow-type draft
tube. They are slightly more efficient hydraulically since there
are fewer bends in the flow passage. A report on "Low-Head Power
Generation With Bulb Turbines" (reference 2) compares an axial flow
turbine with a vertical Kaplan turbine at a specific site. The net
head was 5.8 m and rated output was 7.2 MW for each unit. The
report concluded that the main advantage of the axial flow turbine
was the large savings in construction costs. Savings can be gained
in excavation, concrete volume, reinforcement, forming costs, and
construction time. The overall construction savings for this site
were estimated to be 50 percent of the cost of the civil works. The
report also concluded that the conventional Kaplan turbine is more
economical for heads above 18 m. The advantages of the axial flow
unit increase with decreasing head. The three primary types of
axial flow units are described in the following paragraphs: (1) the
rim-generator turbine, (2) the tube turbine, and (3) the bulb
turbine.

    1. Rim-generator turbine. - The first axial flow unit, the
    rim-generator, was invented in 1919 by L. F. Harza (reference 3).
    The generator is attached to the periphery of the runner and lies
    outside of the water passage (figure 2).

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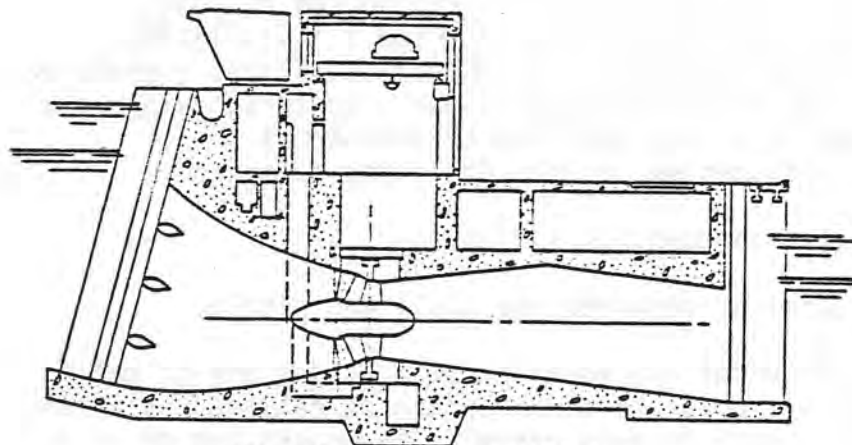
RIM GENERATOR TURBINE
REFERENCE 4 – FIGURE 2

This arrangement creates the following advantages:

   a. Has easy access to the generator.

   b. Eliminates the drive shaft.

   c. Has room for a large generator with a large rotational
   inertia (preferable in small power systems).

   d. Decreases the size of the water passage and civil works
   (compact).

The main problems associated with rim-generators are the connections,
seals, and bearings between the runner and the generator.

Between 1937 and 1951, a total of 73 rim-generator units were
installed in 14 power stations in southern Germany and Austria
(reference 4). Some problems occurred with the connection
between the runner and the outer rim on one of the machines and
the concept was limited to small installations with a runner
diameter of 3000 mm and 10-m head. For these reasons, the
rim-generator was abandoned in favor of alternate designs. There
is, however, a rim-generator on the market today with a new
seal and bearing system. Prototypes of this rim-generator unit
are now being manufactured.

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2. Tube turbines. - In 1930, the tube turbine was patented.
This design was used mainly in Europe for small installations.
In 1960, this idea was reactivated in the United States (reference 2)
at Ozark Lock and Dam on the Arkansas River. In this design, the
generator is driven by a long upstream or downstream shaft
(figure 3). This arrangement has the following advantages:

a. The sealing arrangement is simple.

b. The water passages required are small and simple.

c. The shaft can enter the water passage at any angle, making
it very versatile. Obsolete turbines can be replaced utilizing
the existing structure in many cases. The modernization of a
small hydroplant in northern Wisconsin (reference 5) is a good
example of using a tube turbine to rebuild an old plant.

The tube turbine has the following disadvantages:

a. The long shaft can lead to installation and vibration
problems.

b. For large powerplants, the powerhouse is relatively
expensive to build because of the length of powerhouse required
to accommodate the runner and the generator.

TUBE TURBINE
REFERENCE 4 - FIGURE 3

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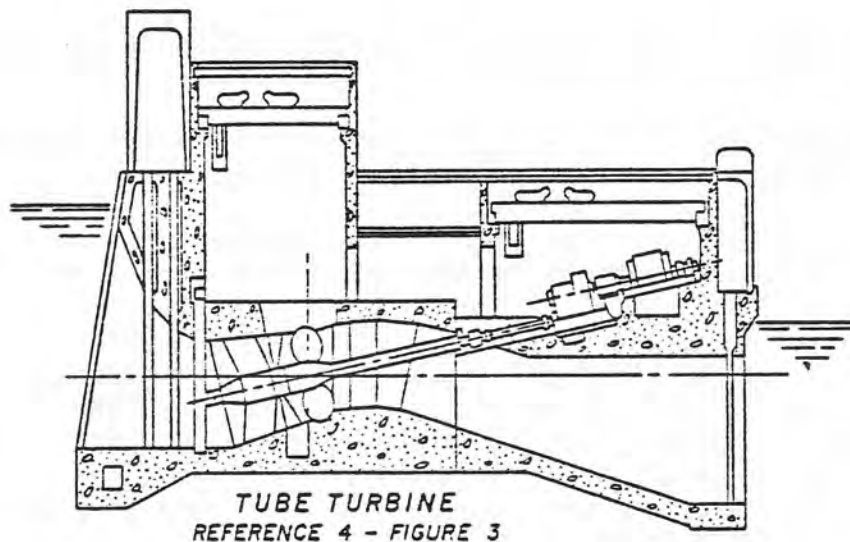
3. Bulb turbine. - In 1933, the bulb turbine was patented. In this arrangement, the generator and turbine are encased in a shell inside the water passage. Figure 4 is a typical bulb turbine installation.

The bulb unit is compact, self-contained, and operationally flexible (reference 2). The main advantage is its good operating record. It lacks the seal problems of the rim-generator and the alinement problem of the tube turbine. There are some disadvantages, such as poor access to the generator, difficult generator cooling, and low-rotational inertia due to the generator being as small as possible to minimize the size of the flow passage required. A minimum runner diameter should be specified for a turbine of a given rated output, since bulb-turbines develop extreme noise and vibration problems when operated at their maximum through-flow capacity. Also, the flow passage entrance must be large to accommodate the bulb unit. Even with these disadvantages, the bulb unit is generally accepted as the best method for developing large-scale, low-head hydropower at this time. The largest bulb units in the world are presently being installed at Rock Island Dam on the Columbia River (figure 5). Reference 2 contains a list of present bulb turbine installations. There are about 125 powerplants in the world using bulb turbines ranging in output from 1.4 MW to 53 MW at Rock Island in the United States. Most of the present installations are in Europe, the U.S.S.R., and Japan, with the Rock Island installation being the first in the United States.
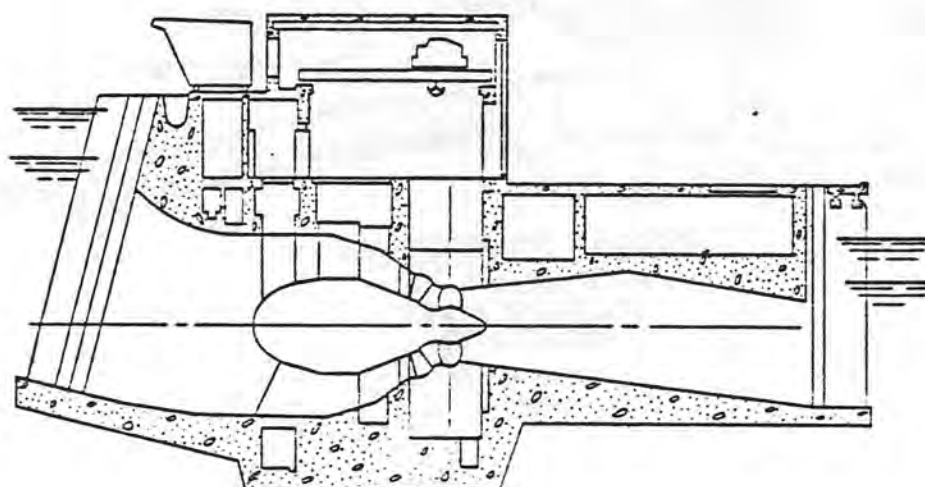
BULB TURBINE
REFERENCE 4 - FIGURE 4
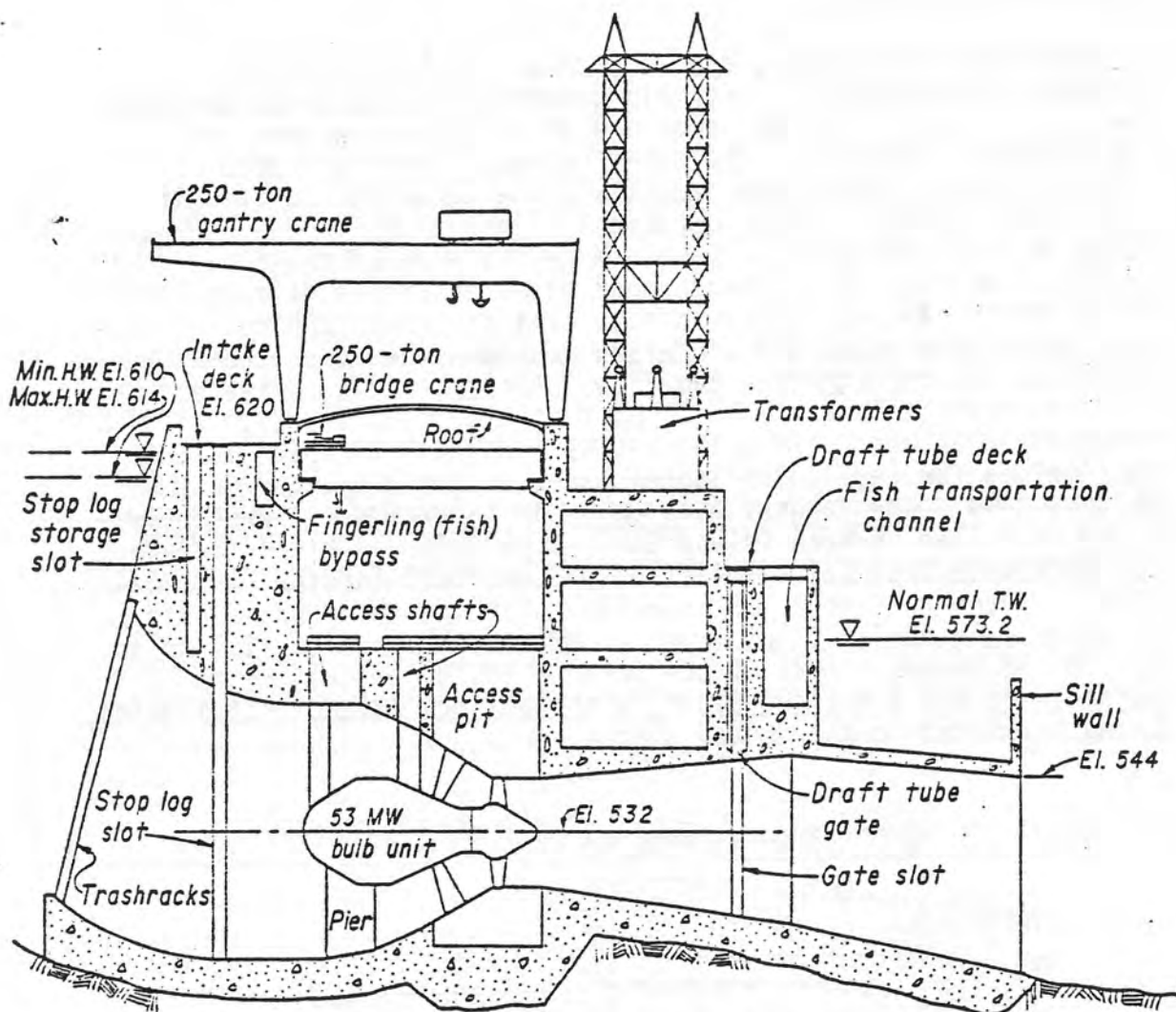
SECOND POWERHOUSE
ROCK ISLAND DAM
COLUMBIA RIVER
WASHINGTON

