Turbine Selection
Upper Kotmale Hydropower Plant

Marlon Fernando

Abstract: This paper outlines general guide lines of selection of hydro turbines, and selection of turbines for the Upper Kotmale Hydro power Project, in particular. In the latter case, *'Final Design Report' Volume 1, Main Report, June 1995'* has established *principle features* necessary for the above task. The J-Power - Japanese Consultancy firm- in collaboration with the CECB, worked as the consultants for the project, and for preparation of specifications.

Keywords: Hydro turbines, Turbine types, Francis turbine, Specific speed, Model test,

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1. Introduction

This paper is based on materials from *Final Design Report' Volume 1, Main Report, June 1995', Bid document submitted by the Consultants, Mitsubishi Heavy Industries -MHI- (Contractor)* submissions and various other publications.

The Francis turbine has been selected for the Upper Kotmale Hydro power Project (UKHP) and thus, this paper is restricted for its selection.

Accordingly, as point of departure, principle features, in selection of turbines, can be cited as follows:

Maximum Gross Head = 491 m
Net Head at full operation = 473 m
Maximum Plant Discharge = 36.9 m3/s (for 02 units)
Annual Energy = 531.9 GWh **(409 GWh)***
Number of Units = 02
Plant Factor = 40% **(30%)***

*due to omission of some Main Streams from the original project proposals, these were changed to above figures in * mark.

2. Types of Turbines

The following types of turbines are widely used for power generation.

1. Reaction Turbine
    a. Francis
    b. Propeller
        i. Kaplan
        ii. Dariez
        iii. Bulb
2. Impulse Turbine
    a. Pelton
    b. Cross flow

Francis

Pelton

Kaplan

Deriaz

2.1 Types of Turbines in relation to Principle features- Head and Flow rate

In selection of hydro turbines, starting parameters are, first, Net Head and then Flow Rate.

Various designers and manufacturers have been able to plot graphs based on their own experience, and with time and application of advance technology such graphs keep changing. Usual today's trend is to go for high speeds, which has been made possible due to development of metallurgy, production technology and other techniques, and which tends to lower costs, too.

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Nevertheless, customer requirements are very important for the consultant and the contractor in determining the exact turbine, because each turbine has its own characteristics, and thus restrictions and limitations.

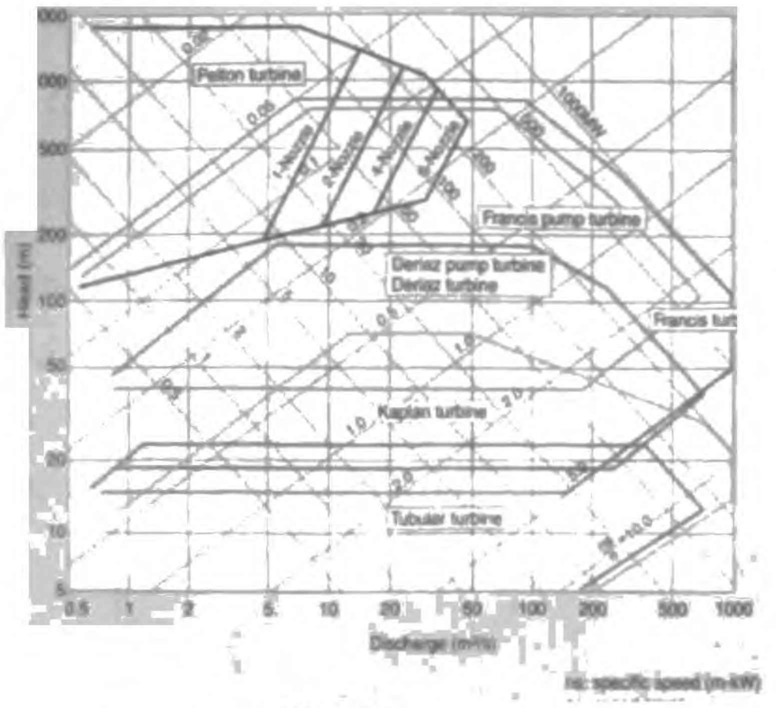


Fig. 2-1
By Mitsubishi Heavy Industries, Japan

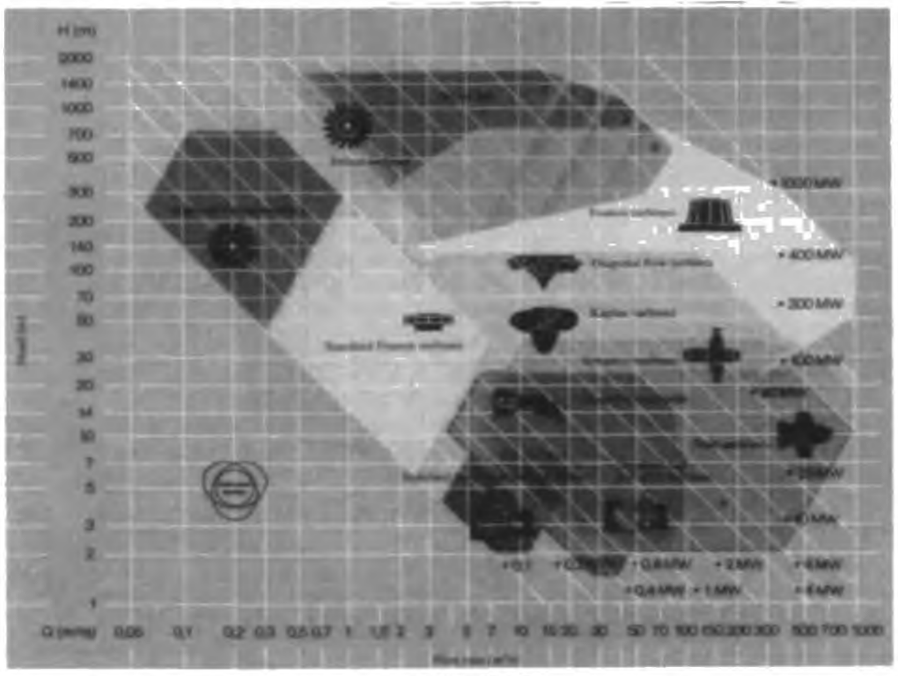


Fig.2-2
By Escher Wyss, Germany

2.2 Comparison of Relative Efficiencies of Various types of Turbines

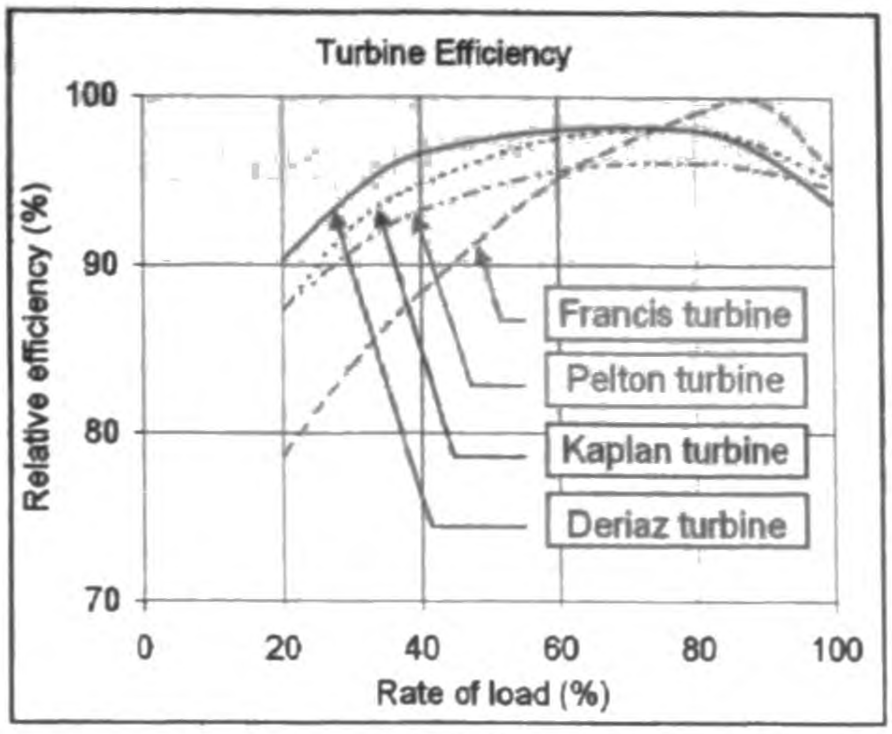


Fig 2-3
By MHI

Accordingly, Francis turbine has the highest efficiency, but within a short operational range.

2.3 Specific Speed and Specific Speed limit of a Turbine

2.3.1 Specific Speed

Definition:
Specific speed is the speed of a geometrically similar turbine which would develop unit power (1kW) under unit head (1meter).

While such a turbine is imaginary, it is called 'specific turbine', specific speed is always expressed in formulas. There are more than six different mathematical expressions.

Out of these, in case of UKHP the following formula has been used.

$$N_s = N \frac{P_t^{1/2}}{H^{5/4}}$$

*Ns - Specific Speed	75.5 m-kW
N - Rated Speed of Machine	600 rpm
P - Power of Turbine	77,000 kW
H - Net head	473 m

*to be calculated later

Specific speed is useful in classifying types of turbines and characteristics of turbines within types.
It influences in selecting: dimensions of turbine, spiral casing, draft-tube; weight and shape of turbine; and turbine setting level (or submergence).

2.3.2 Specific Speed Limit

Specific Speed limit is the maximum specific speed of a turbine, depending on the head.

Example: In case of a Francis Turbine, if the following formula by Japanese Electrotechnical Committee JEC-4001-2006 is considered the following graph can be plotted.

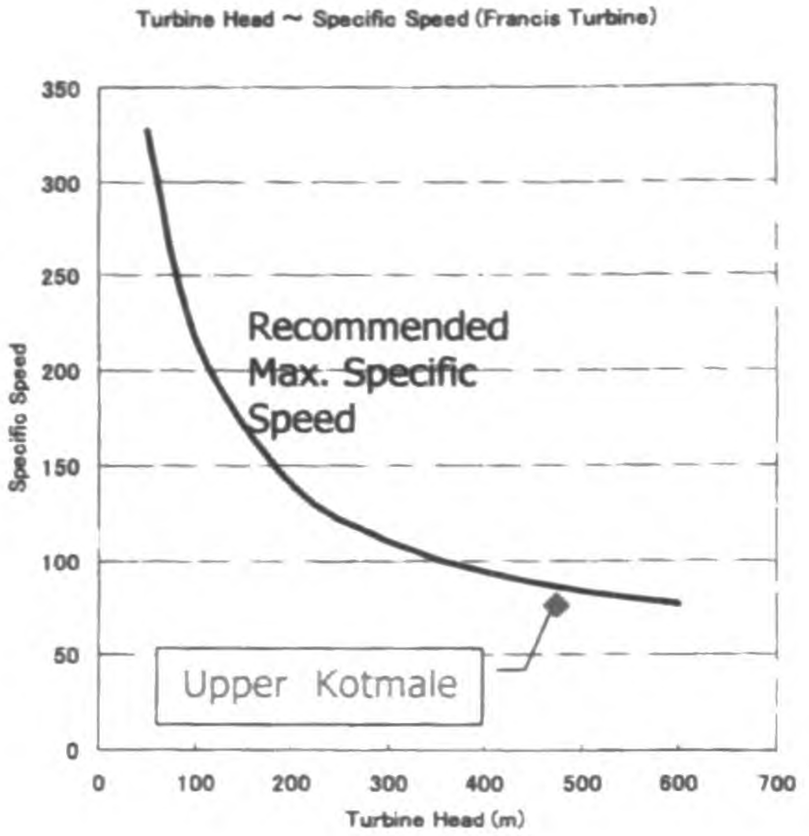


Fig 2-4

$N_s \leqq (23000/(H_e+30))+40$

·····JEC-4001-2006

This is an actual result of manufacturing of turbines by various manufacturers. This equation has been changed several times to make analysis of turbines easy.

