A STUDY OF THE MECHANICAL EQUIPMENT PROPOSED FOR SANTO ANTÔNIO POWER PLANT BASED TO THE REPORT: "ESTUDIOS DE VIABILIDADE DO AHE SANTO ANTÔNIO, VOLUME I-TEXTO-REVISÃO 1 (TOMO II)".

THE STUDY IS BASED ON THE FLOW DURATION OBSERVATION PRESENTED BY ODEBRECHT: TOMA D'AGUA E CASA DE FORCA SECÃO AND DURATION CURVES BASED ON HYDRAULIC DATA IN ESTUDIO DE VIABILIDADE RECEIVED FROM ODERBRECHT.
FURTHER INFORMATION FROM EVALUATION OF TURBINES (ODERBRECHT-ANEXO A-ITEM 14.6) HAS BEEN STUDIED AND COMMENTED IN THIS REPORT.

CONTENT OF THE STUDY:

1. **CHOISE OF TURBINE TYPE WITH COMPARIZON BETWEEN KAPLAN AND BULB TURBINES.**

2. **TURBINE SETTING, OPERATIONAL LIMITATION, CAVITATION, LOSS OF PRODUCTION CAUSED BY LOW EFFICIENCY DUE TO LARGE VARIATION OF TAIL RACE LEVEL (AFFECTS THE ECONOMY OF THE PROJECT).**

3. **HYDRAULIC FORCES ON TURBINES AND POWER HOUSE STRUCTURE DURING EXTREME LOW TAIL RACE LEVELS INCLUDED A BRIEF ANALYSIS OF THE NEGATIVE THRUST BEARING FORCES ON GENERATOR DURING RUNAWAY.**

4. **SAND EROSION AND HARD SURFACE COATING.**

5. **POSSIBLE GOVERNING INSTABILITY PROBLEMS DURING OPERATION DEPENDING ON CONNECTED GRID.**

6. **MATERIAL QUALITY AND STANDARDS, BRAZILIAN PRODUCTION.**

1. CHOISE OF TURBINE TYPE WITH COMPARISON BETWEEN KAPLAN AND BULB TURBINES.

The main dimensions of Kaplan- and Bulb turbines have been presented in volume II in the feasibility study for Santo Antônio project made by Furnas Centrais Eletricas SA.

A comparison between Bulb turbines and Kaplan turbines shows that the distance between the units will be increased by a factor of approximately1.75 if Kaplan turbines are chosen. Further the depth of excavation for the bottom of the draft tube will increase the excavation cost.
However, the heavy generators for Kaplan units would solve the frequency control problems as describes later in this report. A certain number of Kaplan units might be discussed if the frequency stability problem cannot be solved with Bulb turbines.
However, the conclusion so far, without defined data for the electric grid with possibility to run the power plant on isolated grid goes in favor of choosing Bulb turbines for all turbines in Santo Antônio.
However:
In the ODEBRECHT report: ANEXO A – ITEM 14.6: SELECÃO DO TIPO DE TURBINA, a list of advantages is made both for Bulb turbines and Kaplan turbines.
I would like to add some comments to the lists both for Bulb turbines and Kaplan turbines.
A thoroughly analyses of these problems will be made later in my report.

Bulb turbines:

There will be high hydraulic forces transferred to the concrete especially towards the hatch opening for dismantling of the runner chamber and runner. These forces are so high that it has been part of the limitation of maximum head for using Bulb turbines because of movement of the concrete that may jam the runner chamber.

During maximum flood in Santo Antônio the downstream pressure increases to a level close to upstream level. Then the hydraulic pressure on draft tube and runner chamber will be very high so these parts must be reinforced. (See: Chapter 3.)

Bulb turbines have a very small inertia GD^2 with a time constant of approximately 1. Sec. only. This causes a fast increase in speed during load rejection with the danger of a high upstream directed force on the back trust bearing during runaway ("the runner is climbing").

Further the governing stability makes it impossible to operate on isolated grid except for at low load with a strong derivative influence on turbine governor and a voltage governor with voltage/speed droop limited between $\pm$ 1.5 Hz. These limitation-problems must be analyzed in detail with the electric engineers because problems may also occur when connected to long transmission lines.

The overhung shafts are exposed to fatigue and care must be taken not to expose the flange neck to corrosion. The shaft seal should be located on the flange to keep the shaft fillet of the flange protected against corrosion.

Kaplan turbines.

Following advantages may be added.

Runner chamber is embedded in concrete and can allow for the high downstream pressure.

High inertia mass on the generator allows for operation on isolated grid.

Conclusion for choice of turbine type including my remarks:

The choice of Bulb turbines for the Santo Antônio project as made in the feasibility study, is correct because of much higher cost of installing Kaplan turbines.

2. TURBINE SETTING, OPERATIONAL LIMITATION, CAVITATION, LOSS OF PRODUCTION CAUSED BY LOW EFFICIENCY DUE TO LARGE VARIATION OF TAIL RACE LEVEL (AFFECTS THE ECONOMY OF THE PROJECT).

Based on information in ODEBRECT´s report on Hydrology as presented in fig.1, the range of head and submergence during operation is very wide and restrictions of operation of the units must probably be made in order to avoid cavitation erosion on the runners.