REFERENCE 39
FIGURE 5

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IV. STATUS OF STANDARDIZATION

A. Standard "Off the Shelf" Units

A few package-type turbine and generator units are available in the
low-head range (less than 20 m). These predesigned units reduce
equipment costs by eliminating the need for site-specific engineering
and by using standardized manufacturing techniques. However, the
standard flow passage shapes may result in a loss in turbine efficiency.
These units have either fixed runner blades or fixed wicket gate
positions or both, resulting in less operational flexibility. The
economics of each potential site should be evaluated to determine if
the savings obtained with "off the shelf" units outweigh the losses.
The following standardized units are presently available:

1. Tube Turbines

Allis-Chalmers tube turbine units. - Ten standardized packaged
designs are available. Single units have capacities to 5000
kW at heads up to 15 m. Flow is controlled with a butterfly
valve in the intake. The wicket gates are fixed and the
runners are either fixed or adjustable. Figure 6 shows the
arrangement of the standard tube units.

Karlstads Mekaniska Werkstad (KMW) miniturbines. - These units
are available for horizontal or vertical installation for flows
from 1 to 15 m³/s and heads from 4 to 25 m. Outputs range
from 100 to 1800 kW. Figure 7 is a typical arrangement. The
turbines have fixed guide vanes and fixed runner blades,
although the position of the latter can be changed when the
turbine is stationary. Flow is controlled with a butterfly
valve.

2. Bulb Turbines

Fuji package-type bulb turbine and generator. - Available in
19 models covering a range of net heads from 5 to 18 m and
outputs from 300 to 4000 kW. The runner is of the fixed blade
type and the flow is controlled with movable wicket gates. A
sectional drawing of the standard bulb turbine and generator
is shown on figure 8.
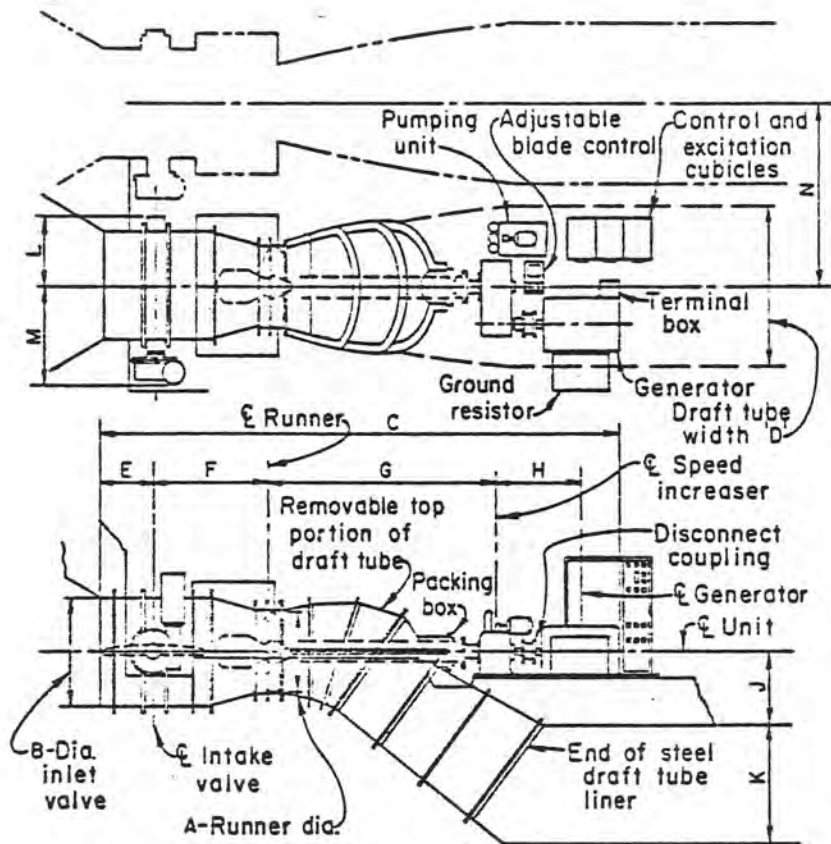
BASIC DIMENSIONS
A= Runner diameter in millimeters (inches) = 1.00
All other dimensions are in proportion from runner diameter

A	750 (29.5)	1000 (39.4)	1250 (49.2)	1500 (59.6)	1750 (68.9)	2000 (78.7)	2250 (88.6)	2500 (98.4)	2750 (108.3)	3000 (118.1)
B	1.43	1.37	1.34	1.32	1.31	1.30	1.29	1.28	1.27	1.22
C	9.28	8.66	8.28	7.80	7.50	7.34	7.19	7.08	7.03	6.97
D	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00
E	0.73	0.70	0.70	0.70	0.66	0.65	0.64	0.64	0.64	0.63
F	1.75	1.60	1.52	1.50	1.46	1.45	1.42	1.40	1.38	1.47
G	3.07	3.04	3.02	2.93	2.88	2.87	2.86	2.85	2.92	2.87
H	1.33	1.32	1.32	1.13	1.13	1.12	1.22	1.13	1.10	1.07
J	0.93	0.93	0.93	0.93	0.93	0.93	0.93	0.93	0.93	0.93
K	1.49	1.49	1.49	1.49	1.49	1.49	1.49	1.49	1.49	1.49
L	0.93	0.90	0.88	0.87	0.80	0.78	0.77	0.77	0.76	0.73
M	1.33	1.30	1.28	1.25	1.26	1.22	1.22	1.22	1.22	1.17
N	3.00	2.75	2.60	2.50	2.43	2.38	2.33	2.30	2.27	2.25