2.4 Types of Turbines in relation to Head and Specific Speed

Type of Turbine	Head Range M	Head Range Meters M	Specific Speed Range - m-kW Ns	Specific Speed Equation
Kaplan	Low Head	10 ~ 80	200 ~ 900	$N_s \leq \frac{21{,}000}{H+20} + 35$
Dariez	Medium Head	40~ 180	150~ 400	$N_s \leq \frac{20{,}000}{H+20} + 40$
Francis	Wide Application range	50 ~ 800	50 ~ 350	$N_s \leq \frac{23{,}000}{H+30} + 40$
Pelton	High Head	200 ~ 1800	10 ~ 25	$N_s \leq \frac{4300}{H+195} + 13$

Fig 2-5
By MHI

2.5 Comparison between Specific Speed V'S Turbine efficiency

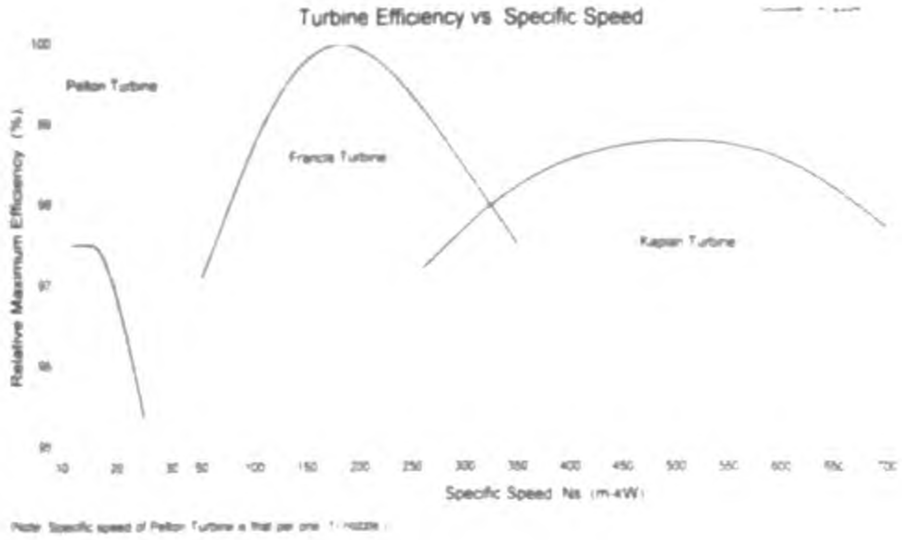


Fig 2-6
By Hitachi

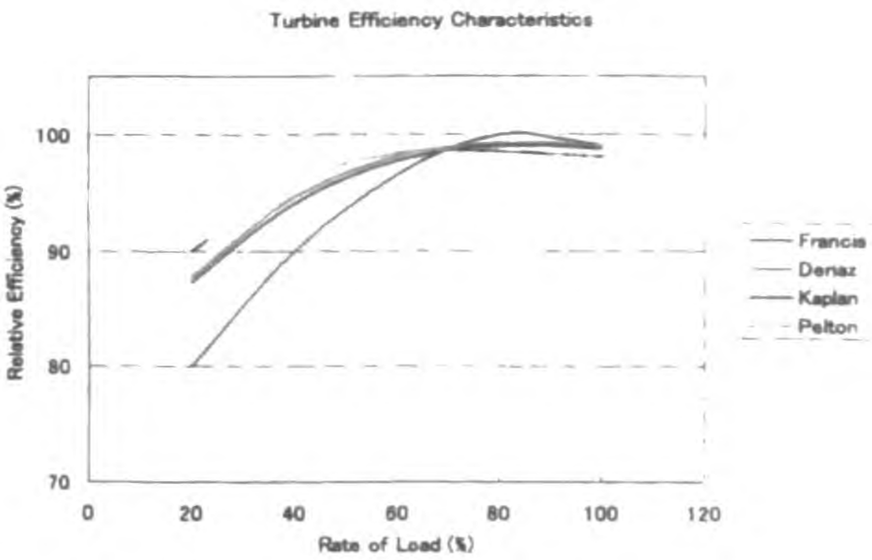


Fig 2-7
By MHI

2.6 Specific Speed vs Head, and types of Turbines

The following specific speed vs. head graph Shows, the types of turbines and shapes of Turbines within a particular specific speed range.

The Pelton is used for high heads and Kaplan for low heads. The Francis occupies between these two.

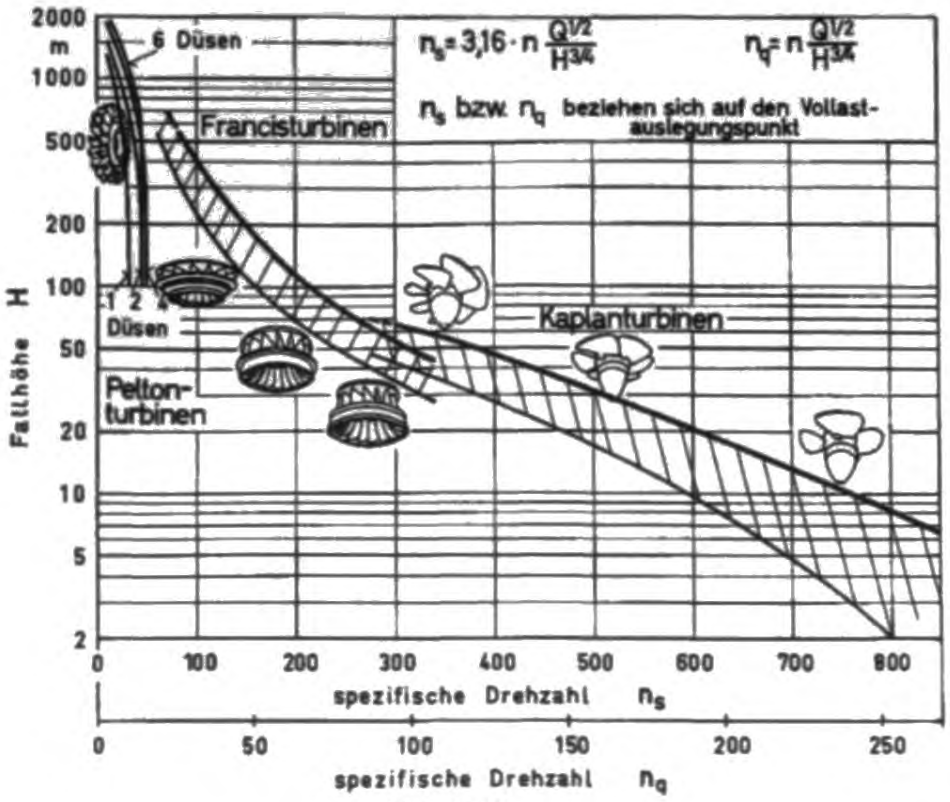


Fig 2-8
By Andriz Hydro

Ns is power based, Nq is flow rate based (Nq, mainly used for pump turbines)

2.7 Specific Speed V'S Shapes of Various Types of Francis Turbines, depending on Heads

The shape of the turbine runner profile is so important that improper runner profile can lead to drop of efficiency, formation of vortex and, thus, cavitation. Therefore, the designers have to consider profile geometry and aerodynamic properties.

The Bid Document Volume 2, Clause 2.1.7.3 specified "The contour of runner vane shall be designed to achieve the best possible operation taking cavitation and efficiency into consideration. The minimum thickness of the runner vane outlet shall be not less than 1.0 % of the runner outlet diameter to enable the easy maintenance against cavitation erosion as well as silt abrasion. Suitable measures shall be applied against Karman Vortex induced behind the runner vane outlet."

As shown in Fig 2-10, a translatory flow around aerofoil section gives rise to formation of vortex at the trailing edge, due to pressure difference on two sides of the vane. This can gives rise to cavitation, too.

Today's advance technology has made possible higher efficiencies by making the runner vanes thinner, particularly at the trailing edge. But, this is too dangerous, because too thin vanes tends to early cavitations. Moreover, thin vanes make repairs difficult. Therefore, the efficiency should be compromised to a certain extent, depending on the requirement of the client. Having considered this, the consultant has specified trailing edge thickness "not less than 1.0 % of the runner outlet diameter". Thus, while the runner outlet diameter of UKHP runner is 1450 mm, trail edge thickness is 17.4 mm.

When compared with other CEB machines, this runner is the one that generates maximum power per unit weight, at high speed.

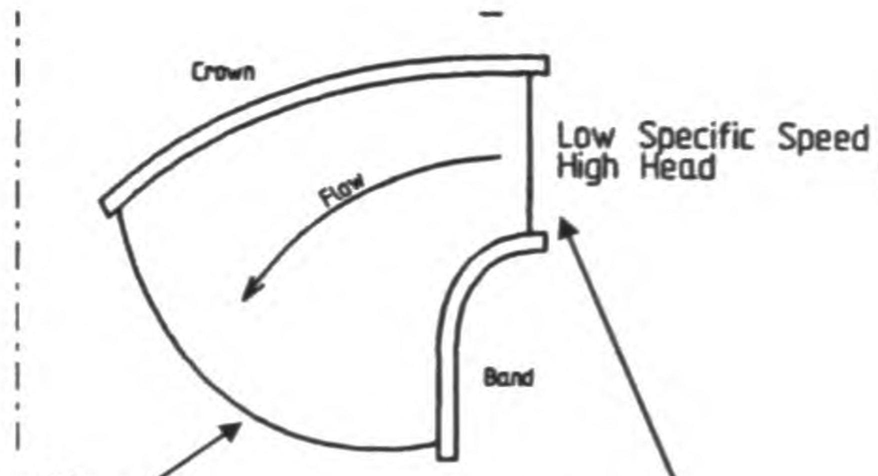
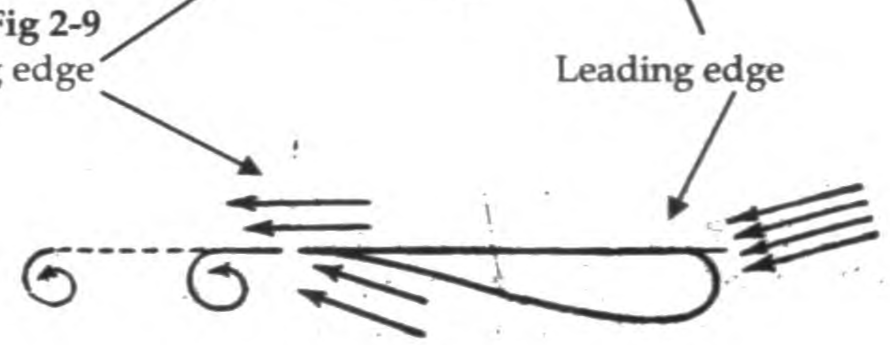


Fig 2-9



Vortex formation
Fig 2-10

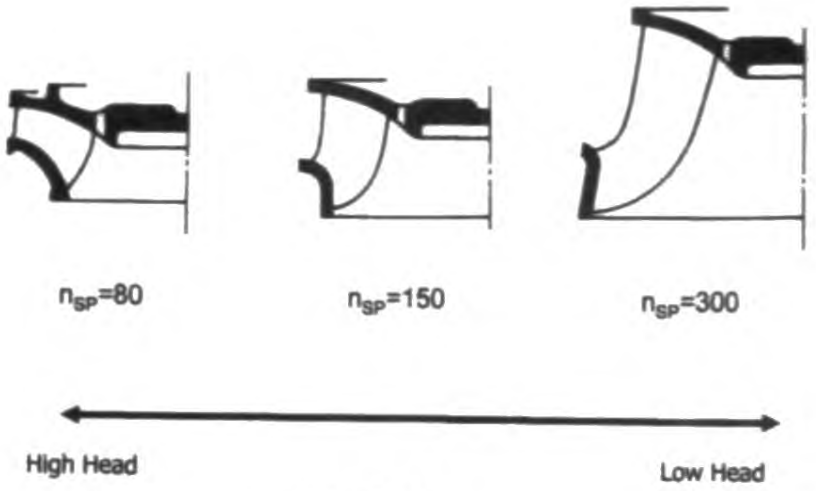


Fig 2-11

Because the UKHP turbine has specific speed (Ns) 75.5 m-kW its shape is close to the above 80.