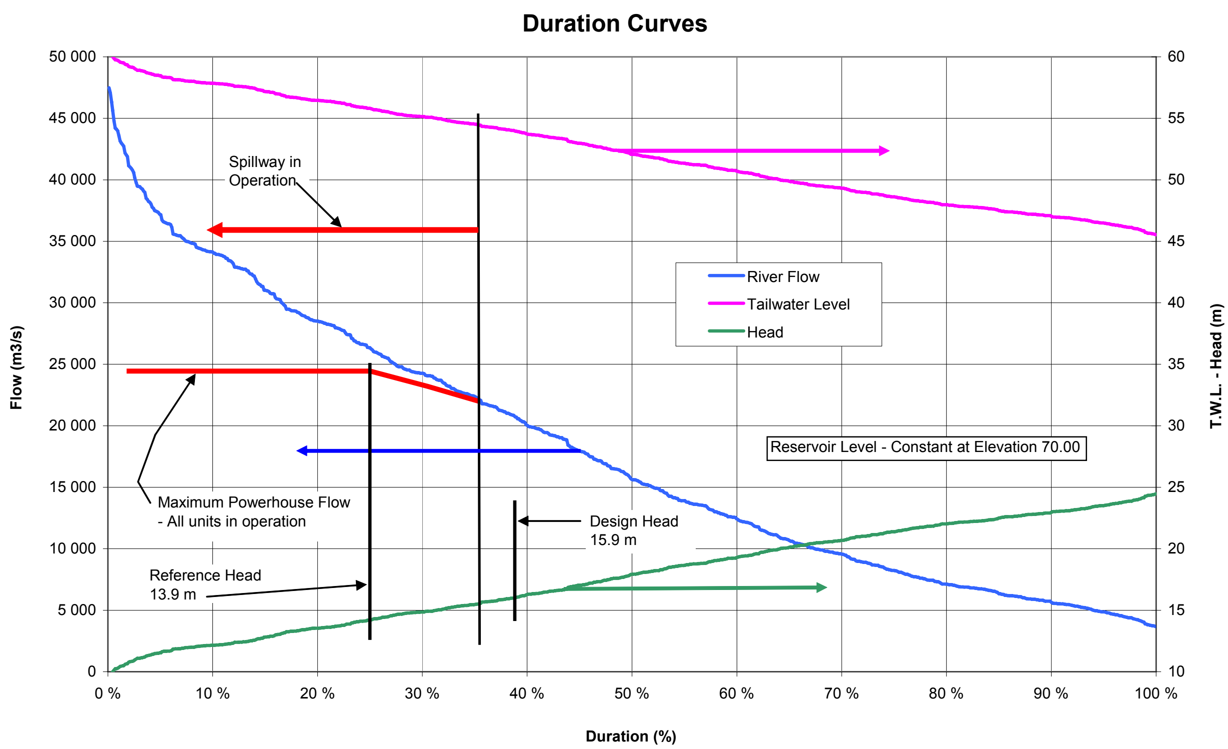

Fig. 1 Duration curves of Santo Antônio received from Giuseppe Stevanella COLENCO.
Ref. ODEBRECHT, TOMO I : Cap.7. ESTUDIOS HIDROMETEROLÓGICOS E FISIOGRÁFICOS.

NECESSARY TECHNICAL DATA TO BE RECEIVED FROM TURBINE MANUFACTURERS FOR SETTING OF THE BULB TURBINES FOR SANTO ANTÔNI.

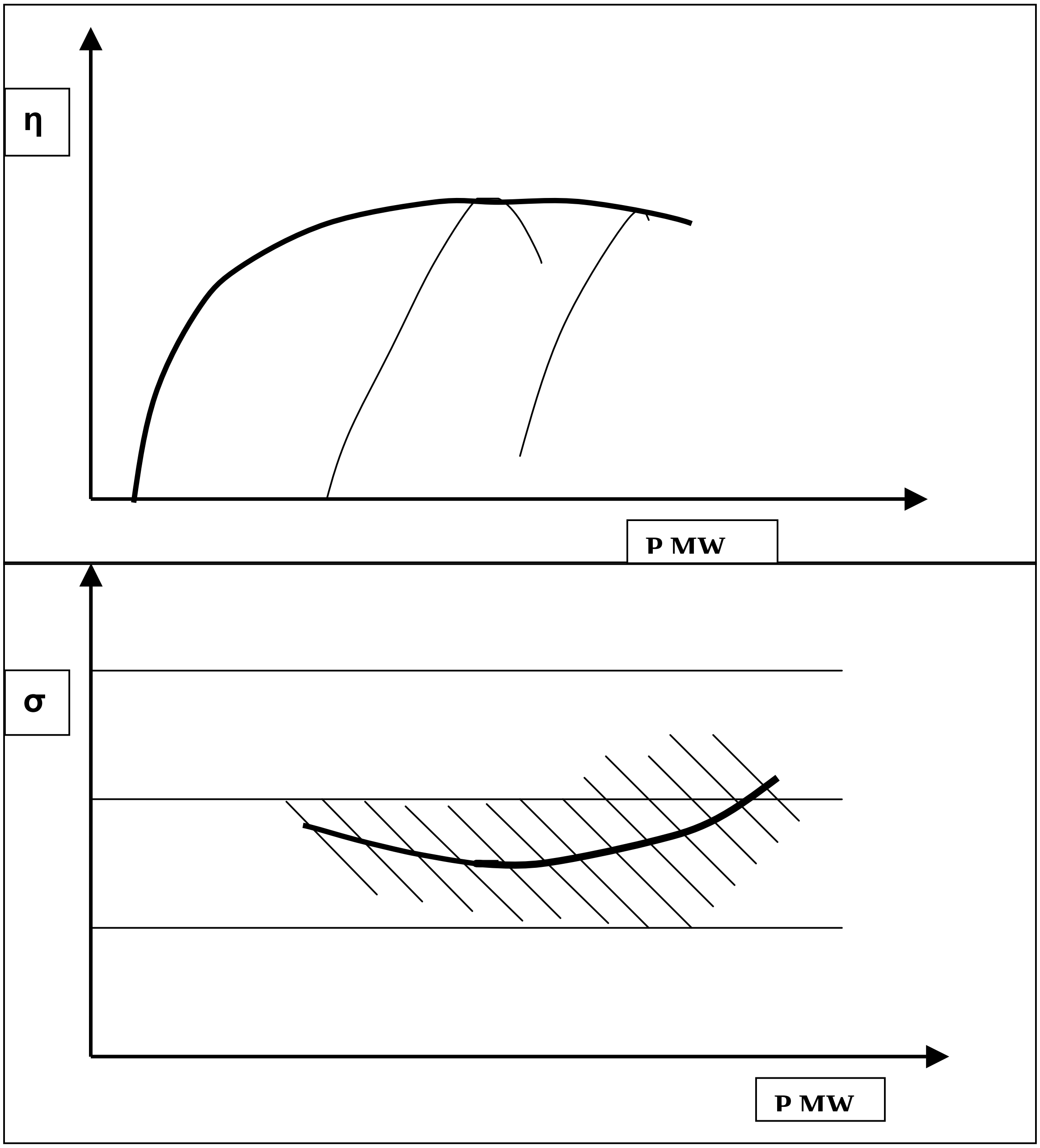

Fig.2 Illustration of needed efficiency diagram and σ values as function of turbine power for the Bulb turbines for Santo Antônio. Note similar diagrams are needed for different operational heads based on a Characteristic Diagram (Hill Chart Diagram) as illustrated later in this chapter.

It is necessary to get reliable information about a safe setting of the units for all possible operational conditions in order to avoid cavitation. This is also important because the danger of cavitation increases if sand erosion is expected.
The power will be limited by the generators at lowest tail race level.
However, it is necessary to receive the critical values of σ=NPSH/Hn for safe operation at different guide vane openings "on cam" especially for operation at lowest tail race levels as illustrated in fig.2.
This information should be collected from more potential turbine manufacturers.
(The technical data for the turbines at reference head are: P= 71.5 MW, n=81.8 RPM, H=13.9 m. The head for best efficiency operation is H_{bep}=15.9 m.)

EXAMPLE OF CONTROL OF NECESSARY SUCTION HEAD:
Assuming a value of **σ=1.0** and the turbines are running at design head =15.9 m then **NPSH=σ*Hn=1.0*Hn**. Further the barometric pressure is assumed to be10 m and the vapor pressure = 0.5m i.e. **h_b-h_{va}=9.5 m**, then **Hs=NPSH-9.5=Hn*σ -9.5 m**.
At design head **Hn=15.9** m**.**
The minimum distance from the **top of the runner chamber** will then be as follows for a turbine with the chosen design head, power and speed:

Hs=**15.9*1.0-9.5**=**6.4 m** to top of the runner chamber **or Hs=10.4 m** referred to the turbine center of an 8 m runner according to IEC Norms.