ALLIS-CHALMERS STANDARD TUBE UNITS
FIGURE 6
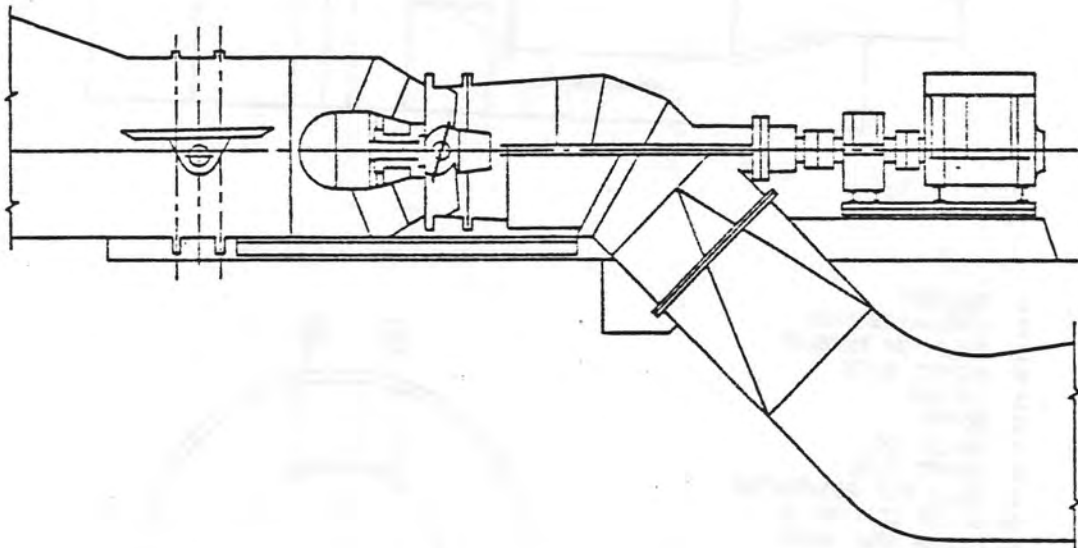
KMW STANDARD MINITURBINES

FIGURE 7
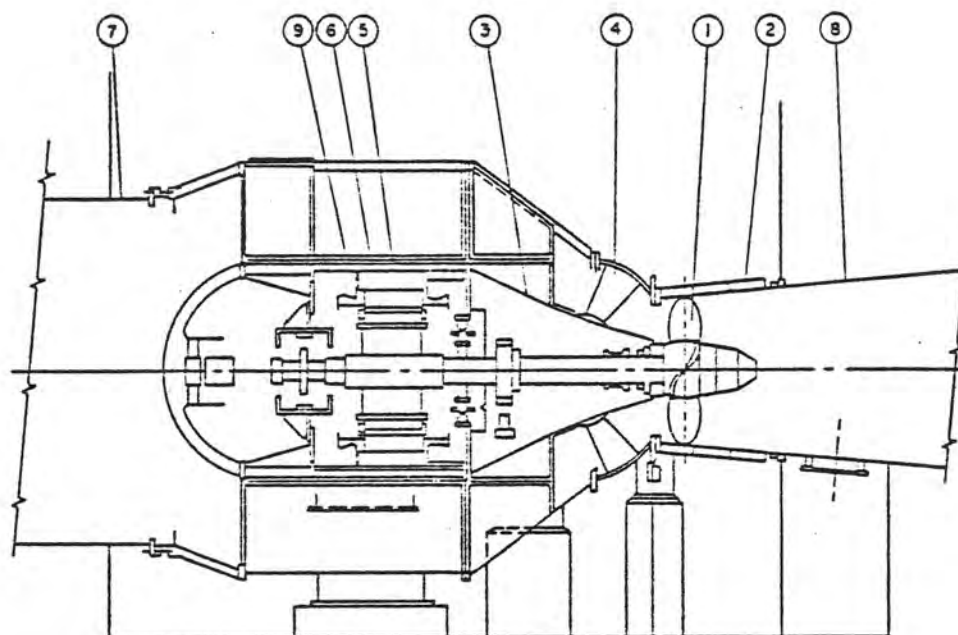
1. RUNNER
2. DISCHARGE RING
3. BULB FOR TURBINE
4. WICKET GATE
5. STATOR
6. ROTOR
7. PENSTOCK
8. DRAFT TUBE
9. BULB FOR GENERATOR
10. UPPER STAY VANE
11. LOWER STAY VANE
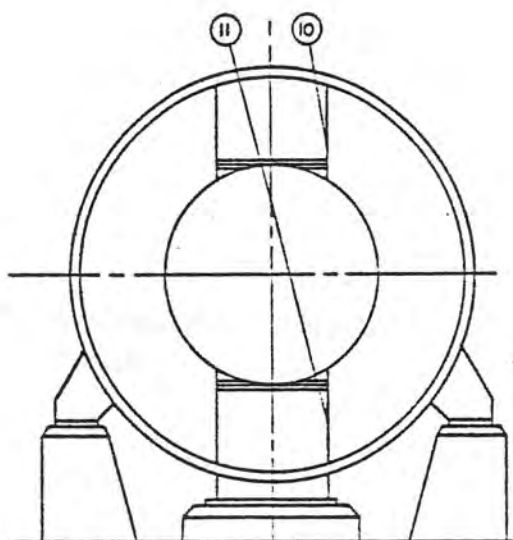
FUJI PACKAGE TYPE BULB
TURBINE AND GENERATOR
FIGURE 8

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3. Cross Flow Turbines

Ossberger "cross flow" turbine assemblies are available in a range of units which have the ability to cope with variations in head and discharge on small dams. The crossflow turbine is a radial impulse-type turbine. The water is forced through a guide vane system and the blades of the cylincrical runners. The water then passes through the runners again to the outlet. Figure 9 is a cross section through a crossflow turbine. The adjustable guide vane allows power generation from 16 to 100 percent of design flow. Although the efficiency for this unit is not as high as for axial flow turbines, it may have widespread application to small sites where limited storage is available and flow and head vary widely.

HORIZONTAL ADMISSION   VERTICAL ADMISSION
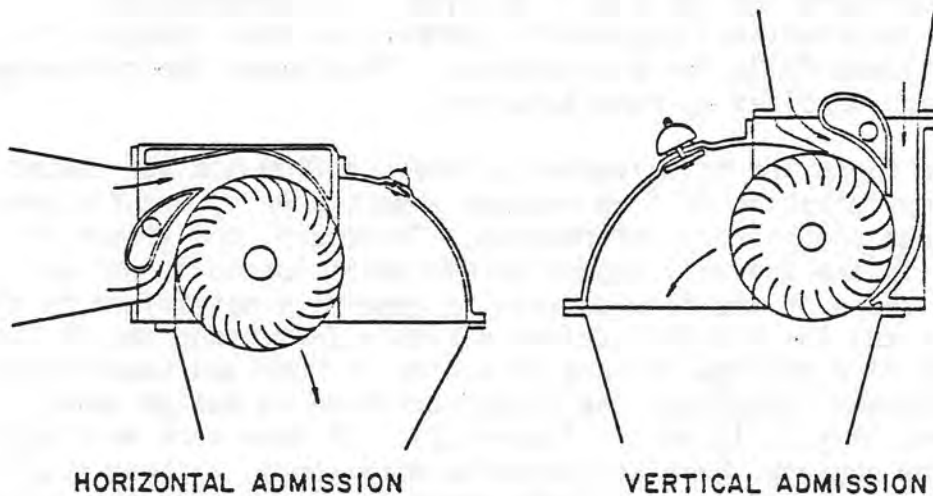
OSSBERGER CROSS-FLOW TURBINE
FIGURE 9

4. Small High-head Turbines

In the high-head area, the number of suppliers for small turbines is limited. Reference 6 contains a partial list of manufacturers involved in supplying small hydromachinery. The turbines used in the high-head range are usually the Pelton impulse type. The design of flow passages for small high-head turbines follows standard hydroelectric engineering practice.

B. Standard Dimensions

A few manufacturers provided drawings with standard flow passage dimensions. These dimensions were determined as a result of model tests and their operational experience over a period of years. However, the data these dimensions were based on were not provided. Figure 10 is a cross section and plan of a bulb turbine according to Escher Wyss' latest design. The dimensions are given in terms of the runner diameter D-1. Figure 11 is Escher Wyss' standard dimensions for the "Straflo" rim-generator unit. Figure 12 is a cross section through a bulb unit showing Fuji's standard flow passage dimensions for a turbine with a 1000-mm-diameter runner. Figure 13 is a consensus configuration for a bulb turbine. This drawing was prepared by International Engineering Company on their project for the city of Idaho Falls for bid purposes. They based their drawings on information supplied by manufacturers.

At this time there are no universally accepted flow passage shapes. Studies on optimization of flow passage shapes done by manufacturers are considered proprietary information. Therefore, the effect of simplifying intake and exit shapes on hydraulic losses is not well known. The shape of the flow passage is generally determined by the manufacturer and the standard design varies widely among manufacturers. The standard flow passage designs according to Fuji and Escher Wyss for bulb turbines illustrate the large variation in design among manufacturers (figure 10 versus figure 12). If data were available relating flow passage shape to hydraulic efficiency, a study could be done to evaluate the cost effectiveness of simplifying the intake shape to reduce the cost of the civil works.

C. Standard Designs for a Line of Turbines

An alternative to standardization of flow passages in general would be to develop a standardized design for a series of powerplants to be installed at similar locations. A feasibility study was performed by Motor-Columbus Consulting Engineers to investigate harnessing of the Rhine River upstream of Lake Constance. The study investigated using 46 standard bulb units in 16 stations. Figure 14 is a general
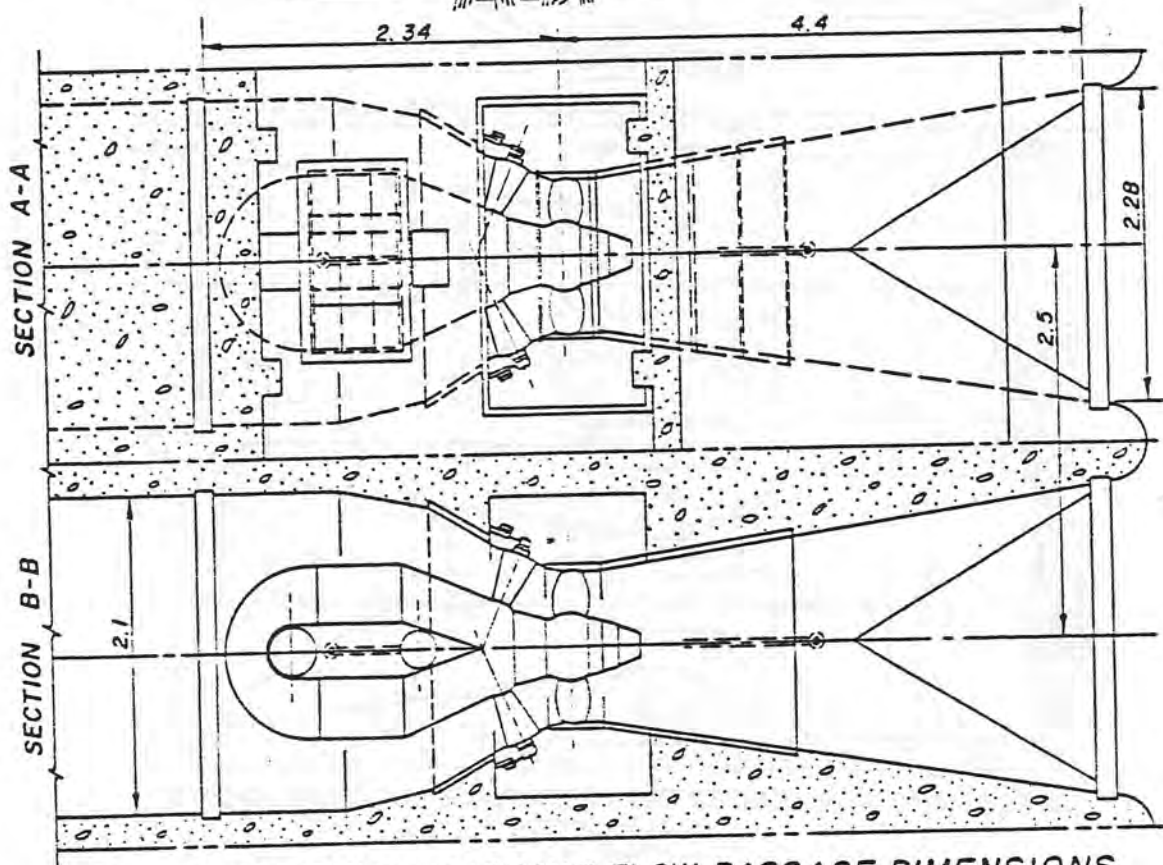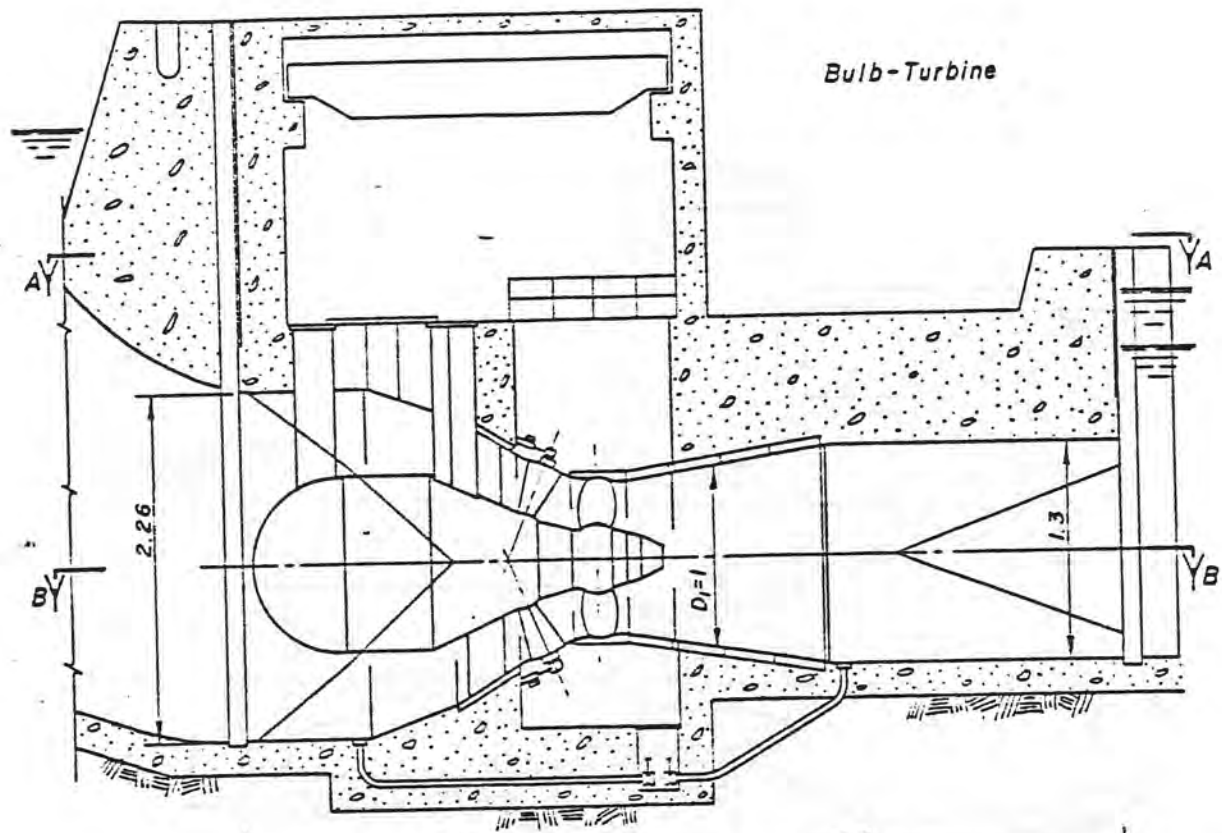
Bulb-Turbine