The UKHP turbine is a very specific case and very rare in this high head, and, therefore, its profile is very important. When compared to many turbines of other power plants in the CEB, its inlet diameter has some waved shape as follows.

Fig 2-12
Model runner

Fig 2-13
Prototype runner showing inlet

Fig 2-14
Proto type runner showing outlet

3. Selection of Turbine for the Upper Kotmale Project-UKHP

When looking at the Figures 2-1, 2-2, and 2-3 both Francis and Pelton turbines are possible.

In case of UKHP the Francis turbine was selected mainly due to the following reasons.

a. Future plan of dam lift of Kotmale reservoir by 30 meters
b. Hydro plants will be used to operate in peak loads in the future when thermal becomes base load. Comparatively, the Francis turbine has better efficiency than that of the Pelton turbine.

If the dam is lifted the Gross Head changes from 491 meter to 461 meter, making this situation possible only for a Francis turbine. The Pelton turbine will become impossible. On the other hand submergence of the turbine will become safer in relation to cavitations issue, even in future dam lift.

Principle parameters are:

Max Gross Head	= 491 m
Net Head at full operation (**H_d**)	= 473 m
Max Plant Discharge (for 2 Units)	= 36.9 m3/s
Max Plant Discharge (**Q**) (for 1 Unit)	= 18.5 m3/s

3.1 Calculation of Turbine Power, Specific speed Limit, Specific speed and Rated Speed

Following formulas are used:

1. Calculation of power output:

   $Pt = 9.8 x Q H_d \eta$ η-turbine efficiency (estimated at 0.899)

   $Pt = 9.8 x 18.5 x 473.1 x 0.899$
   $= 77{,}109$ kW

2. Calculation of Specific Speed limit:

   $$Ns\text{-limit} \leq \frac{21{,}000}{H_d+25} + 35$$

   $$Ns\text{-limit} \leq \frac{21{,}000}{473.1+25} + 35$$

   $= 77.2$ m-kW

3. Calculation of turbine speed:

   $$Ns = N \frac{P\, t^{1/2}}{H^{5/4}}$$

   $$N = \frac{77.2 x 473.1^{1.25}}{77{,}000^{0.5}}$$

   $= 613$ rpm

4. Selection of generator speed:
   The generator speed is selected as close as possible to the above turbine speed value resulting from *maximum Specific Speed*. The general tendency is to go for high speed machines.

   According to the table below, most suitable speed is 600rpm. High speed being 750 rpm can not be selected. Because, it will exceed specific speed limit.

Standard revolving speed of turbine generator

Number of poles	50 Hz	60 Hz
6	1000	1200
8	750	900
10	600	720
12	500	600
14	429	514
16	375	450
18	333	400
20	300	360
24	250	300
28	214	257
32	188	225
36	167	200
40	150	180
48	125	150
56	107	129
64	94	113
72	83	100
80	75	90

$$\text{Revolving speed : } N = \frac{120 \times \text{Freq.}}{\text{No. of poles}} \quad (\text{rpm})$$

Fig 3-1

5. Re-calculation of Specific Speed

$$Ns = N\frac{P\,t^{1/2}}{H^{5/4}}$$

$$= 600\times\frac{77{,}000^{\,0.5}}{473.1^{\,1.25}}$$

Specific Speed = 75.5 m-kW

Rated Speed = 600 rpm (considering the generator parameters)

3.2 Determination of Dimensions of Turbine, Spiral Casing, Draft-tube

These details are required for the civil contractor as preliminary works in order to design power cavern.

3.2.1 Turbine Dimensions

Turbine dimensions depend on Specific Speed which is related to Coefficient of peripheral speed, Ku_1, which is related to turbine inlet diameter, D1:

$$D_1 = 84.6\frac{Ku_1 H^{1/2}}{N}$$

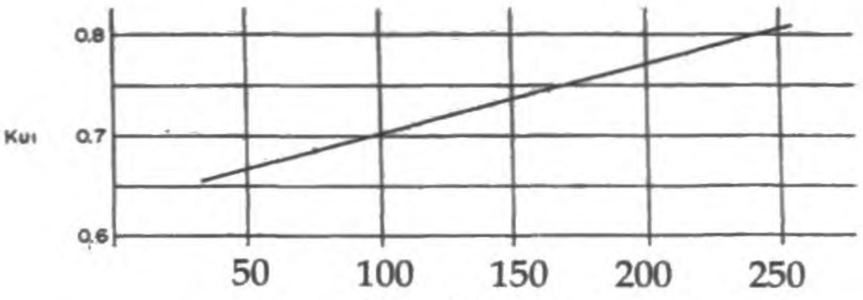


Fig 3-2

Specific Speed vs Coefficient of peripheral speed

After selecting the turbine inlet diameter, D1 all the other dimensions can be obtained from the following charts.

$D1 \infty Ku_1$ Coefficient of peripheral speed
$D2 \infty D1$
$B \infty D1$
$h \infty D1$

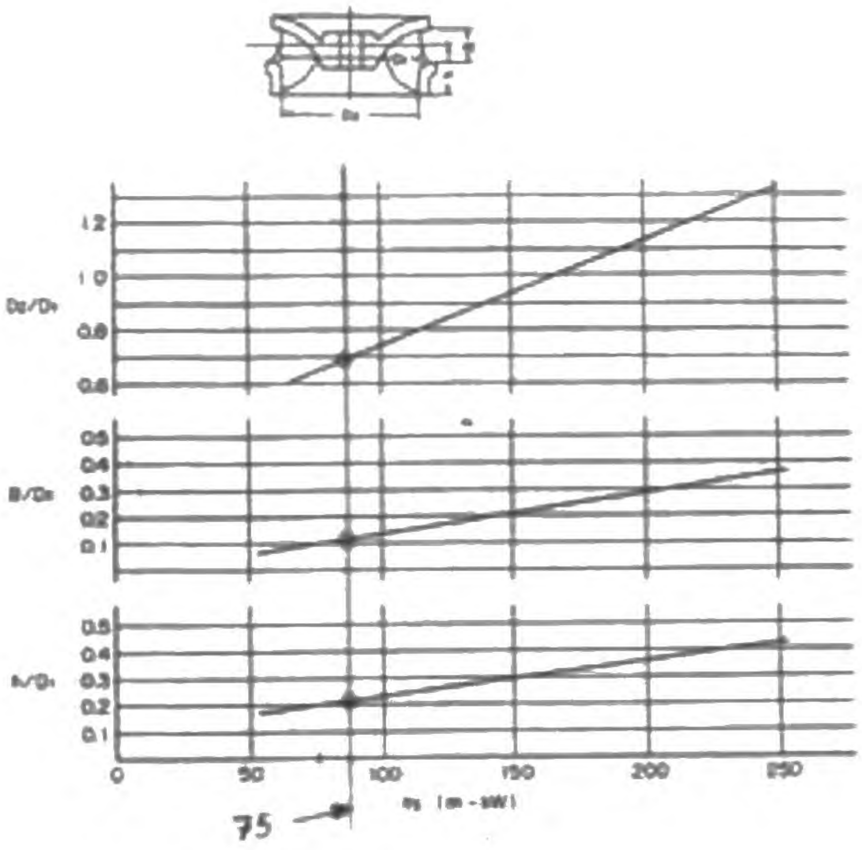


Fig 3-3

3.2.2 Spiral Casing

The governing equations are usually based on turbine inlet diameter, D1:

$a \infty D1$
$b \infty D1$
$c \infty D1$
$d \infty D1$
$L \infty D1$
$M \infty D1$
$Di \infty D1$

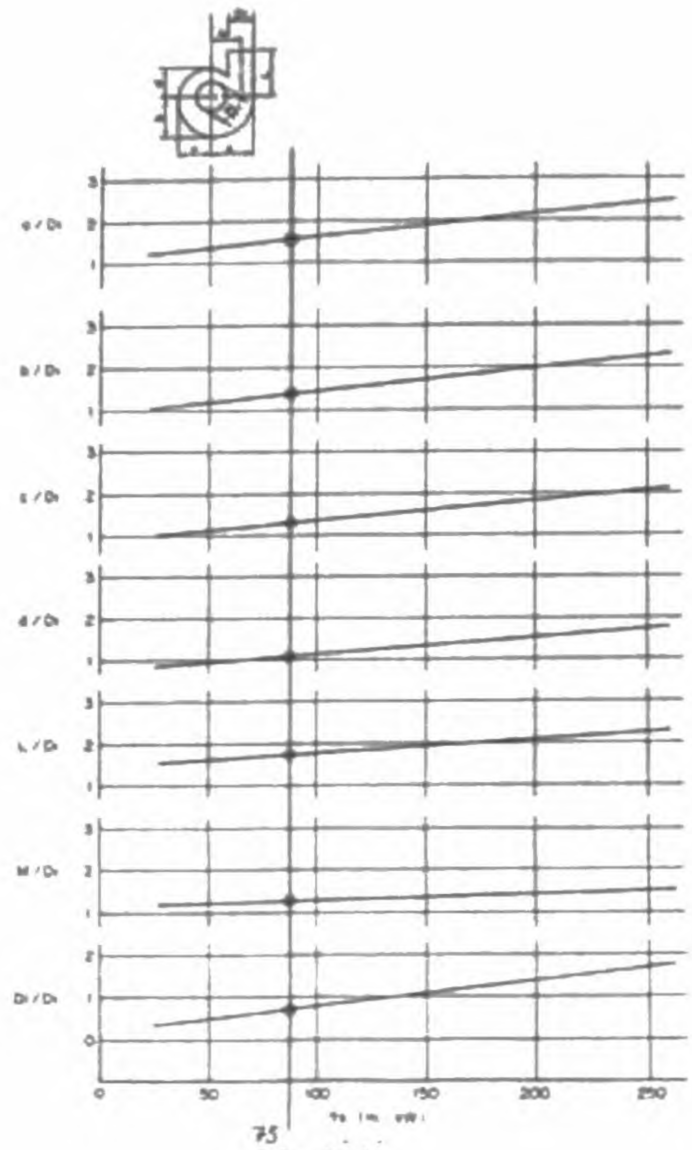


Fig 3-4

3.2.3 Draft Tube

The governing equations are usually based on draft-tube diameter, D3:

Here D3 id considered equal to D2

D3 ∞ H1

D3 ∞ H2

D3 ∞ H3

D3 ∞ L3

D3 ∞ Z

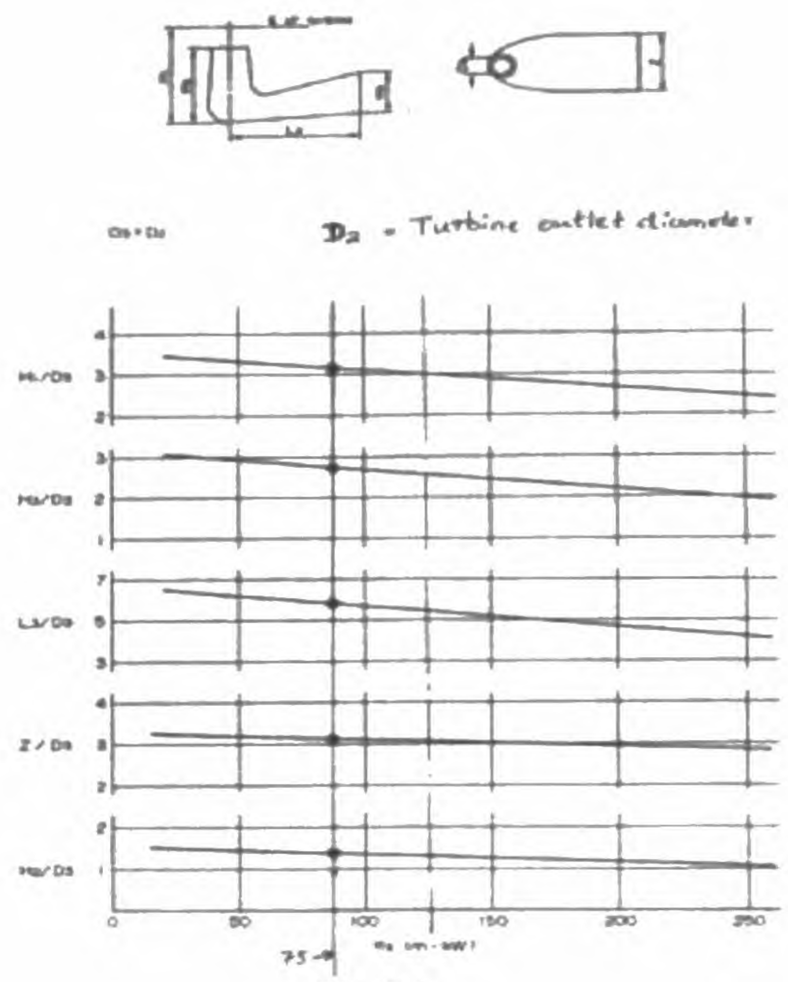


Fig 3-5

4. Turbine Setting Level

In case of reaction turbines, turbine setting center (or submergence level) should be selected to avoid cavitations, because these two parameters are interrelated. Therefore, the turbine setting level should be selected, accordingly. In many cases the turbine should be submerged from the tailrace surface.

Cavitation characteristics is related to Specific Speed, too. In designing work, the following two Cavitations Coefficients are considered.

Critical Cavitation Coefficient (σc)- At this level air bubbles are more, efficiency and output will begin to drop.

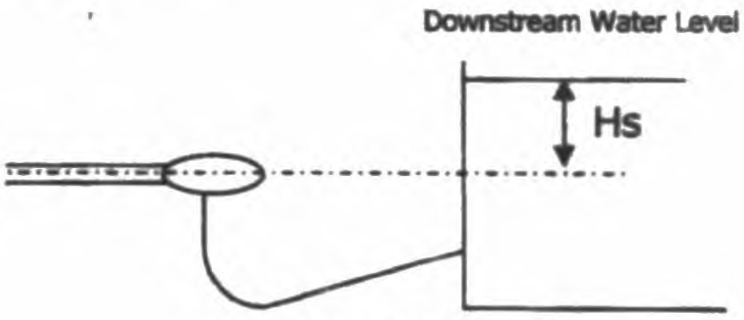


Fig 4-1

$$\sigma p = \frac{Ha-Hv-Hs}{H}$$

Ha - Atmospheric pressure at site

Hv - Saturated water vapour pressure

Hs - Draft head

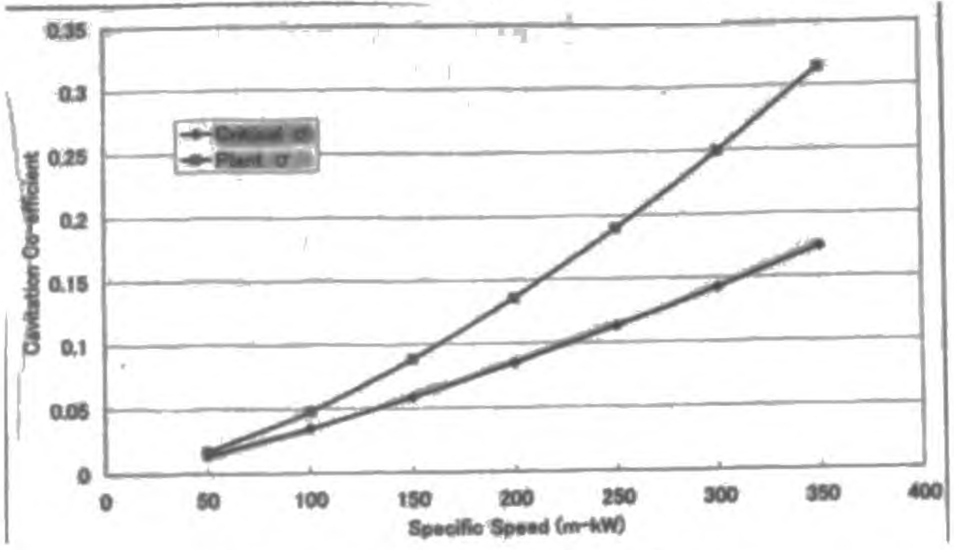
Plant Cavitation Coefficient (σp) - actual turbine setting level of turbine or operational level.

$\sigma p = (1.3\text{-}1.5)\ \sigma c$

Because above both are related to Ns, the following formulas can be used. (Ref Fig 4-2)

$\sigma c = 0.0348\ (Ns/100)^{1.283}$

$\sigma p = 0.0477(Ns/100)^{1.283}$



Specific Speed vs. σc and σp
Fig 4-2

Ultimately for safety of the turbine the actual turbine center is set at the level corresponding to this Plant Cavitation Coefficient σp.

In case of UKHP:

H - Net head 473 m
Ha - Atmospheric pressure at site 9.95 mAq
Hv - vapour pressure at 30^{0C} = 0.43 mAq
σc - 0.0242
σp - 0.0293

Hs - Draft head

$$\sigma p = \frac{Ha-Hv-Hs}{H}$$

Hs= Ha-Hv- σp

Hs= -0.47

But for further safety the submergence level has been selected as 14.00 meter

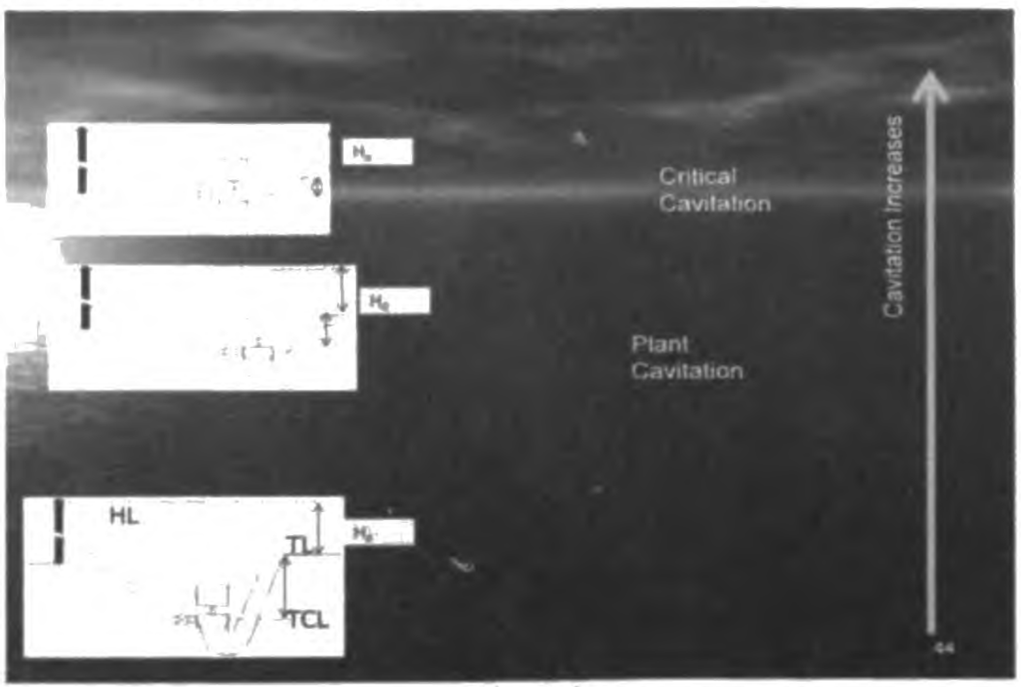


Fig 4-3

5. Turbine Model Test

The performance characteristics of a full-scale turbine can be predicted from the results of tests performed on a homologous scale model in the laboratory. Homologous turbines are geometrically similar turbines.

In compliance with the international code of model acceptance test, "Hydraulic turbines, storage pumps and pump turbines-Model acceptance tests" (IEC Publications No. 60193-1999), a model Test was performed at MHI, Takasago Factory, between October 15-20, 2009.

The model test was carried out for simulated data in consideration of actual operational data of the prototype turbine. Finally, these results have been translated into prototype turbine design.

The following tests were performed:

1. Power and Efficiency Tests
2. Cavitation Tests
3. Pressure Pulsation Tests
4. Steady State Runaway Test
5. Dimensional Checks of Model Turbine
6. Calibration

Model runner outlet diameter was 290 mm or 1/5 of prototype outlet diameter or 1450 mm.