However, the tailrace water level is about 21.5 m at reference head and Hn =13.9 m with a rated power of P=71.6 MW and no restriction in the turbine power should be expected at the reference head.
Because the net head will be as high as 70-45.6=24.4 m at lowest tailrace level the turbine power must be limited to the generator capacity. THIS PROTECTION MUST BE INCLUDED IN THE CONTROL SYSTEM.
HOWEVER, INFORMATION ABOUT σ VALUES FOR THE TURBINE OPERATION AT LOWEST TAILRACE LEVEL MUST BE CONFIRMED AT MAXIMUM ALLOWABLE GENERATOR LOAD.

The equation for calculation of the suction head Hs, when assuming **h_b-h_{va}**=9.5m yields:

$$\mathbf{Hs=h_b-h_{va}-NPSH=9.5-Hn*\sigma}$$

The vapor pressure the barometric pressure **h_b** is assumed to be 10 m, but the vapor pressure **h_{va}** may be higher than 0.5 m under tropical conditions and must be controlled and the value 9.5 must be changed accordingly. (The turbine dimensions have been calculated on a separate SPREAD SHEET from the data given in the report presented by ODERBRECHT as a control of the given dimensions from the manufacturers. This sheet is enclosed. The dimensions are close to the dimensions received from the manufacturers. Data for the efficiency curves in fig. 4 based on fig 3 is also presented in the spread sheet.)
It should be noted that the draft tube pressure will be affected by the small distance between the units at low tail race level, creating a high velocity =C4 downstream of the draft tube outlet: Normal draft tube outlet loss for C_3=3. m/s will be: $\underline{\Delta H=C_3^2/(2g)=0.45}$ m for $C_4 \leq 0.5$ m/s in the river at draft tube outlet. However, it will be reduced to:
$\underline{\Delta H=C_3^2/(2g)-C_4^2/(2g)=0.25}$ m if C4=2. m/s. Then the <u>runner pressure will be reduced by 0.2m</u>. Because of the high velocity the danger of cavitation may increase. (To be discussed.)

LIMITATION OF EFFICIENCY AT EXTREME POINTS OF OPERATION.

AN AVAILABLE MODEL TURBINE CHARACTERISTIC DIAGRAM (HILL CHART DIAGRAM) IN FIG 3 FOR A BULB TURBINE WITH SPECIFIC SPEED $\mathbf{n_s} = nP^{0.5}/H^{1.25}$ **= 323** HAS BEEN USED FOR ILLUSTRATION OF THE INFLUENCE ON THE EFFICIENCY WHEN OPERATING AT EXTREME LOW HEAD AND HIGH HEAD.

THIS AVAILABLE EFFICIENCY CHARACTERISTIC DIAGRAM HAS BEEN USED FOR ILLUSTRATION OF THE PROBLEM ONLY, BECAUSE FOR SANTO ANTÔNIO WE HAVE $\mathbf{n_s}$=815.

THE LOWER SPECIFIC SPEED USED, WILL SHIFT THE EFFICIENCY CURVES IN THE CHARACTERIATIC DIAGRAM (HILL CHART DIAGRAM) TO THE LEFT.
THEN THE EFFICIENCY WILL DROP AT LOW HEAD WITH INCREASED SLOPE OF THE CURVES COMPARED WITH THE DIAGRAM FOR THE HIGH SPECIFIC SPEED TURBINES FOR SANTO ANTÕNIO. (SEE FIG.4)

HOWEVER, TOWARDS HIGH HEAD I.E. IN LEFT SIDE OF THE DIAGRAM, THE EFFICIENCY WILL BE INCREASED WITH A DECREASED SLOPE OF THE CURVES FOR THE LOW SPECIFIC SPEED USED FOR ILLUSTRATION. (SEE FIG. 4).

IT IS IMPORTANT TO RESEIVE CHARACTERISTIC DIAGRAMS (HILL CHARCH DIAGRAMS) AND EFFICIENCY CURVES VITH CRITICAL σ VALUES FROM POTENTIAL BIDDERS IN ORDER TO STUDY THE LOSS IN PRODUCTION CAUSED BY POOR EFFICIENCY AND POSSIBLE LIMITATION IN OPERATION DUE TO CAVITATION AT THE EXTREME TAILRACE LEVELS.

THIS EVALUATION GIVES A STRONG IMPACT TO THE ANNUAL POWER PRODUCTION AND THE CONOMY OF THE PROJECT.

IT SHOULD ALSO BE NOTED THAT MODERN BULB TURBINES HAVE IN GENERAL IMPROVED PERFORMANCE COMPARED TO 30 THE YEAR OLD TURBINE USED IN MY EVALUATION FOR EXPLAINATION OF THE ANALYSIS THAT HAS TO BE DONE.

OPERATING POINT FOR STABILITY ANALYSIS.

Study of the Characteristic Diagram (Hill Chart Diagram).