STANDARD BULB TURBINE FLOW PASSAGE DIMENSIONS
ESCHER WYSS
FIGURE 10
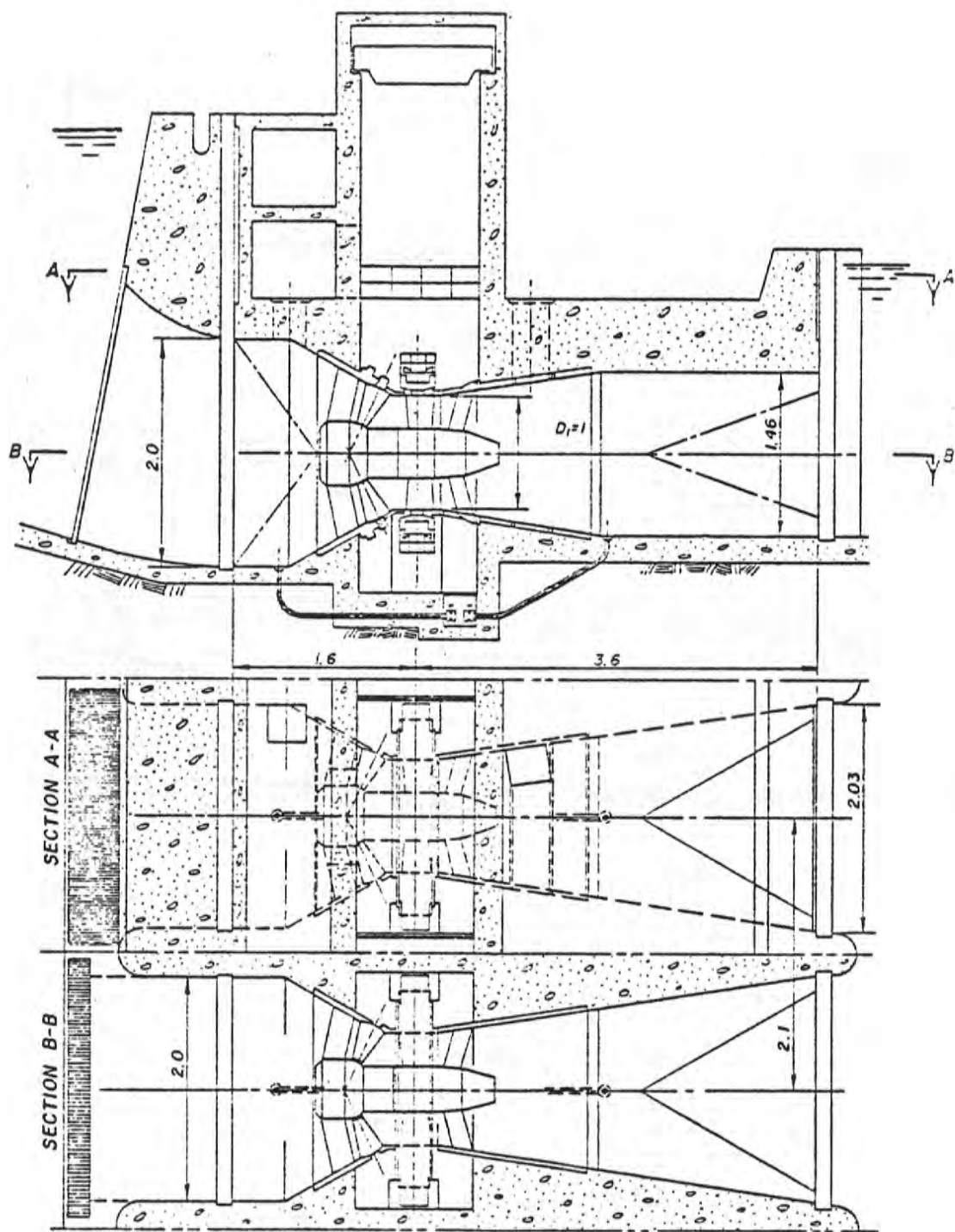
STANDARD RIM GENERATOR FLOW PASSAGE DIMENSIONS
ESCHER WYSS
FIGURE II

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STANDARD BULB-TURBINE FLOW PASSAGE DIMENSIONS
FUJI ELECTRIC CO., LTD. H428510
FIGURE 12

GENERAL PLAN

SECTION A-A

SECTION B-B
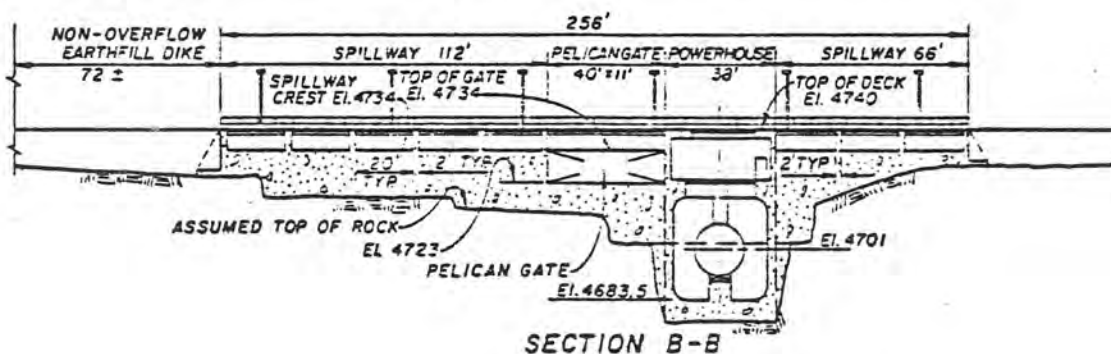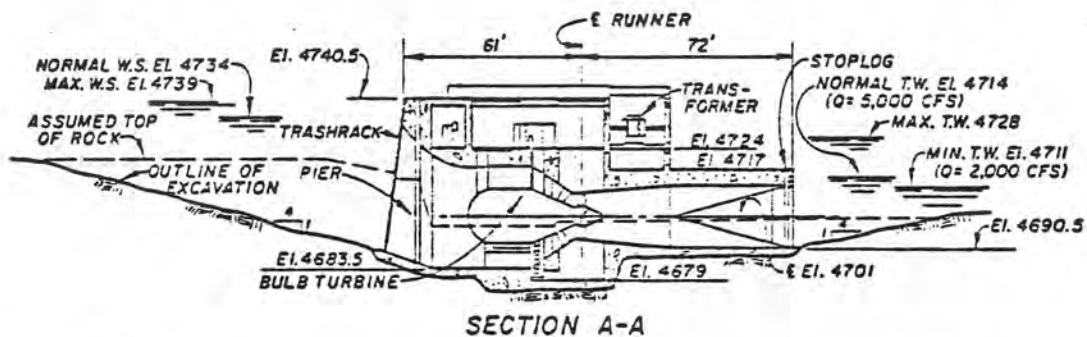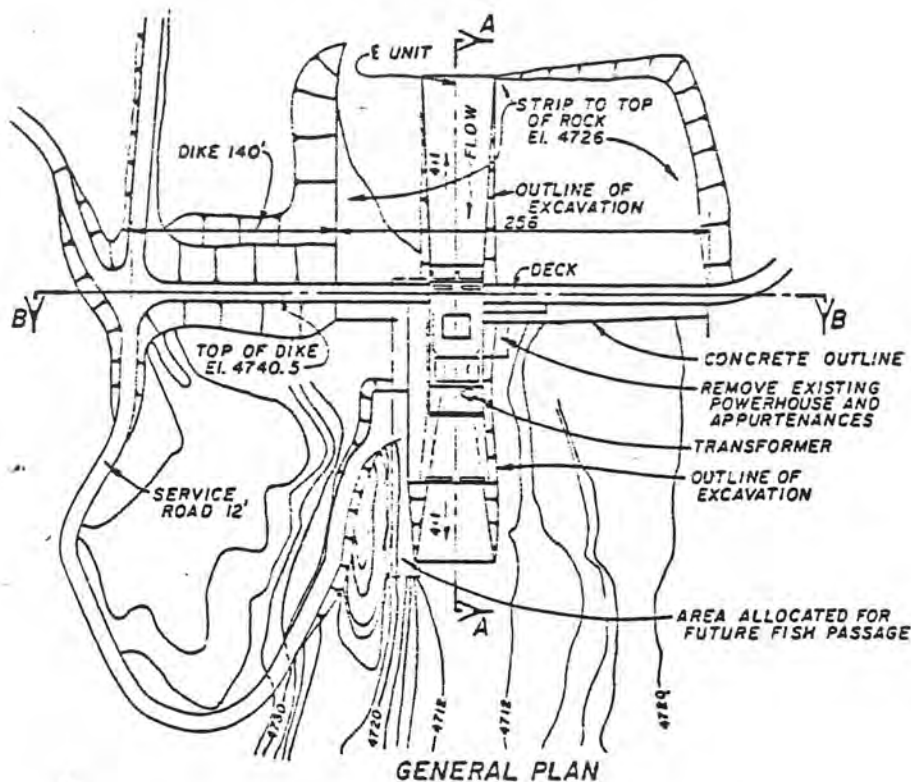
CONSENSUS CONFIGURATION FOR IDAHO FALLS
HYDROELECTRIC PROJECT UPPER POWER PLANT
INTERNATIONAL ENGINEERING COMPANY
FIGURE 13