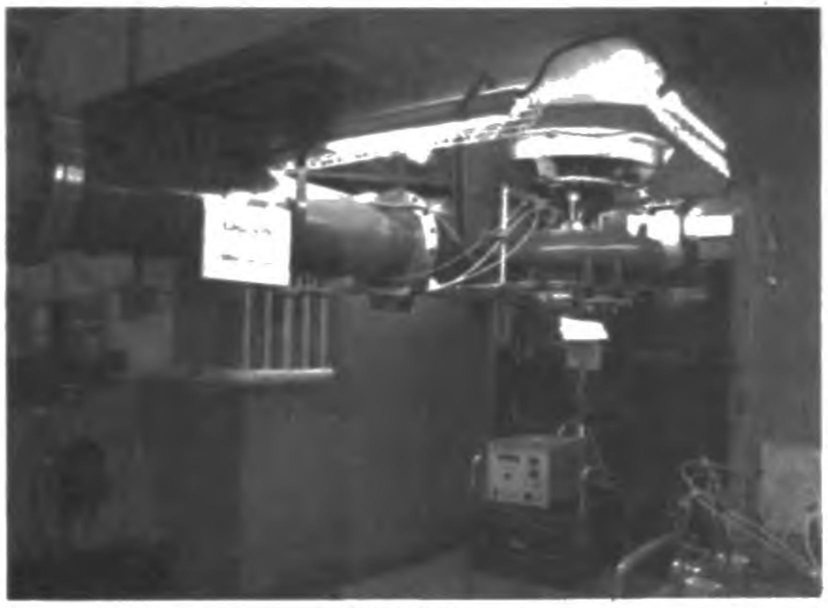


Fig 5-1
Model test equipment

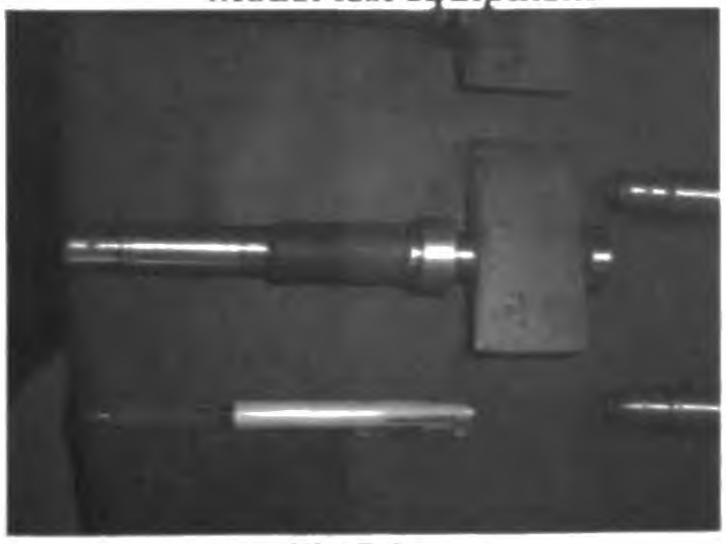


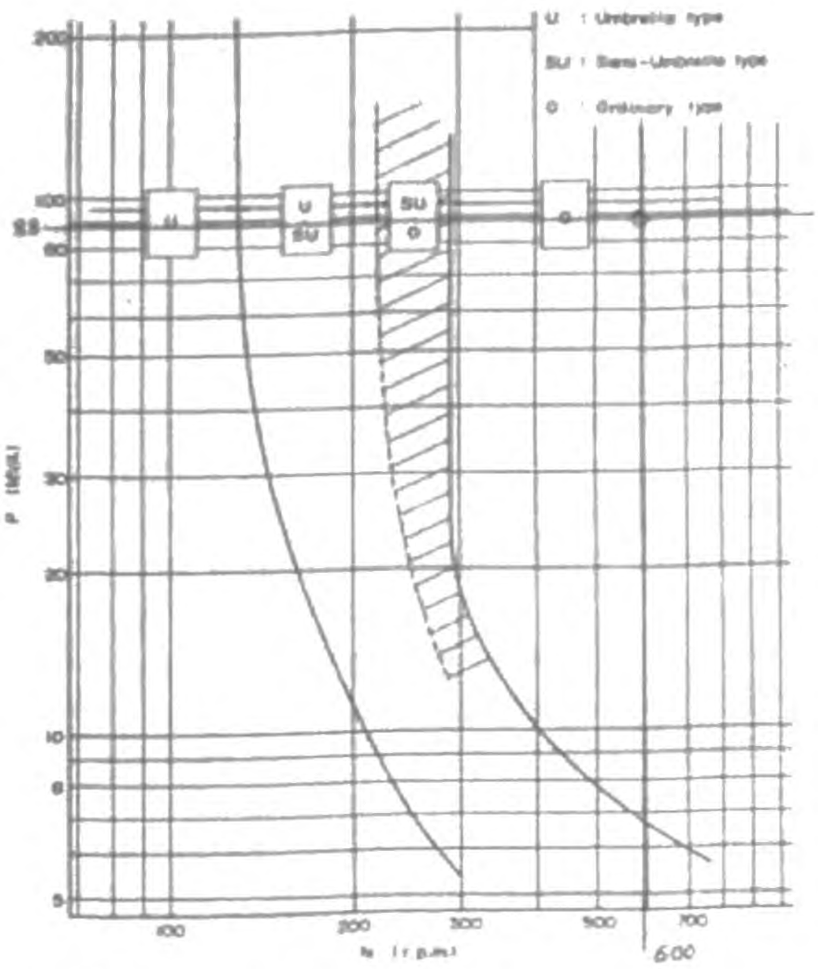
Fig 5-3
Guide vane

Fig 5-4
Prototype Turbine

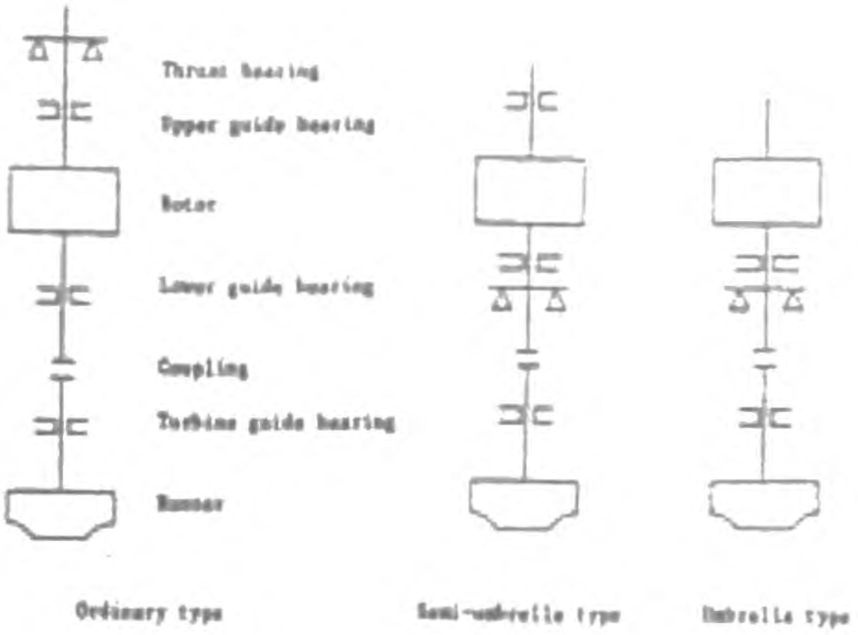
Fig 5-5
Model turbine

6. Bearing Arrangement of Turbine/Generator

Generator output is 77 MW, and with power factor 0.85, generator capacity is 88 MVA.



Speed vs. Generator capacity



Bearing Arrangement of Turbine/Generator

7. Conclusions

In this document, a general outline of turbine selection and its application for the Upper Kotmale is presented. The J-Power has done the preliminary design for the Contract Documents.

Based on the above the MHI has done the detailed design, which will be explained by Mr Nanba, Turbine designer of MHI.

References:

1. Design of electrical and mechanical equipment of hydroelectric power plant (Japanese manual)
2. The Guide to hydropower mechanical design
3. 'Final Design Report' Volume 1, Main Report, June 1995'
4. Bid documents of Lot-04 prepared by J-Power

Turbine Design for Upper Kotmale Hydropower Plant

T. Namba

Abstract: Upper Kotmale turbine, installed capacity of 77,000kW × 2units, has some special features as one of the highest head Francis turbine for conventional hydropower station. In this document, some technical features of the Upper Kotmale turbine, general explanation on selecting the turbine type, and some test results and the verification done during the development and field tests are described.

Keywords: Hydropower, Francis turbine, Turbine type, Model test, Field test

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1. Introduction

Upper Kotmale hydropower station is installed with 2 generating units with 2 identical vertical Francis turbines with unit capacity of 77,000kW. The turbine for Upper Kotmale hydropower station is one of the highest head conventional Francis turbines with net head of approximately 500m. To optimize the turbine performance and to attain the necessary maintainability in the required turbine specification, detail elaborations were required during the designing and development stage. The comparison between the Pelton turbine and Francis turbine, which are the two main options applicable for 500m class hydro turbine, and some technical features applied for Upper Kotmale turbines are explained in this document.

2. Turbine Design for Upper Kotmale

2.1 Turbine Type

Upper Kotmale turbine is the one with effective head of approx.500m and max. output of 77000kW. The type of turbine selected is Francis type which is a type of reaction turbine. For the turbines with effective head of 500m, there are two options we can consider. One is Francis type, and another is Pelton type. Both features are described in Table1, Figure 1, and Figure 2. When we compare the two types of turbines with the same gross head and output, the difference between Francis type and Pelton type are summarized as below.

As Pelton turbine is type of an impulse turbine whose runner has to be installed above the tailrace water level, Pelton turbine cannot make use of the level difference between turbine centre to tailrace water level where reaction turbine like Francis type can make the full use of the water level difference between the upper reservoir and the lower reservoir as the

Table 1- Turbine Type

	Francis (Fig.1)	Pelton (Fig.2)
Type (Fig.3)	Reaction type	Impulse type
Rotational speed	Relatively high	Relatively Low
Size of the Equipment	Relatively Small	Relatively Large
Max. Efficiency (Fig.4)	Relatively high	Relatively low
Part load efficiency (Fig.4)	Relatively low	Relatively High
Effective head (Fig.3)	Relatively high	Relatively Low