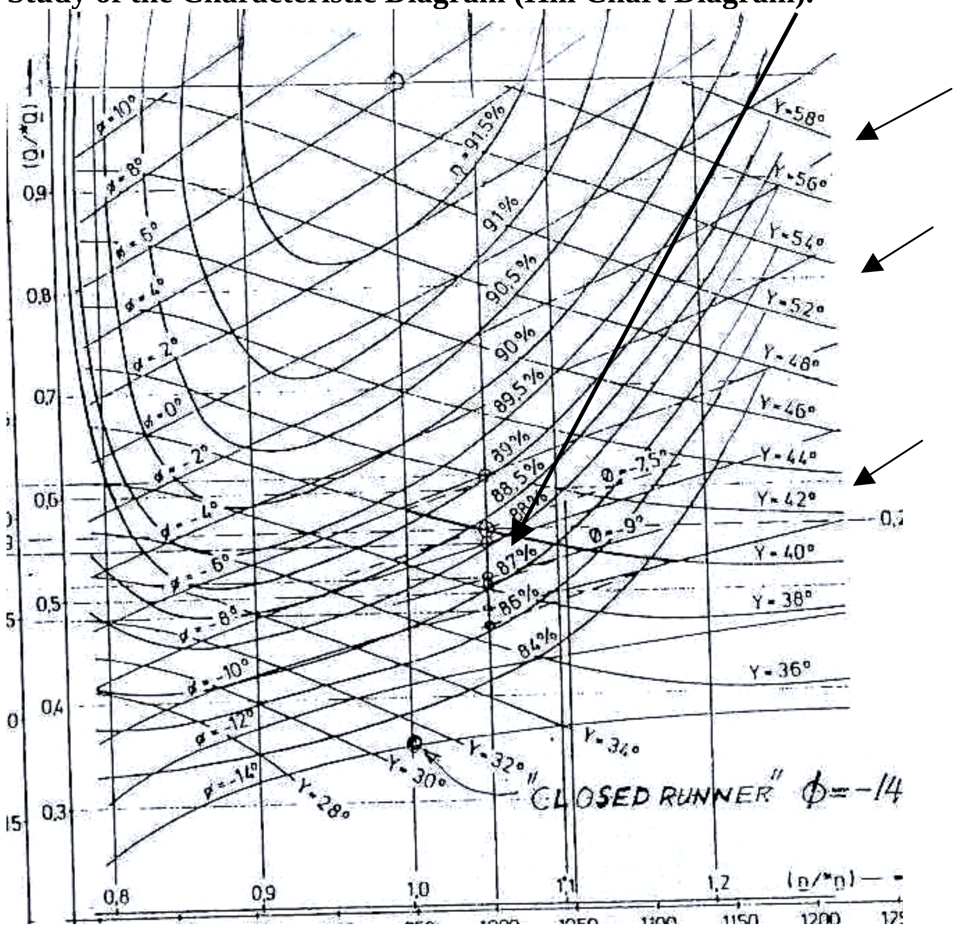

Fig. 3 Characteristic diagram (Hill Chart diagram) for a Bulb turbine with specific speed lower than for the projected turbines for Santo Antônio used for demonstration. A Characteristic Diagram for the chosen specific speed for Santo Antõnio, is needed from the turbine manufacturers. (The presented diagram was used for governing stability analysis in the Ph.D. Thesis of Prof. H. Brekke.)

In the diagram shown in fig.3, the turbine efficiency is valid for operation "on cam" (i.e. the runner blades are adjusted to the guide vane position for optimal efficiency also as function of the head) at constant speed of the turbine.

Definition of data used in the Characteristic diagram:

Along the horizontal axis in the diagram in fig.3 we find the relative speed **(<u>n</u>/*<u>n</u>)** where <u>n</u>=n/(2gH)$^{0.5}$ and n is an arbitrary speed and ***n** is the best efficiency speed and ***H** is the best efficiency head i.e.***<u>n</u>**=*n/(2g*H)$^{0.5}$.
Thus **(<u>n</u>/*<u>n</u>)**=(n/*n)(*H/H)$^{0.5}$ and for constant speed where **n=*n**, the value along the horizontal axis is **(*H/H)$^{0.5}$** i.e. at rated head **(*H/H)$^{0.5}$=1** and for decreasing head the value of (*H/H)$^{0.5}$ will increase proportional to the square root of the head ratio.
In a similar way the value along the vertical axis **(<u>Q</u>/*<u>Q</u>) = (Q/*Q)(*H/H)$^{0.5}$** i.e. the influence of the head on the flow is included.

From the diagram in fig.3 and as illustrated in fig. 4, the efficiency for head variations at given guide vane openings, can be found. It is important to notice that at low head and large openings of the guide vanes a sharp drop in efficiency occurs for heads below H=9 m for n_s=323. Further the efficiency shows a sharp drop towards increased head and a sharp drop occurs for H>27 m.

For the specific speed used for Santo Antônio, the curves moves to the right.

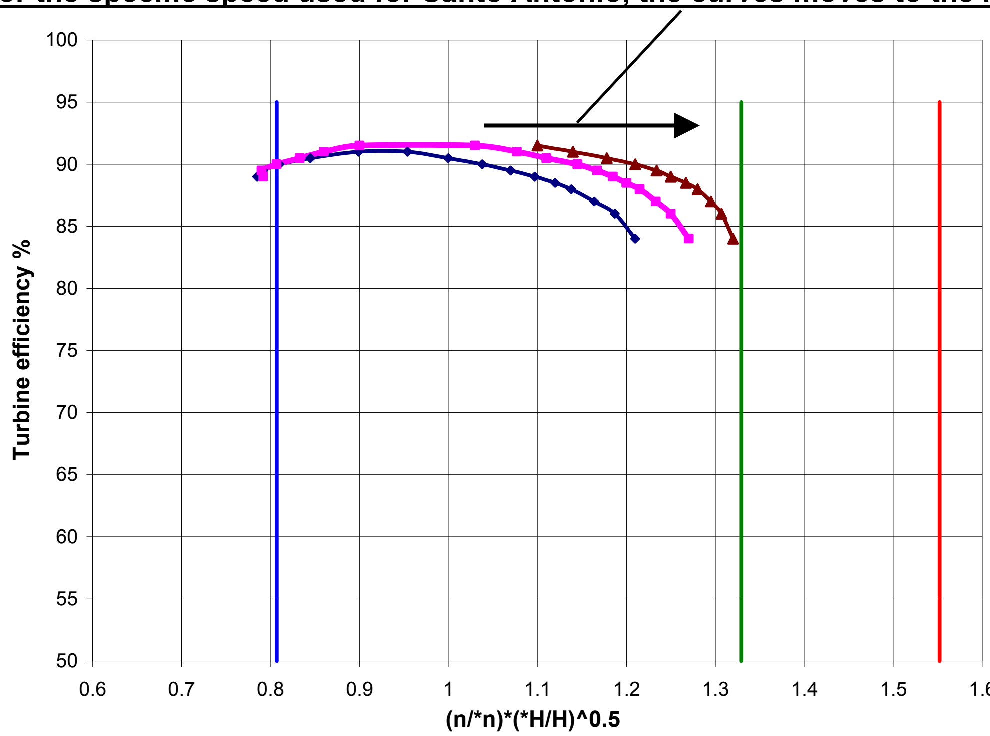

Fig. 4 Efficiency for 3 constant guide vane angles i.e. 58^O brown, 52^O violet and 44^O blue, referred to the Characteristic Diagram (Hill Chart) shown in fig. 3. The vertical lines are valid for the net head H=6.6 m red, H=9.0 m green and H=24.4 m blue.(Note: As Described the efficiency curves in this figure represent a much lower specific speed than for Santo Antônio and a higher specific speed will shift the curves toward right in the diagram.)

Because the power production and economy of the project will be influenced by the turbine efficiency and possible restrictions in operation, it is important to receive information from potential turbine manufacturers presented by the turbine Characteristic Diagram.
Then an analysis as illustrated in fig. 4 and fig.2 will show possible restrictions in operation caused by cavitation and loss in production because of drop in efficiency which has a strong impact on the economy in the Santo Antônio project.

Finally it may be considered to install a minor number of turbines of different specific speed that could operate with high efficiency and safety against cavitation at lowest tail race level when the need of power is high. Such analysis could be made on basis of economy by increasing the production in critical periods with additionally installations. It may also be discussed to install a few cheap high specific speed propeller turbines for operation during flood when the normal turbines are operating at low efficiency at 6.-7. m net head.
However, for a final evaluation, data with Hill Chart diagrams from possible manufacturers is needed.

3. HYDRAULIC FORCES ON TURBINES AND POWER HOUSE STRUCTURE DURING EXTREME LOW TAIL RACE LEVELS INCLUDED A BRIEF ANALYSIS OF THE NEGATIVE THRUST BEARING FORCES ON GENERATOR DURING RUNAWAY.

Forces on powerhouse.

At lowest tailrace level there will be a pressure of 36.65 m on the turbine structures upstream of the guide vanes and at only 10.75 m on the tail race side.