Trash rake and
stop log crane

Trash channel

20-Ton crane

Reservoir
elevation

Injector
shaft

Downstream
stoplogs

Max tail water
elevation

Stoplogs

Machine hall

Min. tail water
elevation

Trashrack

Generator

Turbine

Draft
tube
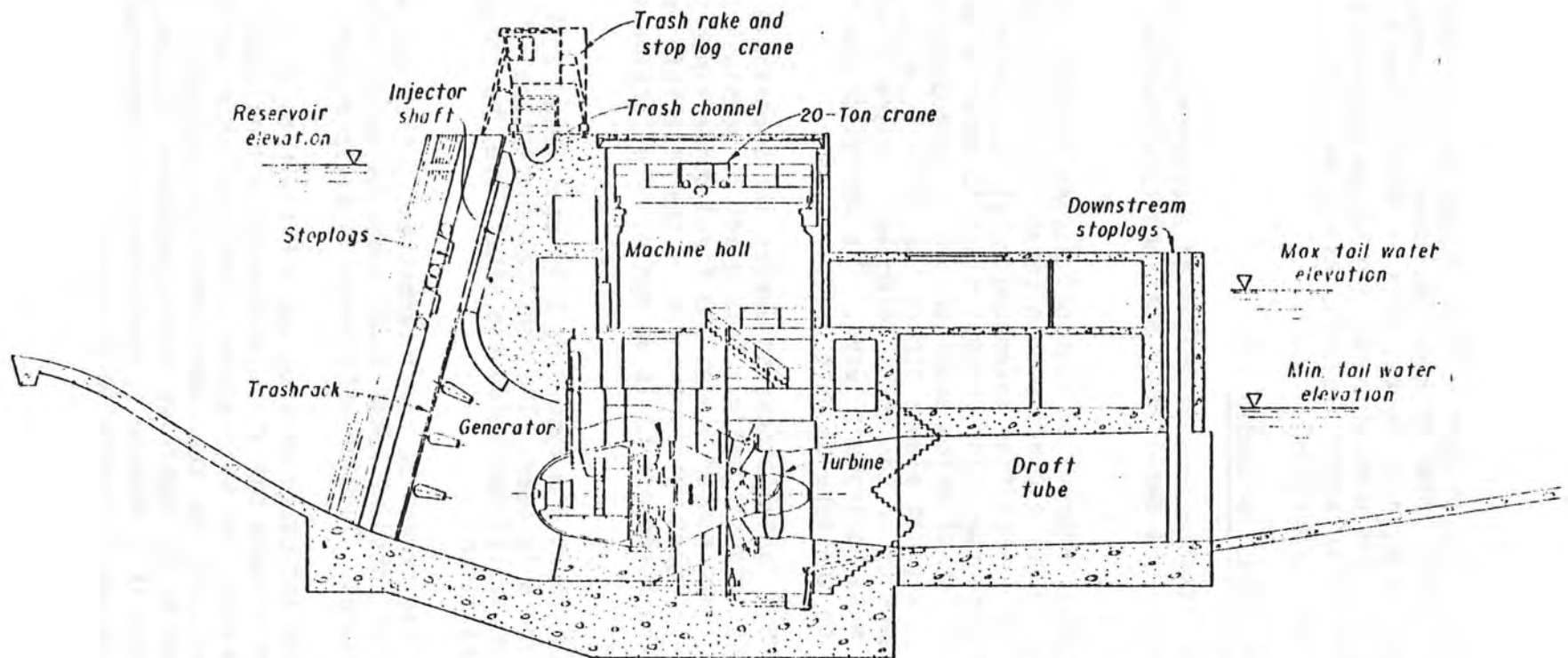
POWERPLANTS ON THE RHINE RIVER FROM DOMAT (EMS) TO FLÄSCH

MOTOR COLUMBUS CONSULTING ENGINEERS

FIGURE 14

layout developed for the project. This concept may be applicable to development of low-head power on American rivers with several possible low-head sites, such as the Ohio or Mississippi Rivers where numerous navigation dams without power generation currently exist. A standard design would reduce the costs associated with site-specific engineering for each powerplant.

D. Mini Power stations in Sweden

The Swedish Power Association has initiated a program to overhaul and replace discontinued small power stations (100-1500 kW) in Sweden (reference 7).

The program includes hundreds of stations having a total output of 1 TWh annually which corresponds with a total installed capacity of about 235 MW. A few simplified automated axial flow units were developed for the program. The present electric power production in Sweden is about 85 TWh of which about 60 TWh is hydropower production. Sweden installed six pilot plants during the 1975-77 period to study operation and economics of the units selected. Figure 15 shows the various types of mini powerplants used. The design of the flow passages depends on the type of unit used.

In most of the small installations in Sweden it is possible to run the power stations intermittently. This makes it possible to design the turbine with fixed runner blades and fixed guide vanes and leave out the regulator. The turbines are run as the reservoir is drained and shut off as the reservoir fills up again. The turbine efficiency is 90-92 percent.

At heads less than about 6 m type B or C is used (figure 15). At heads over 6 m type A is used. The vertical-shaft type is used so the turbine can be positioned below the water level to reduce the potential for cavitation.

If the installations cannot be run intermittently they can be run with regulation for constant water level upstream using units with movable runner blades. Turbine efficiency is 85-91 percent.

The pilot stations indicated that it was difficult to bring the costs down enough to make the units economically profitable. However, the Swedish Government has made economic aid available (up to 35 percent of the cost) for the least profitable projects. Economic aid was considered to be justified since hydropower reduces dependence on imported oil and is a renewable, environmentally acceptable form of energy that would otherwise be wasted.

TYPE A

TYPE C

FLOW →

TYPE B

FLOW →

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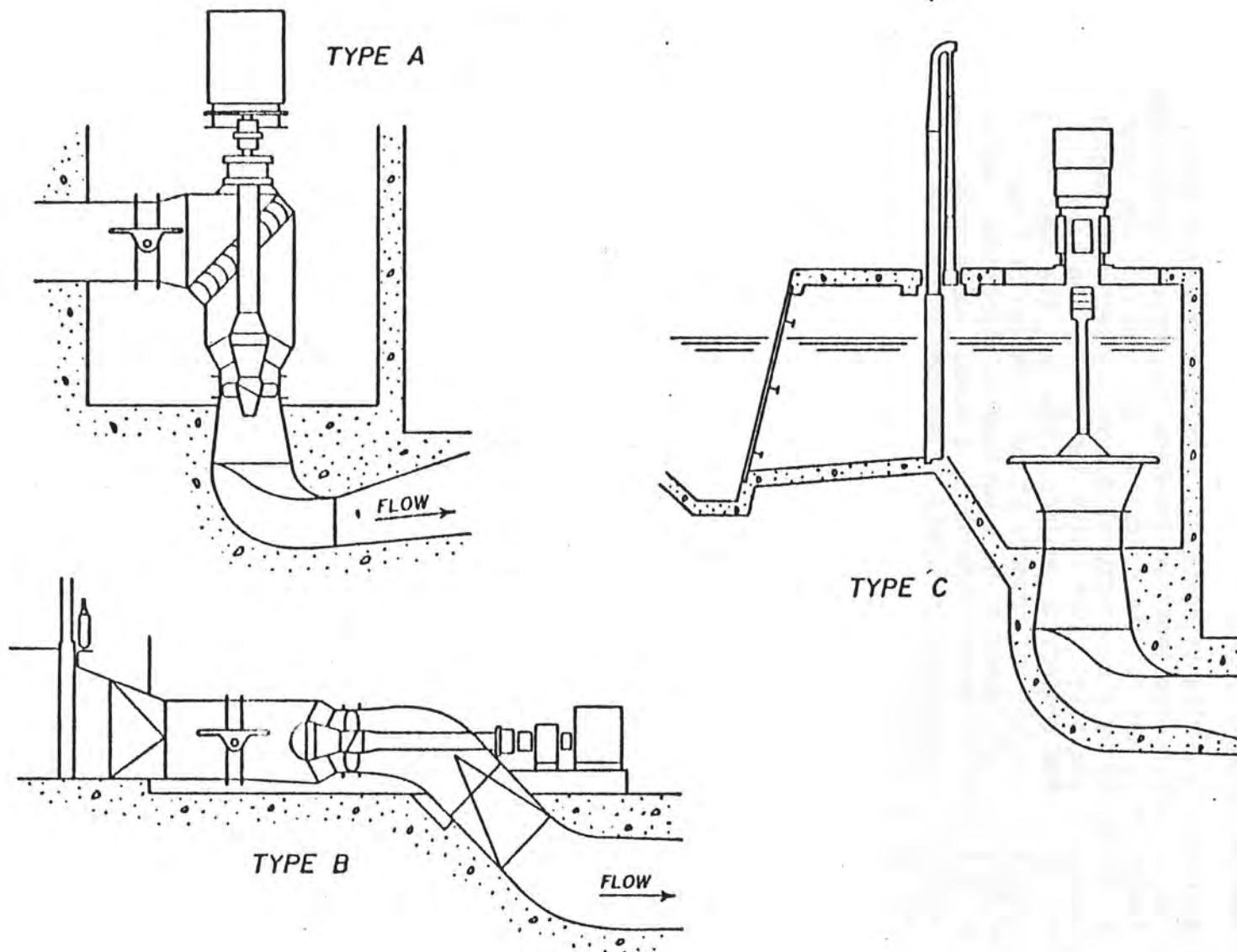
VARIOUS TYPES OF MINI-POWER STATIONS
SWEDISH POWER ASSOCIATION (REFERENCE 7)
FIGURE 15

E. Comments on Standardization

Most designers and consultants contacted felt that standardization of the inlet and outlet sections for low-head hydropower installations will not always bring the optimum solution. The shape of the water passage is influenced by the structural support system requirements and the geology of the site. Even units designed by the same manufacturer vary from project to project. However, the need for a manual to serve as a guide and promote standardization of flow passage designs was considered to be a worthwhile objective. While not all designs would be exactly the same, a manual would bring together information contained in scattered sources and aid in the planning and design process.