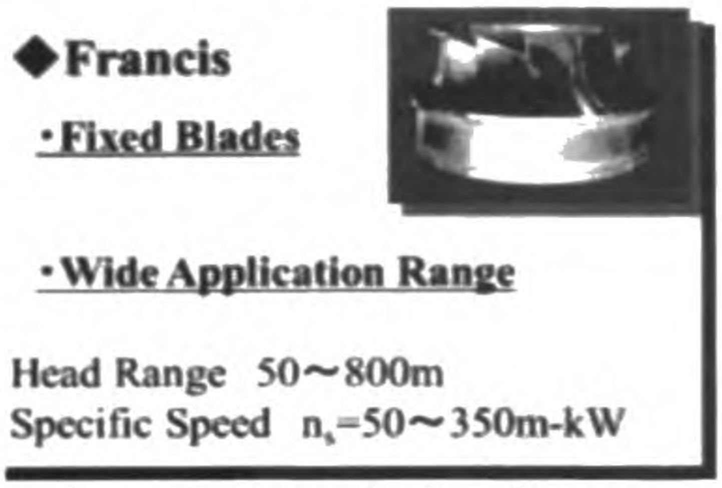

Figure 1 - Francis Turbine Runner

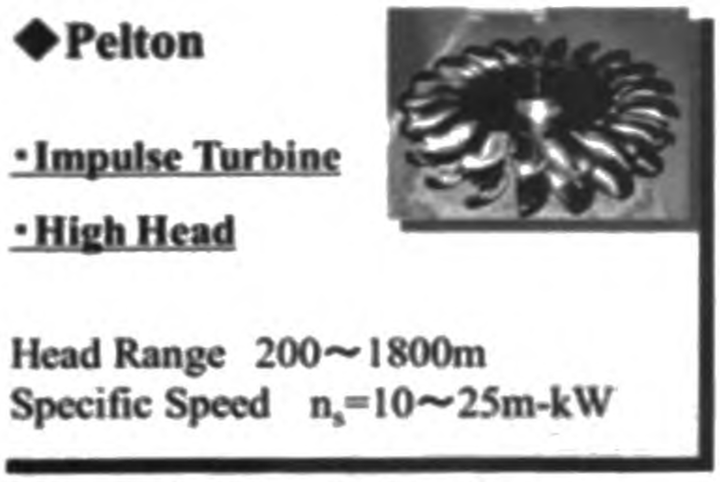

Figure 2 -Pelton Turbine Runner

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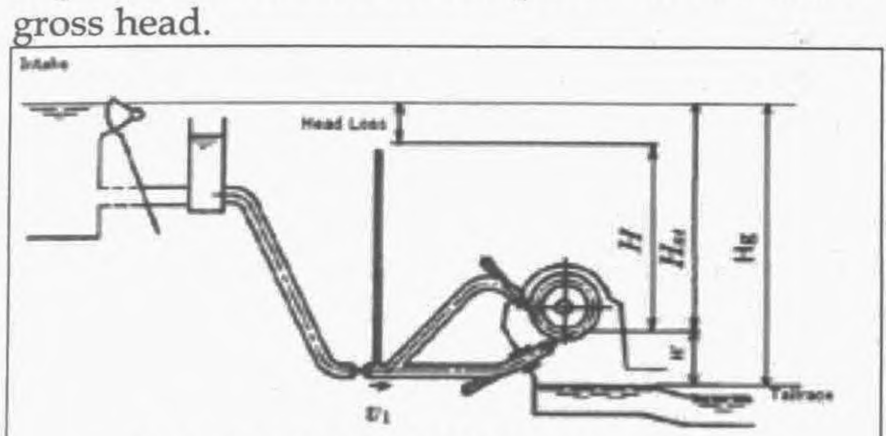
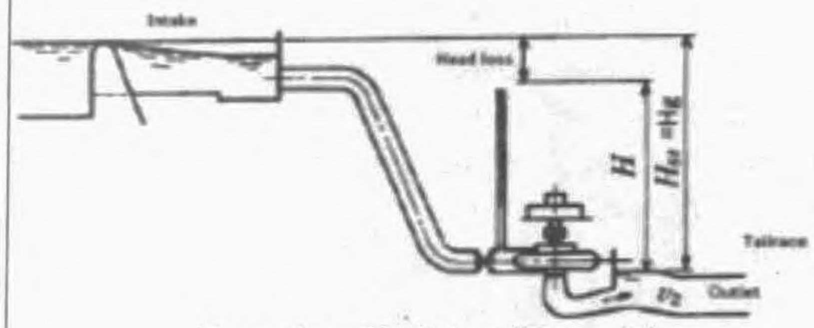
effective head. Brief explanation is given in Fig.3. As reaction turbine can utilize the full head difference between the upper and lower reservoir for power generation, effective head of reaction turbine is higher than that of impulse turbine when compared in the same gross head.



Impulse Turbine (Pelton Turbine)
Effective head : H = Hg-H'-Head loss

H' is the head difference which cannot be utilized for power generation in the case of impulse turbine and usually is approx. 2~4m in water column.



Reaction Turbine (Francis)
Effective head : H=Hg-Head loss-$(V_2^2/2g)$

$(V_2^2/2g)$ is the kinetic energy water flow fromturbine has which is wasted as a loss in the case of impulse turbine and usually is approx.0.2~0.3m.

Figure 3 - Comparison between impulse turbine and reaction turbine.

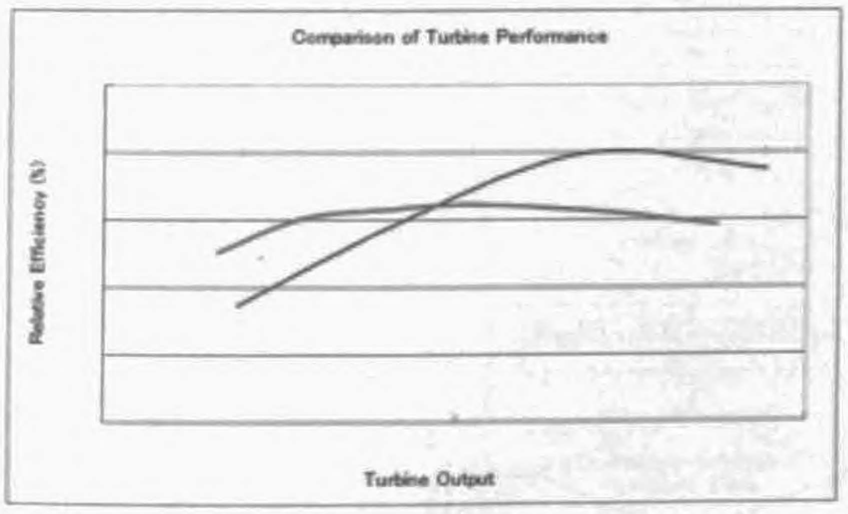


Figure 4 - Comparison of the Efficiency between Pelton Turbine and Francis Turbine

As described in Fig.4, efficiency of Francis turbine is higher than that of Pelton turbine in the range of large output in most cases including the maximum output point. As a result of combination of higher effective head and higher efficiency in large output region, maximum power output of Francis turbine is higher than that of Pelton turbine under the same gross head and the same flow rate. On the other hand, Pelton turbine has wider operation range by changing the number of nozzles depending upon the unit load and has higher efficiency in the range of part load operation.

In summary, when we compare the two options for 500m head class turbines,

- Francis type has advantages when the units are mainly operated at full load operation as it has higher efficiency and has more capacity compared to Pelton Type.
- Pelton type has advantageswhen the units are mainly operated at part load range as it has higher efficiency in low load operation range.
- Francis turbine is more compact than Pelton turbine.

For the turbines mainly operated at full load like Upper Kotmale, Francis type has advantages from the view point of generation capacity and its size as explained above.

2.2 Specification of Upper Kotmale Turbine

Upper Kotmale is the power station mainly responding to the peak power demand, and is mostly operated near the full load condition. Therefore Francis type turbine was selected as the type of turbine. Specification of turbine is summarized in Table-2.

Table 2 - Specification of Upper Kotmale Turbine

Effective Head (m)	487.31	485.31	473.09	458.08
Discharge (m^3/s)	17.19	17.22	17.74	17.7
Output (kW)	77000	77000	77000	76000
Speed (min^{-1})	600			
Specific Speed(m-kW)	75.5			

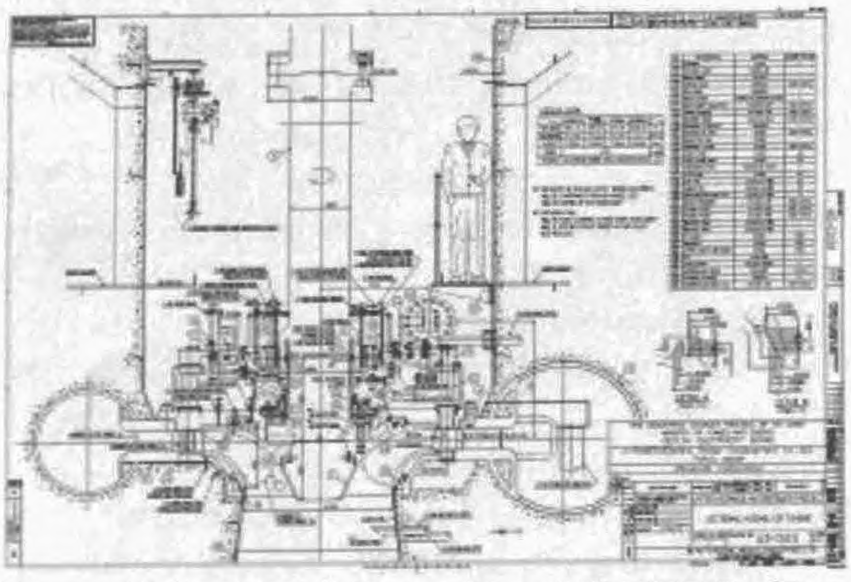


Figure 5 - Sectional Assembly of UKHP Turbine

The position of Upper Kotmale in comparison with our experience record is shown in Fig.6.

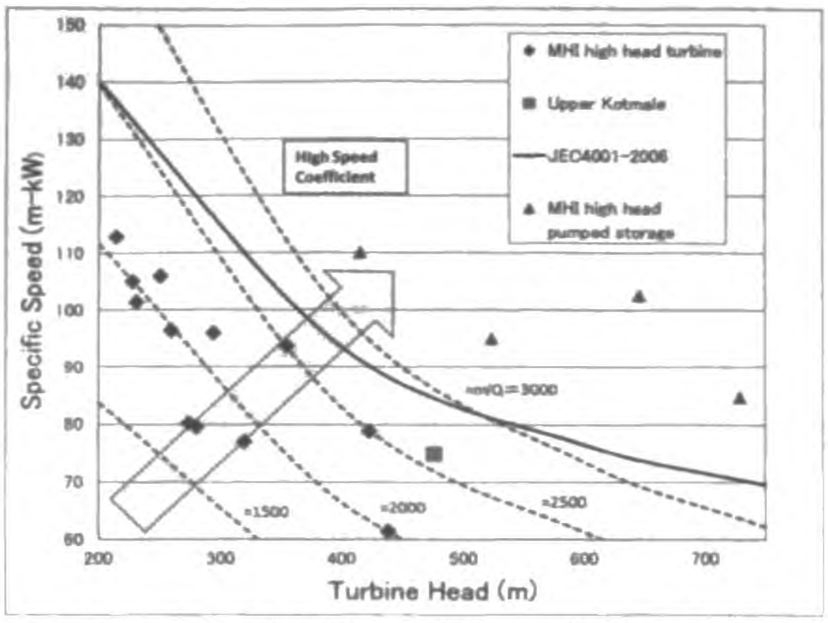


Figure 6 - Position of Upper Kotmale in comparison with our supply record

The turbine for Upper Kotmale is designed at the rated head of 473.09m with rotational speed of 600r/min, and 77,000kW output. Besides the pumped storage units, turbine for Upper Kotmale is designed for the highest head in our experience as a Francis type turbine. Specific speed is a function of turbine head and as the head goes higher, specific speed becomes lower. In the case of Upper Kotmale, as it is a high head turbine, specific speed is 75m-kW which is rather low in the application range of Francis Turbine. When we talk about the speed of the unit, $n\times(Q)^{0.5}$ can be used as a "high speed coefficient" which is more directly related to the speed of turbine and can be used as a parameter to evaluate the speed of the turbine regardless of the turbine head. Constant high speed coefficient lines are drawn in Fig.6 in blue dot line. Limit of speed coefficient based on JEC 4001 is also indicated in black solid line. As shown in Fig.6, upper limit of the coefficient for conventional Francis turbineaccording to JEC-4001 lies between 2500 and 3000. The coefficient of Upper Kotmale is the highest one as conventional hydro, but still within the limit defined in JEC 4001.

Speed coefficient is related to the following issues.

- Relative size of the machine

Diameter of therunner of turbines with same output and onthe same coefficient line is equivalentregardless of the head. As the coefficient goeshigher, the turbine becomes smaller.