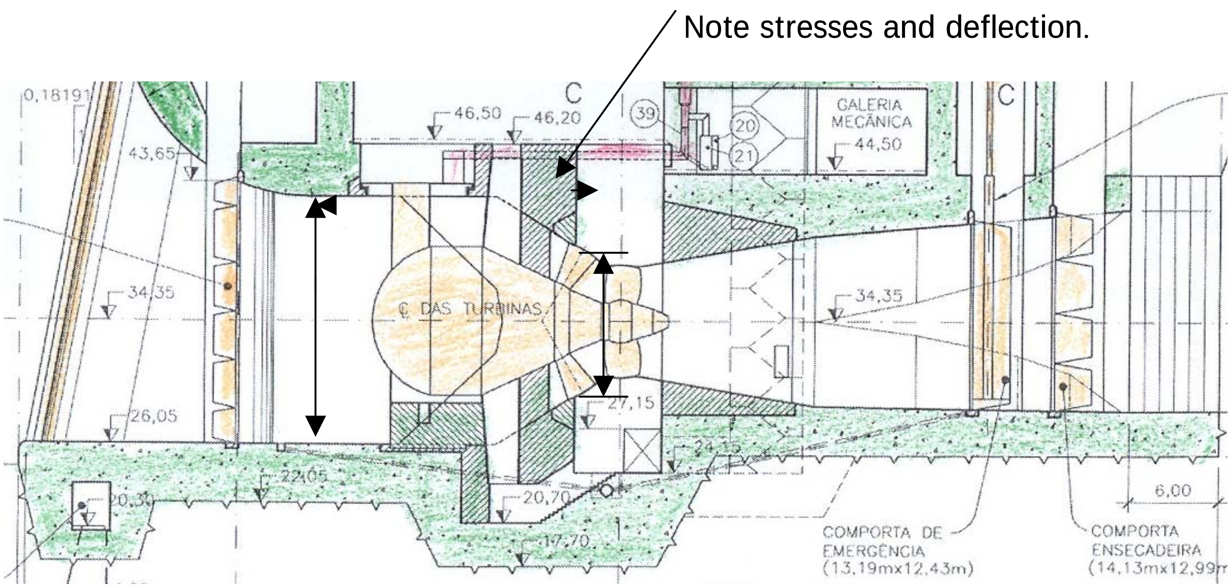

Fig. 5. Illustration of Bulb turbine for Santo Antônio with marked diameters of Bulb turbine lining anchored to concrete and guide vane sealing diameter exposed to forces from upstream pressure and downstream pressure of 36.65 m and 10.75 m respectively.

The dimensions of the turbines for Santo Antônio are taken from the project drawing shown in fig.5. The diameter of the lining on the concrete surrounding the bulb turbine is approximately **18** m and the diameter of the outer sealing of the guide vanes is approximately **10.5** m.
Based on the upstream pressure of **35.65** m and lowest downstream pressure of **10.75** m referred to the centerline of the turbine, following forces will be transferred to the concrete surrounding the opening for dismantling of the runner.

Special attention should be taken to study the deflection of the concrete, because permanent deformation of the turbine connections between runner chamber and draft tube, has been reported from some bulb turbines and the maximum head difference **24.90** m is extremely large for bulb turbines. In following page a brief calculation of the forces is presented.

Note: 1 kN ≈0.1 tonn:
Forces in downstream direction:

Upstream side pressure: $p_u = \rho gH*10^{-3}$ kN/m^2= gH kN/m^2=9.81*36.65 kN/m^2 ≈36.65 Tons/m^2.

Downstream side pressure: $p_d=\rho gH*10$-3 kN/m^2=gH kN/m^2=9.81*10.75kN/m^2 ≈10.75 Tons/m^2.

Total force on the concrete surrounding the hatch opening for runner dismantling and surrounding concrete: (See remark in fig. 5.)

$$9.81*(\pi*(18^2/4)*35.65 - \pi*(10.4^2/4)*10.75) = 80036.\ \text{kN} \Rightarrow 8158.6\ \text{Tons}$$

To avoid problems with the mechanical equipment the deformation should be within the demand from the turbine manufacturer i.e. within a few mm.

However, the critical area is the structure between the generator hatch opening towards and the hatch opening for access to dismantle the runner chamber for maintenance of the runner. Approximately 40% of the forces i.e. forces above center of the turbine, will be transferred to the concrete via the steel lining, loading the cross section shown in fig. 5 marked with notation "Note stresses and deflection."
Before the final dimensioning of the power house a thoroughly Finite Element Analysis (FEM) must be made in collaboration with the turbine manufacturer to get the deflections and stresses as accurate as possible. This is for safety and to avoid possible problems with permanent distortions of the concrete in the future.

Axial thrust from runner.

The pressure difference from upstream side to downstream side of the runner in a Bulb Turbine is more than 60% of the net head. It is assumed that the diameter of the runner is 8.00 m and the hub 2.5 m according to drawings in the report and my experience indicating the ratio d/D≈0.3 for the given specific speed.

Then the force at highest operational net head 24.4 m will be as shown in following equation:

$$F=0.6gH\pi(D_r^2-d_r^2)/4=0.6*9.81*24.4*PI()*(8\hat{}2-2.5\hat{}2)/4=6514\ \text{kN}\Rightarrow 664.0\ \text{Tons}$$

It should be mentioned that during runaway of the turbine the axial thrust shifts to a relative high negative value depending on the speed and blade angle and angle of the guide vanes.
It is impossible to indicate this value without data from the turbine manufacturer that has the model turbine in position.
Request of these values from the turbine manufacturers should be made with question about the closing speed of the runner versus guide vanes.

IN GENERAL FOLLOWING RULES SHOULD BE RECOMMENDED FOR SETTING OF THE RUNNER BLADES SPEED**:**

THE RUNNER SHOULD HAVE A FAST OPENING SPEED SO THE RUNNER IS OPENING WITH THE SAME SPEED AS THE GUIDE VANES.
THIS IS IN ORDER TO AVOID TRIPPING WITH THE COMBINATION OF LARGE GUIDE VANE OPENINGS AND SMALL RUNNER BLADE OPENINGS, WHICH GIVES A VERY HIGH RUNAWAY SPEED.

THE CLOSING SPEED OF THE RUNNER BLADES SHOULD BE SLOWEST POSSIBLE IN ORDER TO REDUCE THE RUNAWAY SPEED WHICH IS INCREASING WITH DECREASED OPENINGS OF THE BLADES.

THE NEGATIVE THRUST FORCES AT RUNAWAY, INCREASES ALSO WITH DECREASING BLADE OPENING OF THE RUNNER.

4. SAND EROSION AND HARD SURFACE COATING.

Sand erosion

The sand content in the water will be affected by the centrifugal force in the rotating flow as illustrated in fig. 7 and the blade tips will be the part of the turbines that is normally most severely eroded together with the runner chamber. Also the guide vanes may be eroded and end leakage flow may cause damages.
However, the end leakage flow and the high speed flow at the blade tip ends are the locations for attention of sand erosion and on frequent repair. I addition sand erosion has been observed on the inlet edge of the blades and at the surface on pressure side towards outlet, where we find the highest velocities combined with the highest angles of impact of the sand.
It should be noted that also the guide vans will be eroded.
The reason for the severe erosion on location exposed to high velocity is that sand erosion is depending on the angle of attack and is increasing proportional to the velocity in the power 3.