V. CURRENT DESIGN PRACTICES

This section summarizes material in technical literature, and comments made by manufacturers and designers concerning flow passage design. Main points covered in the references are discussed; however, the references should be referred to if more detail is required. The hydraulic design of the draft tube is customarily done by the manufacturer, since it is considered an integral part of the turbine in determining turbine performance. The hydraulic design of the other flow passages is normally considered a part of the overall plant design. However, the manufacturers of standard designs usually control the design of the other flow passages also.

A. Forebay

The design of the entrance channel in the forebay should provide a uniform flow distribution to avoid the tendency for vortices and to minimize trashrack losses. The entrance channel is designed according to common engineering practice. If the geometry of the approach channel is in question, a model study can be done to assure acceptable flow conditions. Flow velocities and surges are of particular concern on navigable rivers. Run-of-river low-head dams are typically designed with the powerhouse on one side of the river and an overflow weir-type dam on the other side. In some cases an entrance channel to the powerhouse is excavated in the bank. If a navigation lock is included in the dam it is usually on the opposite bank from the powerhouse. Another possible configuration in a narrow dam is to have a gated weir over axial flow turbines.

B. Intake

1. - Vortex formation. - Intakes should be designed with a sufficient submergence to avoid vorticies. Air-entraining vortices may decrease turbine efficiency and pull floating debris into the turbine. Reference 8 describes the development of design criteria to avoid vortices at low-head intakes, based on a study of 29 existing hydroelectric intakes.

The intakes studied have the general configuration shown in figure 16. The factors which appear to affect the formation of a vortex are: geometry of the approach flow, velocity at the intake, the size of the intake, and submergence. This article concentrates on the effects of velocity, intake size, and submergence on vortex formation. The following relationship was developed by trial and error:

$$S = CV_d^{1/2}$$
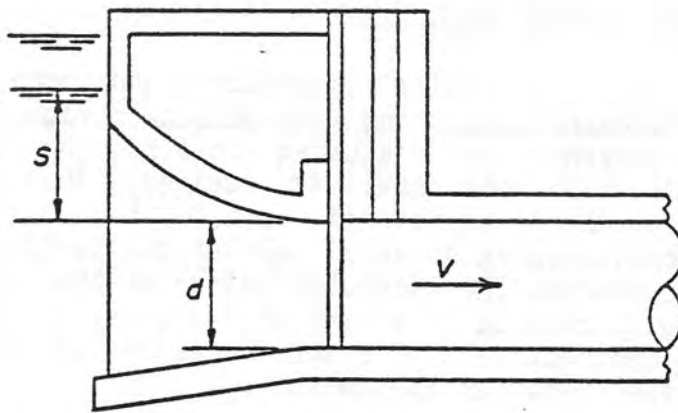
GENERAL CONFIGURATION OF AN INTAKE
REFERENCE 8
FIGURE 16
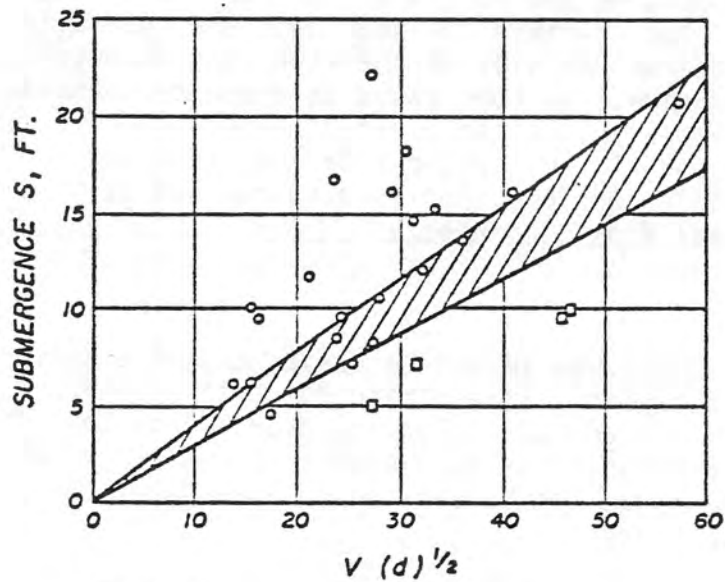
LEGEND:

□   Intakes with vortex problems

○   Intakes with no vortices

▨▨▨▨▨   Recommended min. submergence

RECOMMENDED MINIMUM SUBMERGENCE
REFERENCE 8
FIGURE 17

The effects of lateral approach flow were not evaluated in this study. However, Montreal Engineering Company of Canada uses a coefficient C of 0.3 for intakes with symmetrical flow and 0.4 for lateral approach flow to determine minimum allowable submergence. These coefficients correspond to the lower and upper limits, respectively, of the shaded area on figure 17.

If the available submergence is not adequate, a model study can be done to study vortex formation. However, accurate prediction of vortex formation using a scale model is difficult. The effect of surface tension and air entrainment are important and are difficult to study in a Froude scale model. Reference 9 describes a general method for investigating potential scale effects on free surface vortices, so that a projection to prototype operating conditions can be made. According to reference 9 "projection of model vortex severity to the prototype implies that the Reynolds number must be increased without changing the Froude number, and that the trend of a parameter related to vortex severity be documented." The curves of vortex severity in figure 18 were generated by observing vortex severity in a model by varying water temperature (T), thus varying the viscosity and Reynolds number. Froude ratios were also varied by increasing or decreasing model discharge. The Reynolds number can be increased in the model by changing the water viscosity or changing the model scale ratios. The easiest method is to operate the model at different water temperatures to change the viscosity. The prototype vortex severity is predicted by noting the vortex severity at the prototype Reynolds number for a Froude number ratio of 1. The vortex severity trends must be extrapolated to the prototype Reynolds number in many cases.
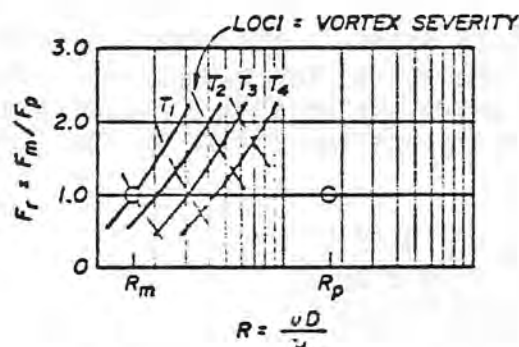
VORTEX SEVERITY VERSUS FROUDE AND
REYNOLDS NUMBERS
REFERENCE 9
FIGURE 18

27

Reference 10 investigated the use of rafts placed under the water surface and other devices to prevent formation of air entraining vortices at Grand Coulee Third Powerplant. The hydraulic model study included vortex tests on the effect of trashrack structures, upstream channel geometry, intake modifications, and submerged and floating rafts. The report concluded that the prediction of prototype vortices using a model is suspect. The value of the results of this report and the vortex suppression techniques described would be enhanced by prototype observations.

Reference 11 reports on the use of the flow "injector wall." This device has the same effect as a raft. Because of an increase of surface current, the vortices are suppressed. The injector was developed for bulb turbines on run-of-river plants to increase flow velocity on the surface to promote movement of floating debris toward the trashrack on the front of the powerhouse. A flow injector is shown on figure 14. The bibliography in reference 9 contains additional references which may be helpful in determining minimum submergence.

2. Pier design. - In typical run-of-river plants a flow separating pier divides the powerhouse and spillway. Piers also divide the intakes where there are multiple units. The size and shape of the pier should be carefully designed to avoid stagnant areas or flow disturbances which may cause excessive head losses or eddies. Hydraulic Design Criteria (HDC) by the U.S. Army Corps of Engineers (reference 12) contains design information useful in designing piers and intakes.

Reference 51 has information on German practice in designing the size and shape of flow separating piers between a powerhouse and weir. The pier design procedure takes into account the location of the powerplant with respect to bends in the river, alinement with the weir, and the entrance channel to the powerplant cutting into the riverbank. Figure 19a shows the general shape of the flow separating pier. The following are the limits on the length and width of the pier:

$$0.85\ B \leq L \leq 0.95\ B$$
$$0.28\ B \leq B_1 \leq 0.35\ B$$
$$0.72\ B \geq B_2 \geq 0.65\ B$$

The following formula defines the width of the pier (B).

$$B = CQ^{2/5}$$

The coefficient C varies between 0.7 and 1.4 depending on the factors mentioned above. Reference 51 should be referred to for details of this design method.

FLOW SEPARATING PIER
BETWEEN POWERHOUSE AND WEIR
REFERENCE 51
FIGURE 19a

H_d = DESIGN HEAD
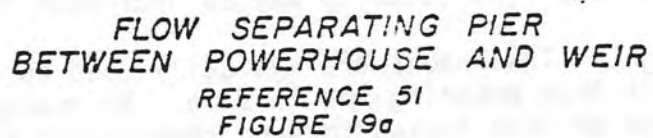
PIER NOSE SHAPES
TYPE 3A DIMENSIONS IN PARENTHESES
REFERENCE 12
FIGURE 19b

29

HDC sheets 111-5, 111-6, and 122-2 describe pier effects for gated overflow spillways. Abutment effects are described in sheets 111-3/1 and 111-3/2. When spillways are operated with adjacent bays closed the piers adjacent to the closed bays produce abutment-type effects. Although these pier design criteria are for spillways and not for submerged intakes they do show the relationship between pier shape and flow contractions. Figure 19 shows the five different pier nose shapes investigated. Pier types 1 and 4 (see figure 19b) are the least desirable from the standpoint of negative pressures. Types 2, 3, and 3A were recommended for general use with high heads. Type 3A had the most desirable flow contraction characteristics.