- Required suction head

Roughly speaking,turbines on the same coefficient line requiresame NPSH (submergence) from the viewpoint of cavitation performance.

- Strength of the equipment.

As the coefficientgoes higher, stress working on the turbineincreases.

Turbine for Upper Kotmale is a low specific speed machine as a Francis turbine. But from the view point of speed coefficient, it is a high speed turbine whose specification meets the latest guide line for the high head Francis turbine.

2.3 Turbine Design Flow

To achieve the performance of UKHP turbine, development of homologous model turbine was done by utilizing the latest CFD analysis technology. Turbine design flow during the model development process is indicated in Fig.7. In the initial design stage, overall dimension of turbine is determined based on our in-house data by using the following design parameter.

H : Turbine Head
Q : Turbine Flow
N : Rotational Speed

Those are the basic design parameter of hydraulic turbine that shall be settled prior to starting the turbine design. After determining the overall dimension of turbine primary performance evaluation is done based on our past experience and after that, optimization of each component is started. By trial and error, evaluation is repeated considering the matching interface of each component in the design process to optimize the overall performance of turbine.

Some examples of the latest technology for optimization of turbine performance introduced for Upper Kotmale turbine are described in 2.4.

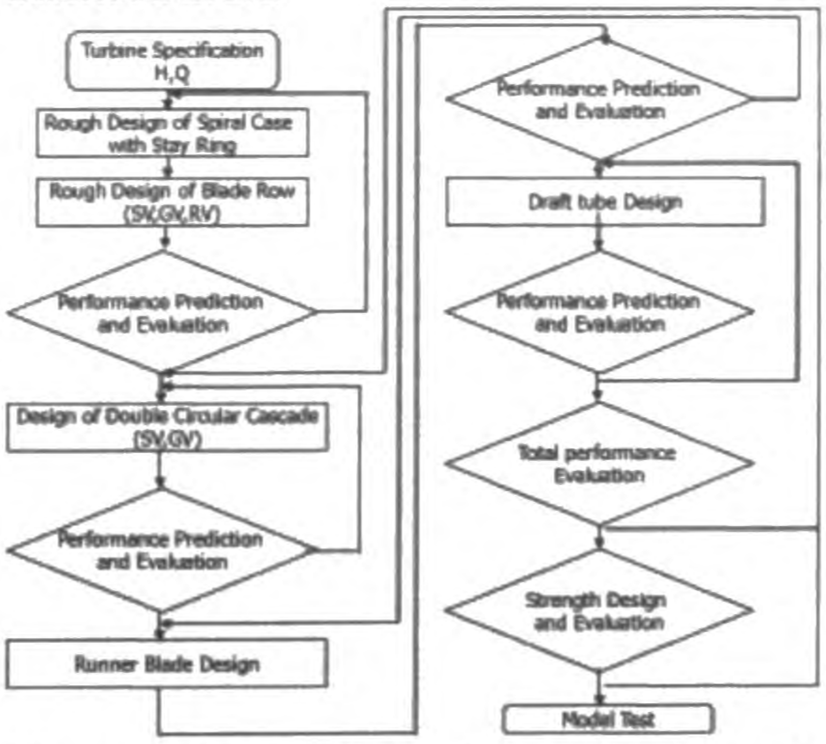


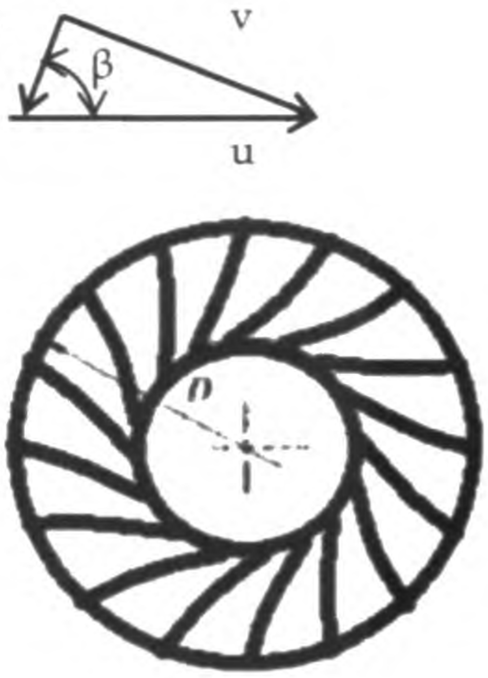
Figure 7 - Turbine design flow in development stage

2.4 Latest Design Concept Introduced for Upper KotmaleTurbine

For the optimization of the turbine performance, various studies have been made by numerical simulation and model tests. By utilizing the know-how accumulated through years of development, homologous scale model for Upper Kotmale was developed, manufactured, andtested.

2.4.1 Runner Inlet Shape

At the runner inlet, water flow from guide vane outlet flows into the runner. As the runner itself is rotating, relative flow angle shall be considered as the in-flow angle to the runner inlet.



where

V : water flow flow the GV outlet
u : velocity of runner inlet
w : relative flow to runner inlet
β : relative in-flow angle to runner inlet

Figure 8 - Velocity triangle at runner inlet

Water flows into the runner through the blade rows i.e., stay vanes and guide vanes. In the flow passage located near the wall, flow velocity is low due to the boundary layer. (On the surface of the stationary wall, flow velocity is zero. In the boundary layer, flow has velocity gradient due to the viscosity of the liquid). Therefore, the flow velocity near the upper and lower wall of the water passage is slower as described in Figure 9.

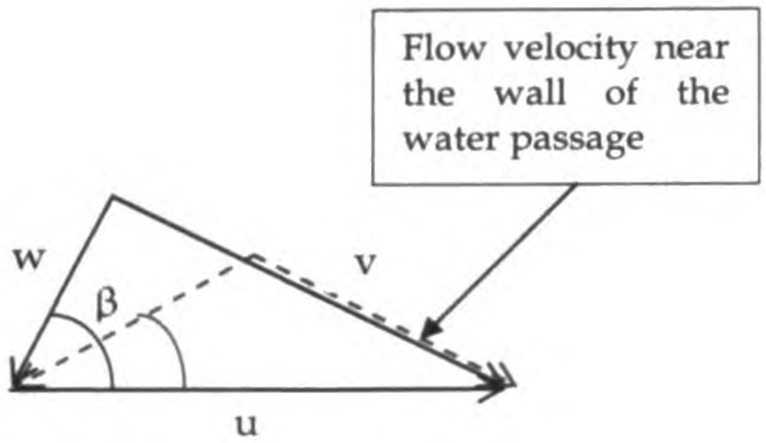


Figure 9 - flow velocity triangle at runner inlet near the upper and lower wall

As the flow velocity near the wall is low, relative in-flow angle to the runner becomes acute at near the runner crown and shroud side as shown in Figure 10.

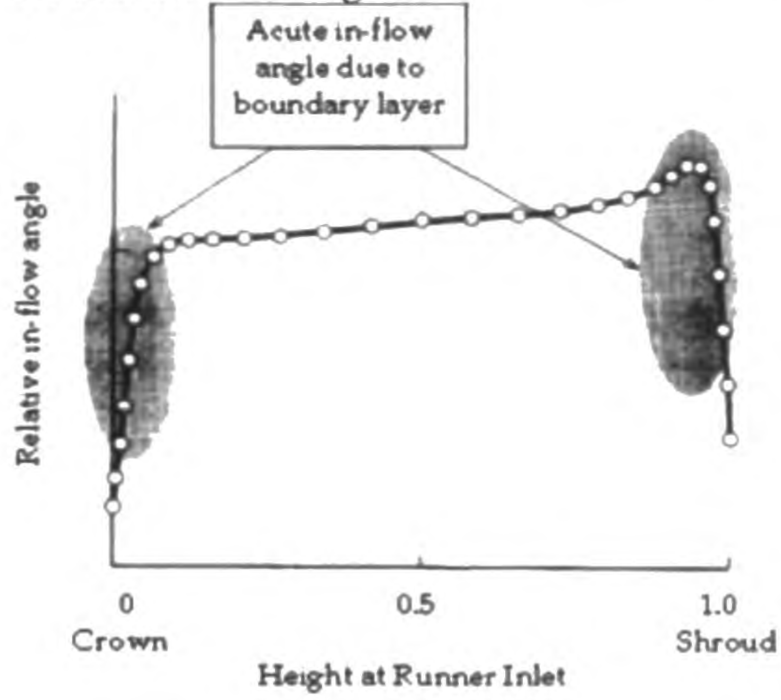


Figure 10 - In-flow angle distribution at runner inlet

Runner blade inlet angle is designed to have variation considering the in-flow angle variation from the shroud side to crown side into account based on the CFD analysis result.

Figure 11 - Runner on balancer

2.4.2 Runner Outlet Shape

In the turbine runner, due to the combination of centrifugal force by rotation, centrifugal force due to curvature of the passage, as well as the Coriolis force, complicated secondary flow is generated. This secondary flow causes a large loss when it collides with the main flow. For the optimization of the turbine performance, how to control the secondary flow in the runner is a key.

Fig.12 is a picture of the model runner for Upper Kotmale. The blade outlet of the runner for Upper Kotmale has convex shape. This is to control the secondary flow that could be a source of loss at the runner outlet.

Figure 12 - Model Turbine Runner

3. Turbine Model Test and Field Test

In October 2010, turbine performance model test was carried out at Takasago Research and Development Center located in Hyogo, Japan. Fig.13 is the model turbine assembled in the test stand. The items confirmed through model test are,

- Turbine Performance (efficiency)
- Cavitation Performance
- Pressure Fluctuation (Draft Tube)
- Air Admission
- Runaway Speed

Figure13 - Model Turbine Assembled in Test Stand

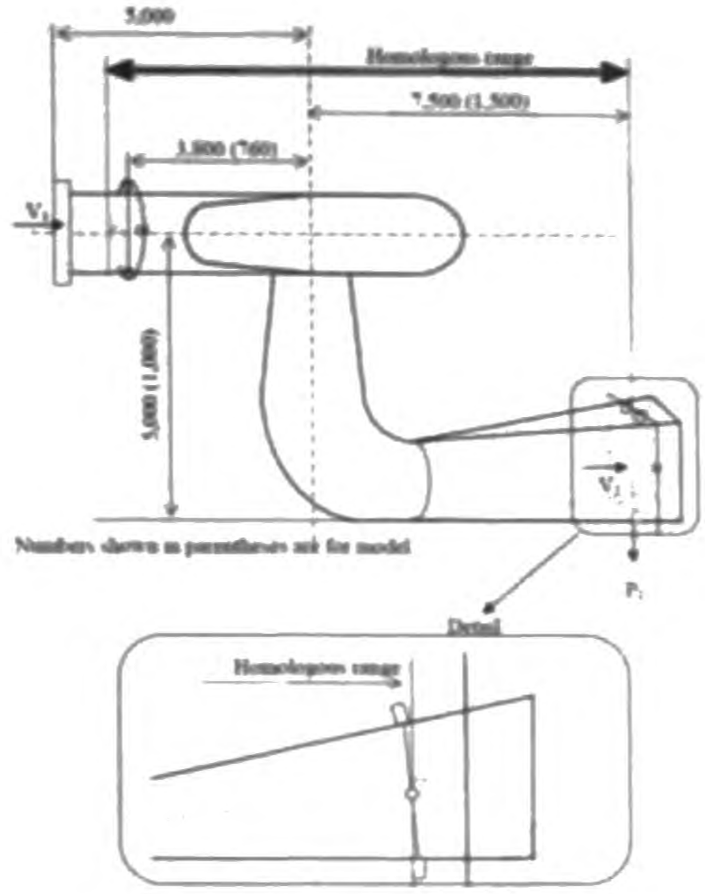


Figure 14 - Homologous Range of Turbine Model Test

The scale ratio of the model is 1/5 of the prototype and its homologous range is as described in Fig.14. The model test was done in accordance with the international code of model acceptance test, "Hydraulic turbines, storage pumps and pump-turbines - Model acceptance tests" (IEC Publications No. 60193-1999).