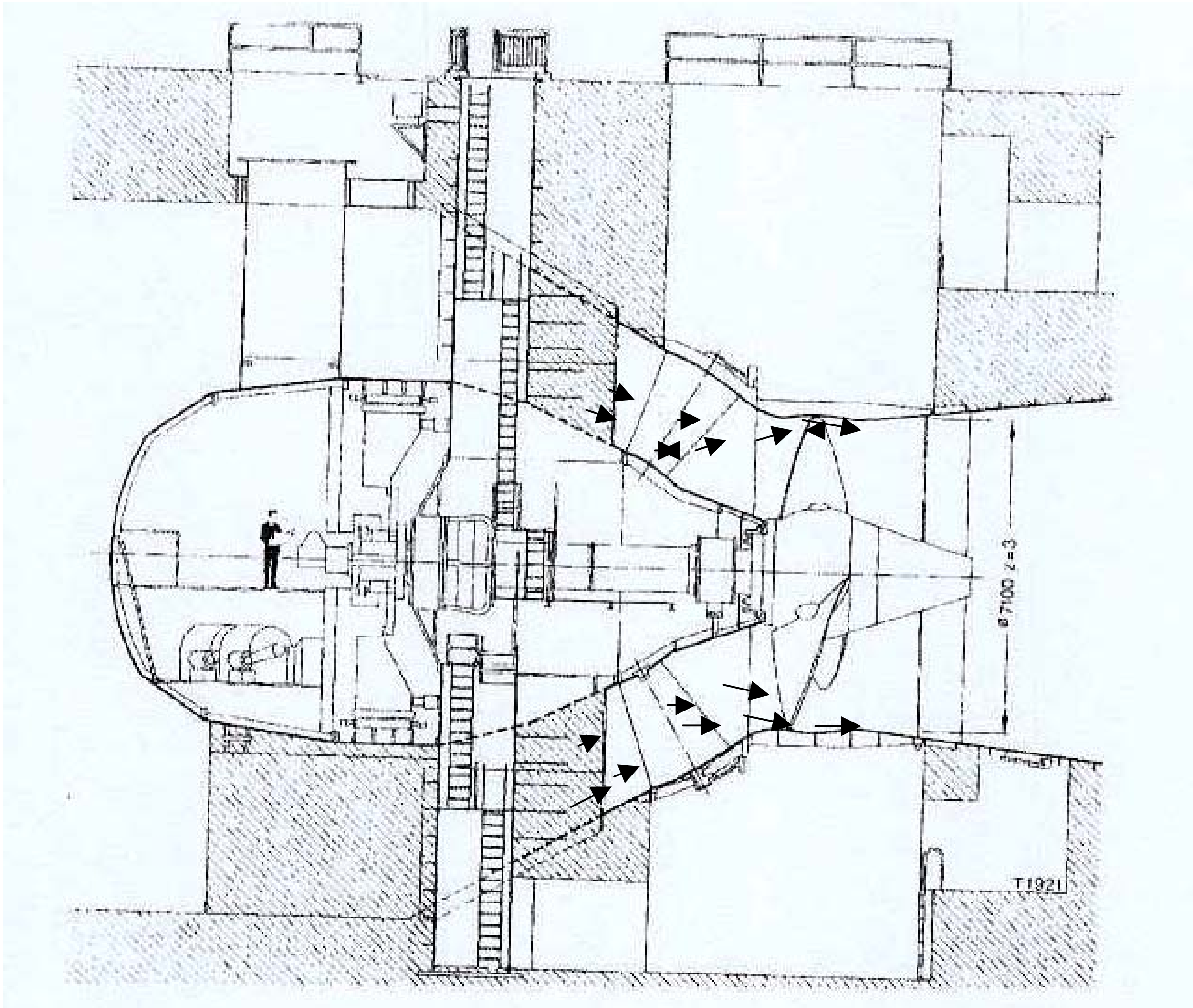

Fig. 6 Illustration of sand movement in a Bulb Turbine

Erosion resistive coating.

The time between repairs of sand eroded surfaces is depending on the flow velocity and flow direction as described above and the amount of sand, combined with the hardness and grain sizes weighted against the hardness/brittleness and ductility of the material in the turbine parts exposed to sand erosion.

At the Norwegian University of Science and Technology (NTNU) research work on hard surface coating was made in 1991-1994 for the hydropower industry and owners as well as the oil companies working off shore outside the Norwegian coast.

In the research work also the influence of corrosion was included mainly because the off shore components exposed to sand erosion for the oil companies, were exposed to sea water.

This influence can be ignored for water turbines with stainless steel blades of 13% Cr, 4% Ni or 16% Cr, 5% Ni blades with Mo and other anti corrosive alloy components.
The result of the research work concluded in that Tungsten Carbide of the quality marked by an arrow in fig. 7 was the best choice.
The time between repairs will be increased by a factor of approximately 5 compared to normal stainless steel qualities.
For Kaplan turbines and Bulb turbines the access for spraying is not limited as for a Francis turbine runner with narrow blade spacing and a sufficient quality and thickness of the coating should be obtained on all critical surfaces.
However, distortion of the blades is a problem that requires an experienced turbine manufacturer to spray the blades. Some manufacturers have access to similar patented qualities of coating as shown in fig.7.
The very thin layers of Diffusion or Ion Nitrated surfaces have proven that the thickness of the hard surface layer was too thin to give protection.