The design of the flow separating pier could benefit from further research. If the contraction could be reduced without using a wide separating pier the flow capacity may be increased considerably.

3. Trashrack design. - Trashracks are typically used to prevent large debris from entering the turbine. Reference 13 contains information on head losses through rectangular bar trashracks. This article evaluates previous work on trashrack and baffle losses and introduces additional data to supplement design information. Equations are developed to calculate the head loss as a function of the approach velocity and clear spacing between the rectangular bars. The head loss through the trashrack was found to be a minimum for bars with depth over thickness (d/t) ratios of about 3.0 for any solidity (S) between 0.25 and 0.75. Where solidity is the ratio of the flow area blocked to the total flow area. If the trashrack blocks one-half of the flow area the solidity would be 0.50.

The following equations were found to accurately predict total trashrack and baffle head loss.

$$h' \ (d/t < 3.0) = \frac{v^2}{2g} \left(\frac{1}{(mK)} - 1 \right)^2$$

$$h'' \ (d/t > 3.0) = \frac{v^2}{2g} \ \frac{1}{K^2} \left(\frac{(1-m)}{m^2}^2 + (1-k)^2 \right)$$

in which m = the coefficient of contraction, and K = b/(t + b), where b = the clear spacing between bars, t = the thickness of the bars, and $\frac{v^2}{2g}$ = the approach velocity head.

Harza Engineering Company suggests velocity through the gross area of the trashracks should be limited to 1.5 m/s at rated turbine discharge. A large spacing of rack bars (150 mm) can be used for axial flow turbines. The relatively high approach velocity does not produce excessive trashrack head losses if the approach flow is not oblique.

30

4. Intake shape. - Intake shapes vary with the type of turbine used. Intakes are either circular or have a rectangular to circular transition. The face of the intake is either vertical or inclined. However, vertical intake faces simplify design and construction including slots for intake gates, bulkheads, and trashracks.

The hydraulic design of the intake is based largely on engineering judgment; the classical bellmouth criteria are used as a starting point. The entrance end of the bellmouth curve is then truncated to obtain a short intake profile which achieves required contraction in a length equal to approximately 1 runner diameter. General information on the effect of intake shape on hydraulic losses is not available. References 1 and 46 indicate, however, that the size of the intake could possibly be reduced without introducing excessive additional hydraulic losses. The ideal bellmouth entrance design was developed for high-velocity conduits. The velocity for low-head turbines is much lower. It is possible that savings could be achieved in trashracks, bulkheads, entrance gates, and other areas by reducing the entrance size.

It might also be possible to simplify the framework by using flat surfaces to approximate curved surfaces. The cost effectiveness and practicality of these changes require further study to determine if reducing entrance sizes and simplifying intake shapes are worth pursuing.

5. Fish passage allowances. - If passage of fish is a factor, the maximum allowable intake velocity or other intake design parameters may be controlled by the design of fish passage facilities.

C. Runner Diameter and Turbine Setting

The turbine setting is determined by the allowable specific speed. The specific speed is usually controlled by the cavitation potential. From an economic standpoint, it is desirable to select the highest specific speed possible since this results in a smaller, lighter generator. The cavitation potential of an axial flow turbine is generally lower than a Kaplan unit. This allows the axial turbine to have a higher setting or a higher speed (reference 2). Other factors such as smooth operating range and submergence also enter into the design. Reference 14 used data from 130 power stations to develop a design aid to determine runner diameter and the powerhouse size. A design chart was developed to determine the approximate diameter of a tubular (axial flow) unit. The upper diagram in figure 20 makes it possible to determine the capacity in kilowatts. In the lower diagram the runner diameter can be determined. The head H is marked on the lateral scales and a line is drawn connecting these
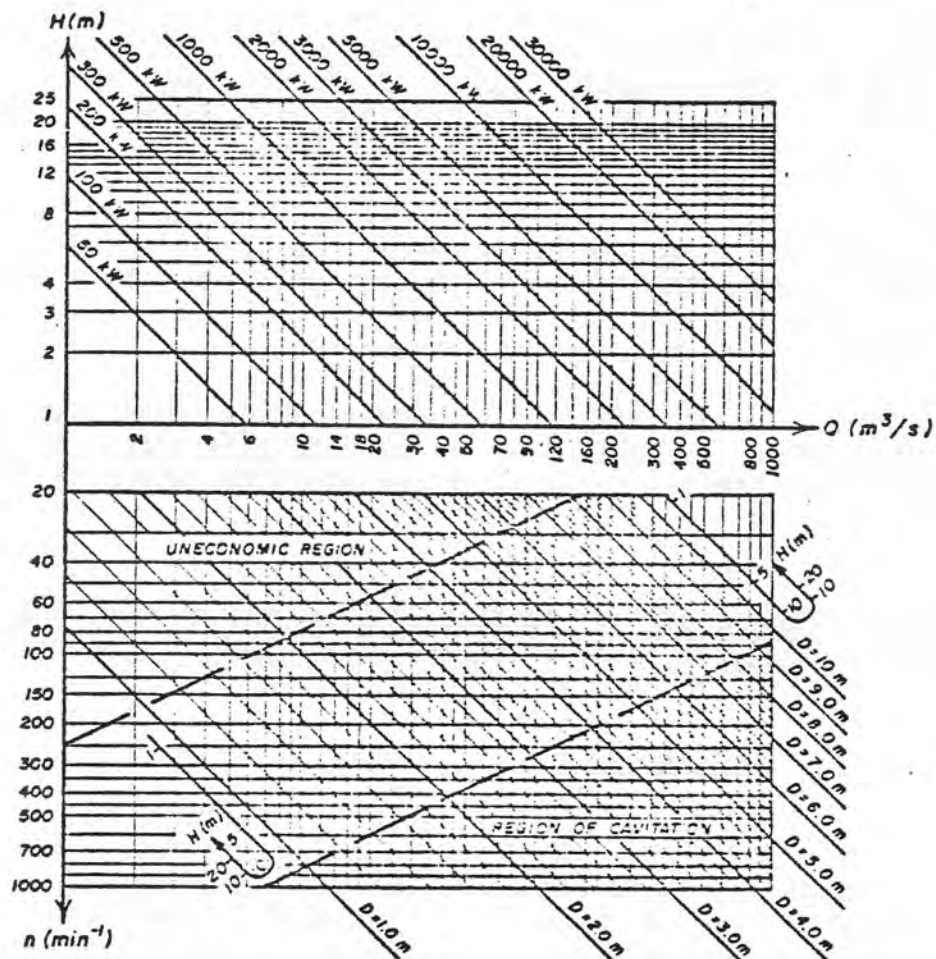
RUNNER DIAMETER DESIGN CHART
REFERENCE 14
FIGURE 20

32

points. The runner diameter can then be determined for a specific discharge (Q) or specific speed (n). The range of applicability is bounded by a region of uneconomic operation and the region of cavitation. The assumptions are checked by comparing existing projects with predictions using this method.

Reference 15 also gives charts for finding the approximate runner diameter and rules of thumb enabling the approximate dimensions for an entire unit to be found. Figure 21 is an empirical plot of the relation between diameter (D) and the ratio of power to head (Hp/H). The formula is

$$D = 0.24 \left(\frac{Hp}{H} \right)^{0.4}$$

in metric units. For power in terms of kW of turbine output the formula is

$$D = 0.272 \left(\frac{KW}{H} \right)^{0.4}$$

This reference also contains an extensive bibliography on axial flow turbines.

Reference 16 presents a method for developing a nomograph for determining the setting of a hydraulic turbine relative to the tailwater level. The turbine manufacturer must supply data concerning cavitation performance to develop the nomograph. This method is useful only for the installation under consideration.

D. Draft Tube

The hydraulic design of the draft tube is customarily determined by the turbine manufacturer, since it is considered an integral part of the turbine in determining the turbine performance. However, there is potential for decreasing the length of draft tubes, thus decreasing the cost of the civil works. According to reference 17, the most efficient conical draft tube would be about 10 runner diameters long with a total included angle divergence of about 7°. Economics usually limits the length of the draft tube to 4.5 to 5 runner diameters with the total angle of diversion of about 13 to 15°. A study done on the feasibility of low-head hydroelectric generation in 1969 (reference 18) concluded that, "The section of the civil works structure where improvement would be most significant is the draft tube." The draft tube accounts for about 30 percent of the cost of the civil works in a low-head structure. There are several possible methods presented in reference 18 for preventing boundary layer separation, thus shortening the draft tube (figure 22).
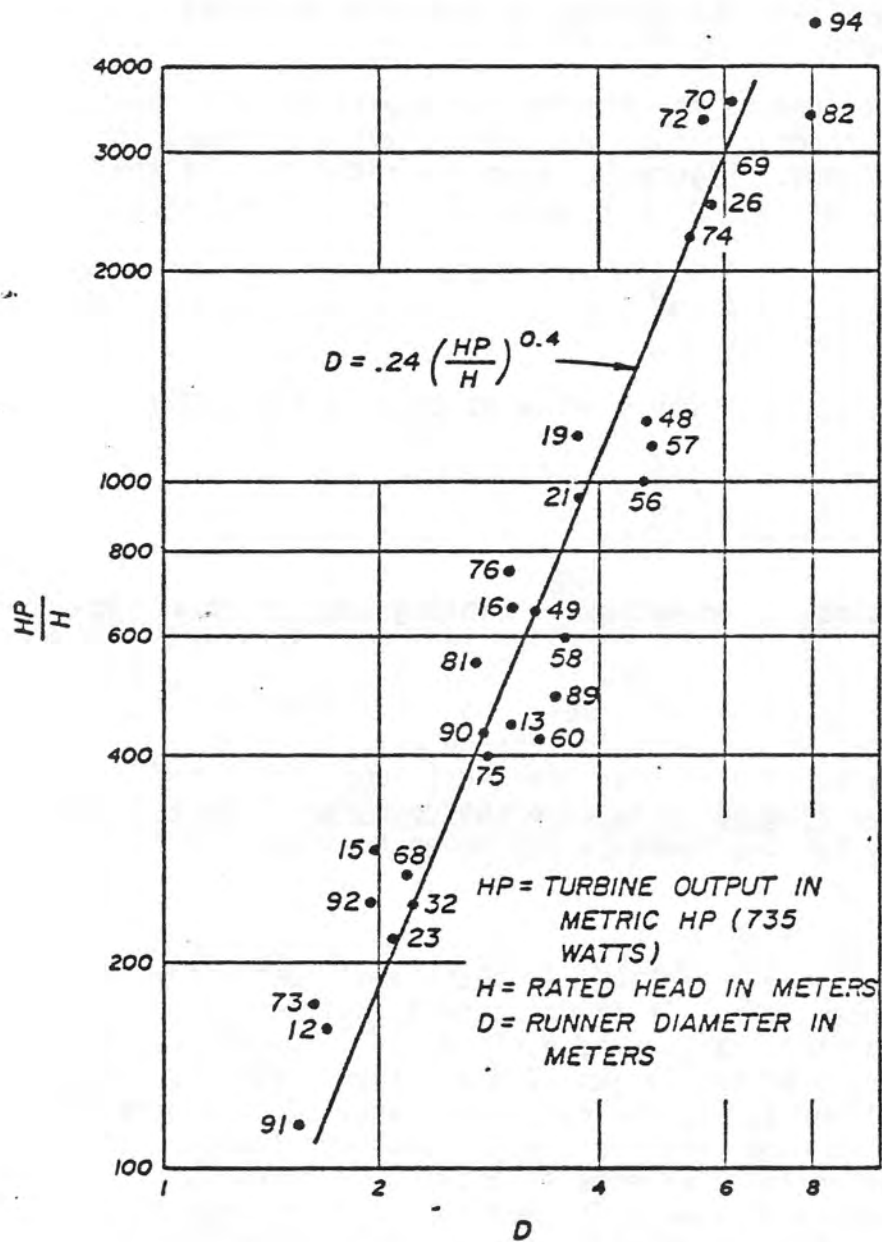
TUBULAR TURBINES (DIAMETER vs $\frac{HP}{H}$)