3.1 Guide Vane Opening and Output Test

Relation between the guide vane opening and generator output is shown in Fig15 in comparison with the calculated value based on the turbine model test result.

Figure 15 - Relation between Servomotor Stroke and Generator Output
(Comparison between calculation result and field test result)

As shown in Fig.15, field measurement result is in good agreement with the calculated result.

3.2 Air Admission Test and Pressure Fluctuation in Draft Tube

To confirm the necessity of air admission to the draft tube to improve the operating condition at part load, air admission test was done during the model test and the pressure fluctuation level and turbine efficiency were evaluated.

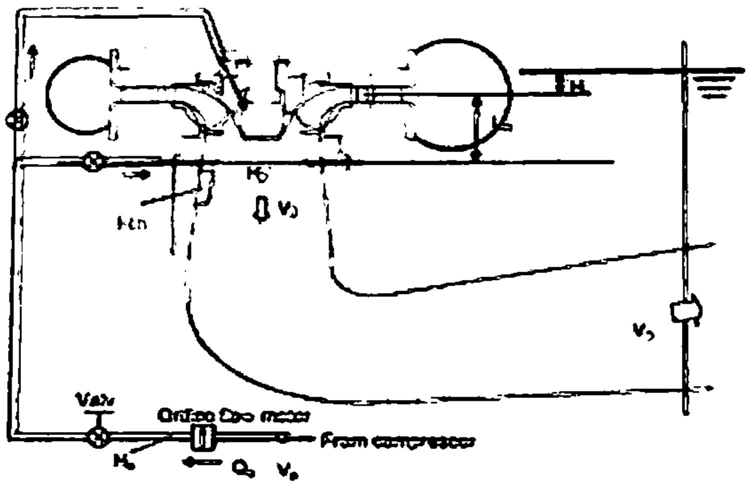


Figure 16 - Arrangement of the air supply pipe on model stand

The measurement result of pressure fluctuation level in draft tube by changing the air flow rate is as shown in Fig.17. Pressure fluctuation level in Fig.17 is normalized in percentage of the effective head. As shown in the graph, pressure fluctuation level decreases by increasing the air flow rate especially at the partial load operation. But the pressure fluctuation level is low enough even without air admission and the maximum value is approx. 2% at 40% guide vane opening. Through the test, it was confirmed that the pressure fluctuation level in draft tube is well below the operation criteria of 4% without the air admission.

In Fig.18, to evaluate the effect of air flow rate on turbine performance, turbine efficiency test results are compared by changing the air flow rate. As the air flow rate increase, turbine efficiency drops.

Based on the air admission test result, we decided not to provide air supply line on the prototype because of the following test result.

- Pressure fluctuation level is low enough even without the air supply.
- Turbine efficiency drops when air flow rate increases.

This is because pressure below the runner outlet rises by supplying the air to the upper draft tube. This pressure rise results in less pressure difference between the inlet and outlet of runner.

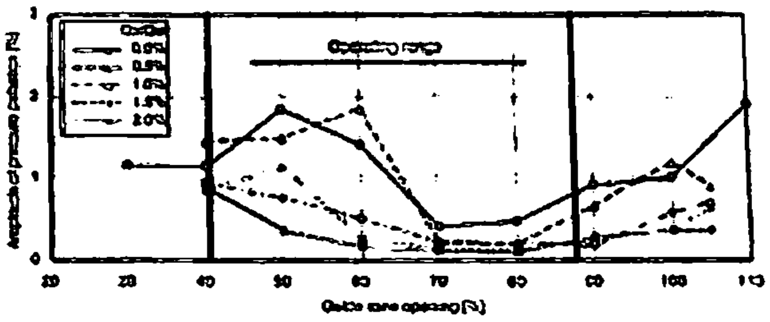


Figure 17 - Pressure fluctuation level in draft tube (air admission test)

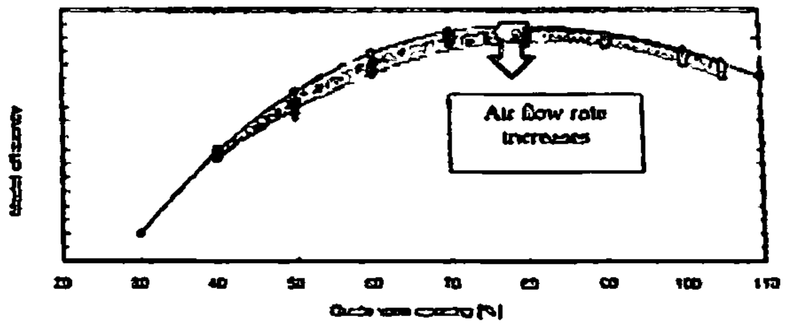


Figure 18 - Turbine Efficiency during Air Admission Test

To enable the installation of the air admission pipe in the future if necessary, provision of the embedded pipe with stop flange has been made on the prototype.

Comparison between the model test result and the field measurement result of the pressure fluctuation in draft tube is as summarized in Fig.19. It was confirmed that the turbine of Upper Kotmale can be operated stably without air admission to the draft tube as was expected based on the model test result.

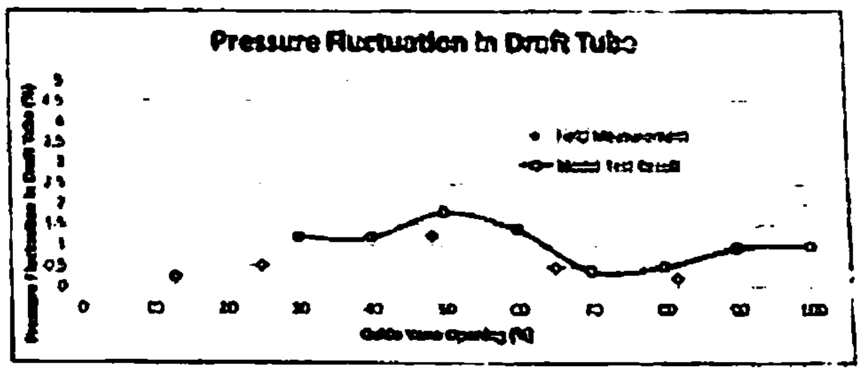


Figure 19 - Pressure Fluctuation in Draft Tube

4. Conclusions

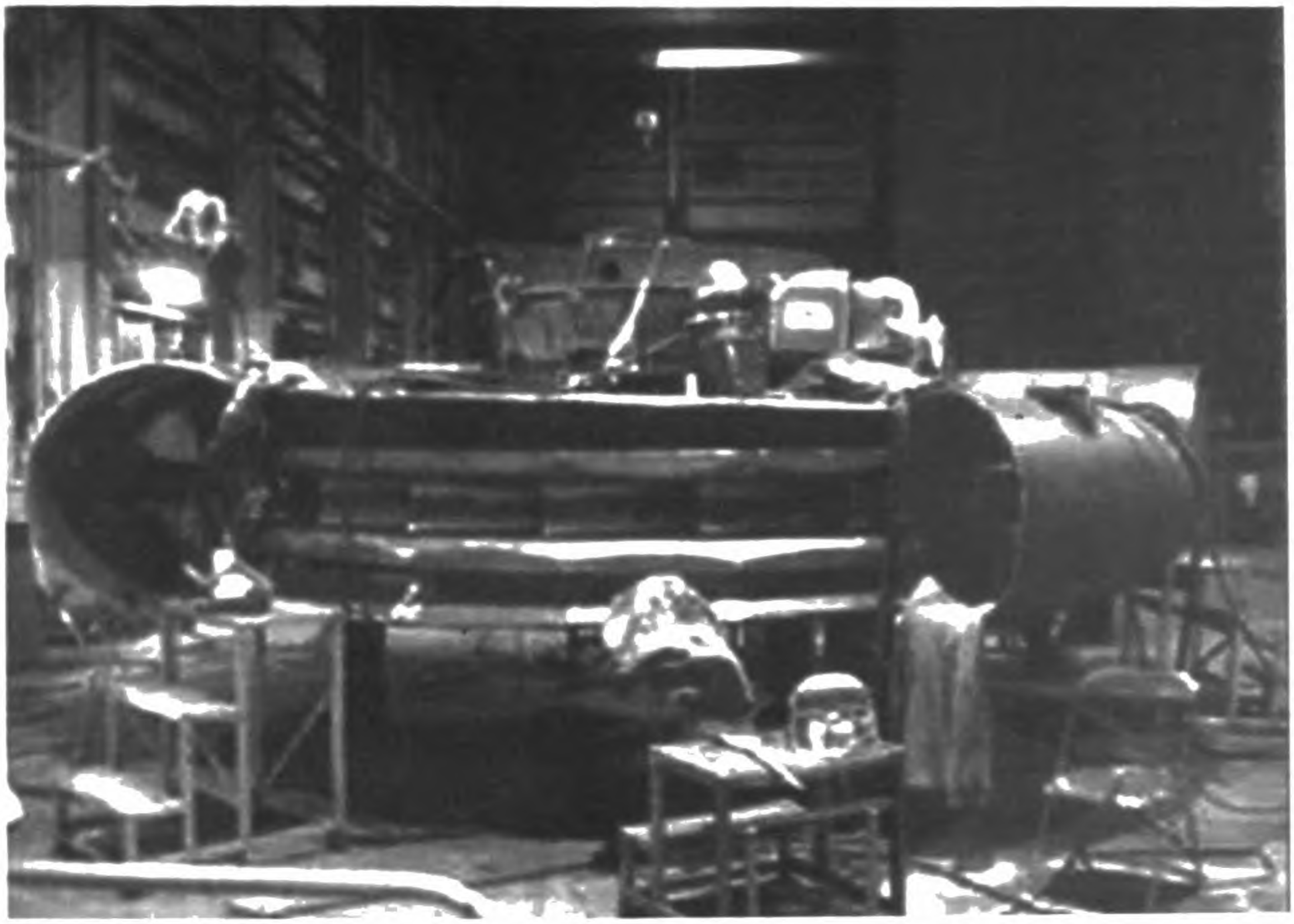
In this document, design features and characteristic points of Upper Kotmale turbine are presented.

As Upper Kotmale is one of the highest head conventional Francis hydro turbines, density of energy of water flowing through the turbine is higher than that of low head turbines, and turbine size becomes compact for its power generation output. In design stage, attentions were paid to the accessibility to the positions where the maintenance is required to enable the required service and maintenance work as easy as possible.

The main units of Upper Kotmale have been handed over to CEB and are contributing to the power generation service in Sri Lanka.

5 References

1. K. Miyagawa, Development of High Performance Hydraulic Turbine Based on New Design Concept. Mitsubishi Heavy Industries, LTD Technical Review, Vol.37 2000.



Spiral Casing of the Turbine