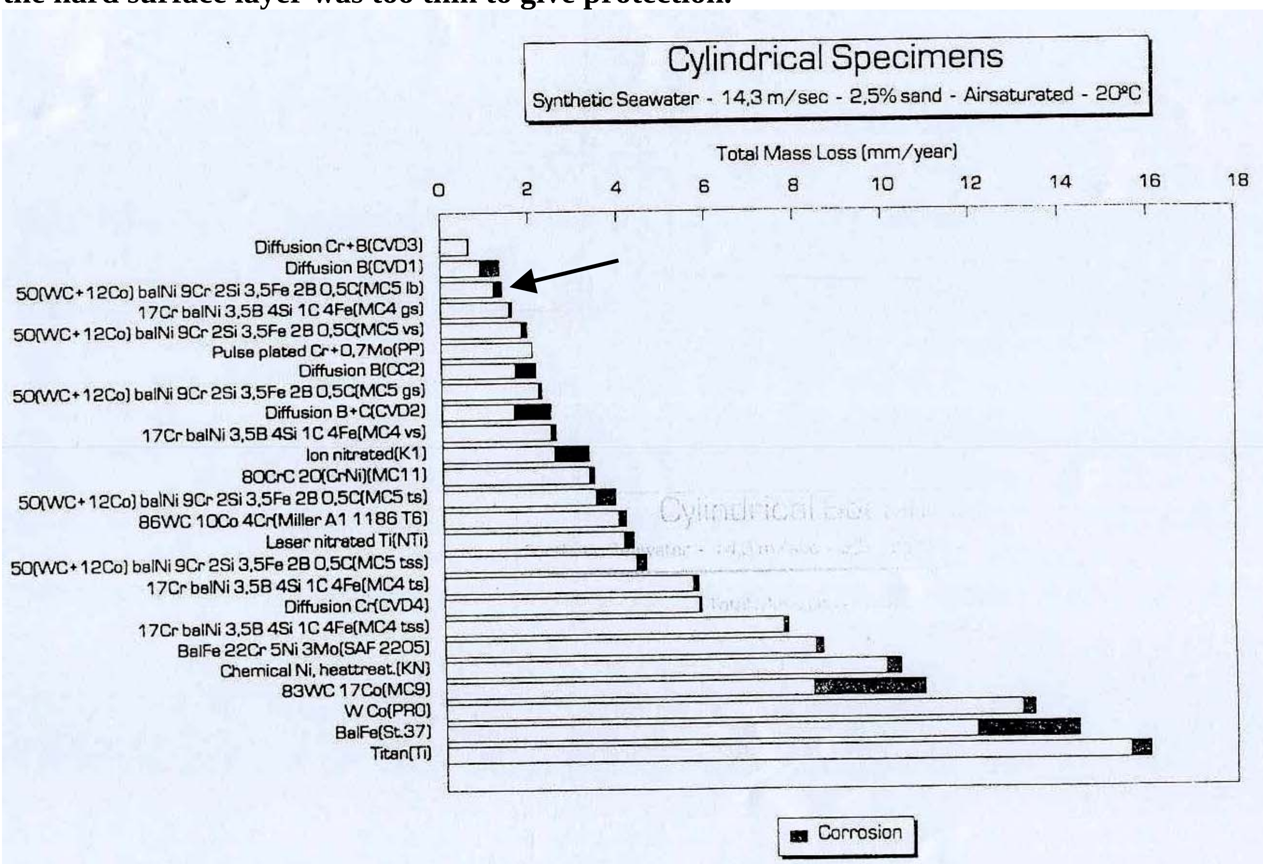

Fig. 7 Hard surface coating tested at the Norwegian University of Science and Technology. (NTNU) Ref: Paper: H.Brekke: "Norwegian Research Work on Erosion Resistive Coating for Water Turbines". XVII IAHR SYMPOSIUM Beijing, China 1994.

In fig 8 is illustrated the proposed areas for coating of runner blades for Kaplan and Bulb turbines.

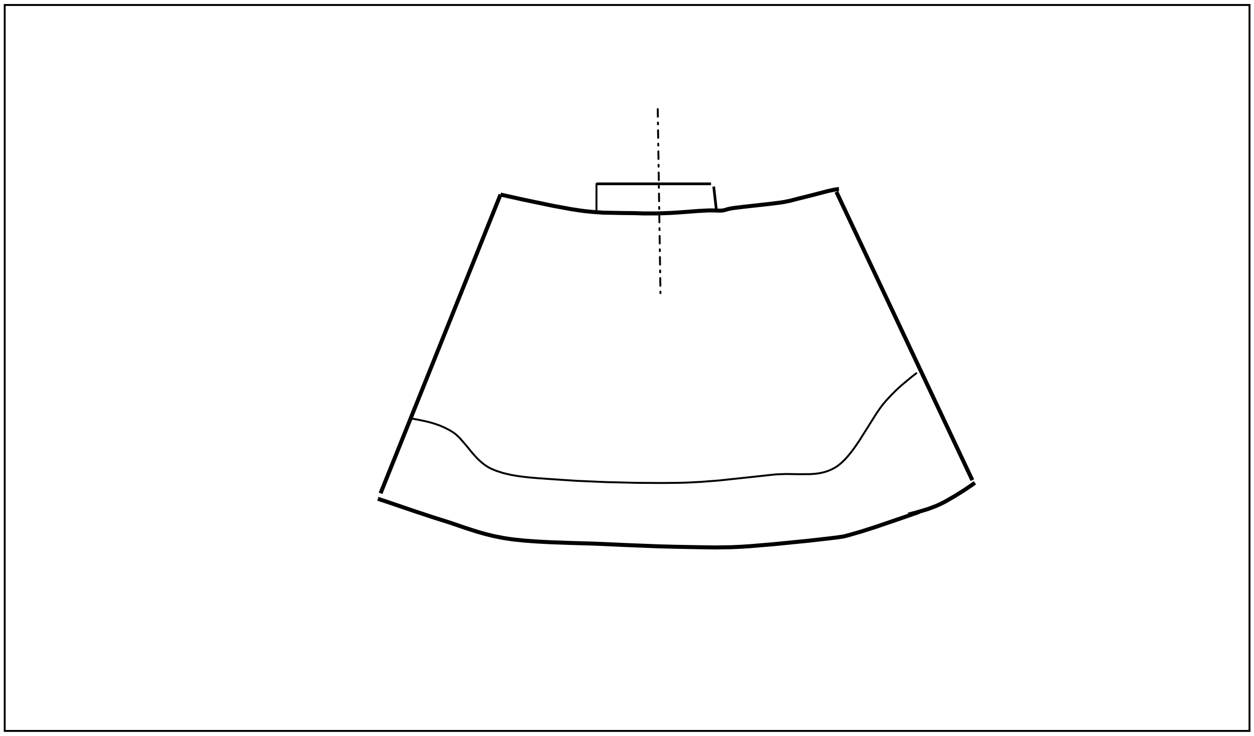

Fig. 8 A Proposed section for coating of a Bulb- or a Kaplan turbine blade.
Ref: Book: "Abrasive Erosion Corrosion of Hydraulic Machinery": Imperial College Press, ISBN 1-86094-335-7.

Conclusion:

Because of the cost of hard surface coating it should be recommended to start the first units without coating in order to find out if it is a right economical solution to implement hard surface coating.
In any case a work shop with a milling machine with sufficient diameter for machining of the runner chamber and blades held in position at the correct diameter of approximately of 8 m during machining, should in any case be installed at the power plant.
If coating and re-machining of the blade profiles shall be made a NC controlled machine will be necessary because distortion of the blades is expected during the spraying process and reshaping is required.

5. POSSIBLE GOVERNING STABILITY PROBLEMS DURING OPERATION DEPENDING ON CONNECTED GRID.

The turbine.
Kaplan turbines, Bulb turbines and Propeller turbines have all an unstable characteristic, i.e**. the power increases with increasing speed** at all operating points in the Characteristic diagram.

In the following short description, complex plane analysis with Laplace transformed equations will be used, but a physical description of the problem will be given.
However, the stability analysis of the turbines must be included in the total analysis of the voltage governor and the electric grid because of the size of Santo Antônio power plant.
Further it will in principle be impossible to run the power plant on isolated resistive grid with Bulb turbines with unstable characteristic at full load. (Increasing power for increasing speed and with small inertia mass of the generator.)
Part load operation of the turbines on isolated resistive load, may be possible if the voltage governors are furnished with voltage/frequency droop (limited to $\pm$1.5 Hz) and the turbine governors have a strong derivative influence with fast operating control valves.
This will be explained in the following:
In order to handle the unstable turbine characteristic, the turbine governor must be of the PID type as defined in following equation: (Note complex plane stability analysis is used and S=jω.)

$$\mathbf{PID = ((1+T_N S)/(b_t(1+0.1T_N S))*((1+T_D S)/T_D S)*1/(1+T_Y S)}$$

A stability analysis confirmed by a frequency response test at site, was made for the Bulb turbine at **Kongsvinger Power** plant in Norway operating at around 40^{O} opening of the guide vanes i.e. about 50% load.
The runner blade opening angle was 6.5^{O} and the turbine Characteristic diagram is shown in **fig. 3**.