REFERENCE 15

FIGURE 21

34

FLOW

FLOW

SHORT DIFFUSER WITH CONTROL VANES

SHORT DIFFUSER WITH BOUNDARY LAYER SUCTION

FLOW

MULTIPLE CONE DIFFUSER
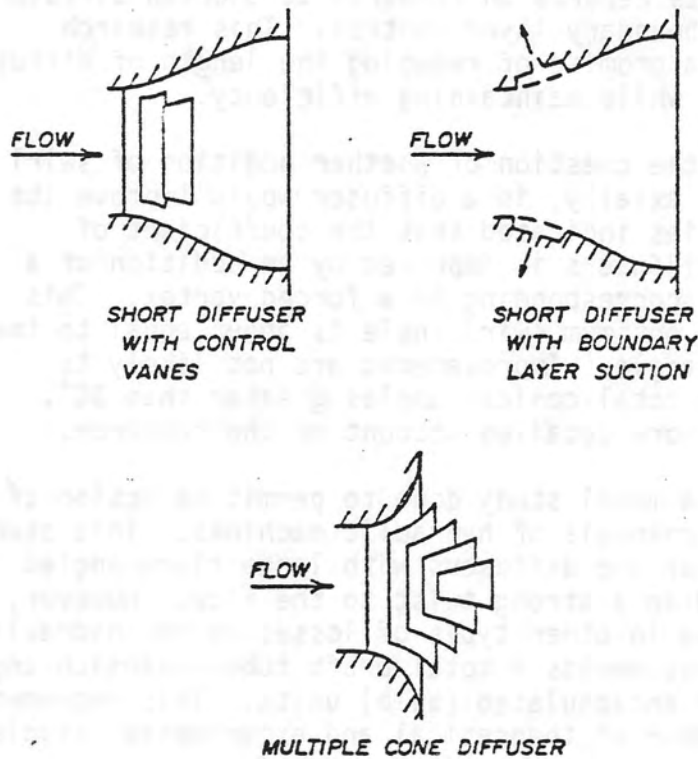
POSSIBLE METHODS FOR ·SHORTENING DRAFT TUBES
REFERENCE 18
FIGURE 22

The study concluded that a need existed for research in the design of draft tubes to apply modern principles of boundary layer control to reduce the length of draft tube diffusers. Research in this area has not been done on hydraulic turbines. The concept of boundary layer control has been bypassed in favor of simple conical diffuser designs because of the potential risks of complications involved with a new concept. Shortening of the draft tube in this manner would probably require a successful demonstration project before it would be accepted. Reference 19 reports on research to shorten diffusers for wind turbines using boundary layer control. This research indicates that there is a promise of reducing the length of diffusers by an order of magnitude while maintaining efficiency.

Reference 20 deals with the question of whether addition of swirl to fluid, otherwise flowing axially, in a diffuser would improve its performance. Model studies indicated that the coefficient of performance of conical diffusers is improved by an addition of a swirl velocity component corresponding to a forced vortex. This paper concluded that the optimum swirl angle is about equal to the diffusers total conical angle. Improvements are not likely to extend to diffusers with total conical angles greater than 30°. Reference 21 provides a more detailed account of the research.

Reference 22 reports on a model study done to permit selection of dimensions for diffuser channels of hydraulic machines. This study states that performance of the diffusers with large flare angles would be improved by adding a strong twist to the flow. However, this leads to an increase in other types of losses in the hydraulic machine. This article recommends a total draft tube expansion angle of 15-17° for horizontal encapsulated (bulb) units. This recommendation resulted from a number of theoretical and experimental studies in the U.S.S.R.

Reference 23 reports on a study that investigated the interaction and losses as flow moves through the various elements of the flow passage in a bulb turbine. Flow parameters were measured in various turbine operating modes, characterized by blade angles, $\emptyset = 0, 5,$ and 20° and rotating speeds N = 110, 140, and 180 r/min with various settings of the wicket gates. Based on these data, energy losses in individual sections of the flow passage are presented in a chart for several operating modes. Figure 23 gives energy balances for two turbine operating modes. It should be noted that the selection of design data should be based on minimum total losses in the turbine as opposed to minimum losses in each section of the waterway. The results from this study can be used to estimate energy losses in the elements of flow passages in actual turbines

(a)   (b)

ENERGY % (vertical axis, values 100, 90, 80, 70)

Q'_I, ℓ/S
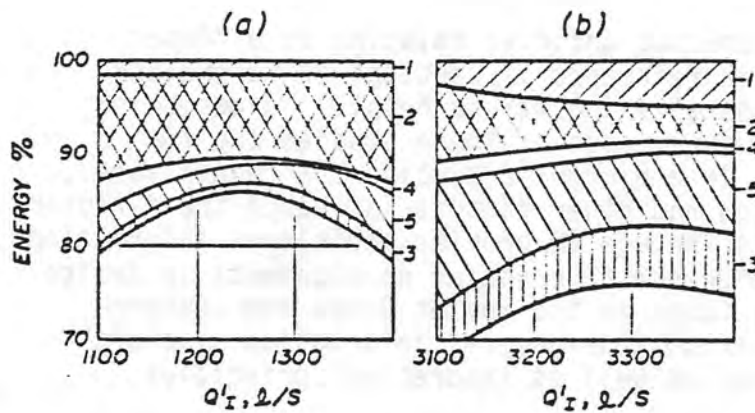
ENERGY BALANCES OF HYDRAULIC TURBINE AT (a) $\phi = 0$,
$n'_I = 140$ rpm (b) $\phi = 20°$, $n'_I = 180$ rpm.
ENERGY LOSSES — 1-APPROACH CHAMBER, INCLUDING
WICKET GATES. 2-RUNNER. 3-DRAFT TUBE.
4-EXIT VELOCITY. 5-ROTATION LOSSES

REFERENCE 23
FIGURE 23
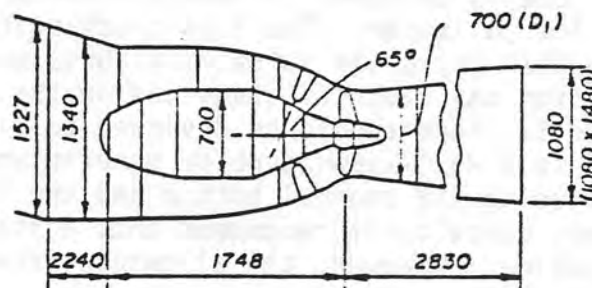
* ALL DIMENSIONS IN MILLIMETERS

WATERWAY OF MODEL BULB TURBINE
REFERENCE 23
FIGURE 24

37

if the flow passage shapes are similar to those used in this study.
Figure 24 shows the dimensions of the model.

The literature contains numerous articles relating to diffuser
design. The biliography in reference 17 contains 39 references
(mostly for airflow diffusers) which may be helpful in designing
draft tube diffusers. However, most of these studies are for
uniform flow conditions. In a hydraulic turbine the runner speed,
amount of swirl in the flow and other factors influence the diffuser
efficiency. References 24 through 28 provide additional information
on draft tube design. Reference 52 contains developments in design
and construction of draft tubes in the Soviet Union and Western
Countries. This reference contains general information on draft
tubes of hydraulic turbines as well as theoretical principles
of draft tube design.

The designer is not always free to select the optimum flow passage
dimensions. Other considerations such as structural requirements or
fish passage facilities may affect the shape. For example, allowances
for fish passage facilities may make it necessary to change the flow
passage geometry.

Gates should not be located immediately downstream of the runner in
the draft tube. The design of this section is critical to the
efficiency of the draft tube. The draft tube gates at Rock Island
Dam are located about 2 runner diameters downstream.

F. Tailrace

The outlet from the draft tube is designed to have a minimum submer-
gence of 0.75 to 1.0 m at low tailwater. The flow discharging from
the draft tube at low tailwater may cause surges and turbulence and
the rise of the channel bottom may cause fluctuations in the tailwater
level. The channel and the tailrace should be designed to minimize
these problems. According to a well-known turbine manufacturer, the
relationship between the rise in the channel bottom and the tailwater
is not known. At this time, consultants recommend that a steep rise
in the channel bottom be avoided; however, the allowable rate of
rise is not defined.

VI. GENERAL COMMENTS

A. Tidal Power

Tidal power is one area in low-head hydropower which is receiving a great deal of attention. References 29 through 33 are general articles on tidal power projects using bulb turbines. Reference 29 gives complete details of development of the bulb turbines for the tidal powerplant at La Rance. The La Rance Project pioneered tidal power development. References 30 through 33 give general information on tidal power.

Flow passage shapes for tidal power developments usually allow for power generation in both directions. The intake as well as the outlet sections act as draft tubes. Consequently, the flow passage shapes are different than in other low-head installations. The flow passages are designed for the individual installation.

B. General References

There are many general articles on the advantages and potential for development in the low-head hydropower area. References 33 through 50 are general articles which provide background information in low-head hydropower. These articles present many case studies from existing installations and discuss the need for low-head power generation; however, they do not contain very much actual design information on flow passages for low-head plants.

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