The time constants and amplifying constants in the equation above that was set for Kongsvinger yields: TD=15 sec, TN=0.7 sec, TY=0.2 sec, bt=2.6.
Further the generator inertia time constant Ta=2.3 sec. referred to P=9.2 MW where full load was 19.1 MW.
The other data of turbine constants taken from the characteristic diagram were:
The influence on the flow from the speed: Q_n=0.55
The influence on the flow from the guide vane movement: Q_y=0.46
The influence on the flow from the runner blade movement: Q_Φ=0.37
The influence on the efficiency from runner blade movement: E_Φ=0.044
The influence on the efficiency from the flow: E_q=0.113
The influence on the efficiency from speed: E_n= -0.18
Further the influence of the runner blades on the flow compared with the guide vanes: K_L=4.7
Finally the time constant for the runner blade movement yields: T_L=1.5 sec
All constants are evaluated from the turbine characteristic diagram and the mechanical leverage with time constants of the PID governor and guide vane servomotors and runner blade servomotor together with the generator inertia mass. It will be too complicated to explain the total theory in this report, but the block diagram of the system is shown in fig.9. Ref.: (PhD) thesis of Hermod Brekke.

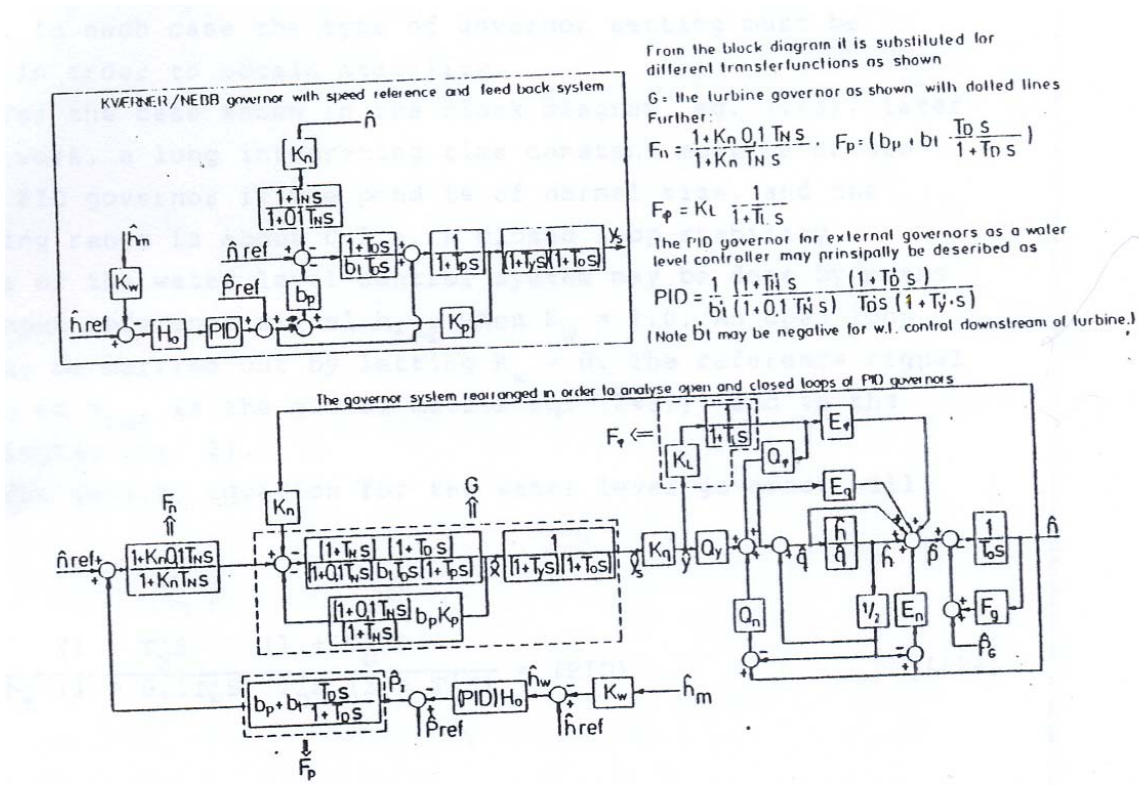

Fig 9 Block diagram for the system of Bulb turbine with turbine governor connected to an isolated resistive electric grid.

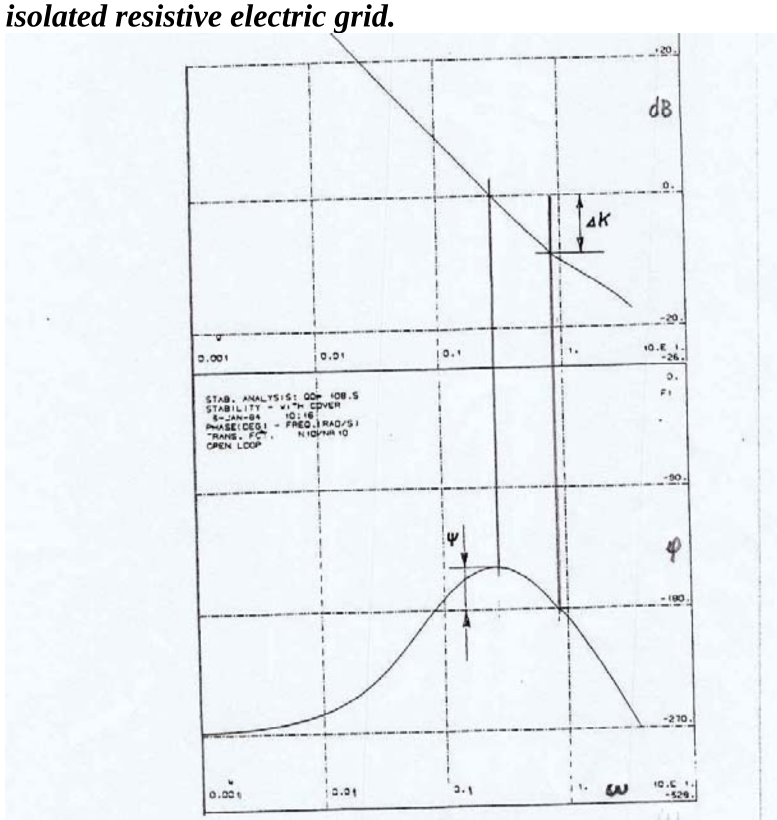

Fig 10. Open loop analysis of the bulb turbine at Kongsvinger on isolated resistive load.

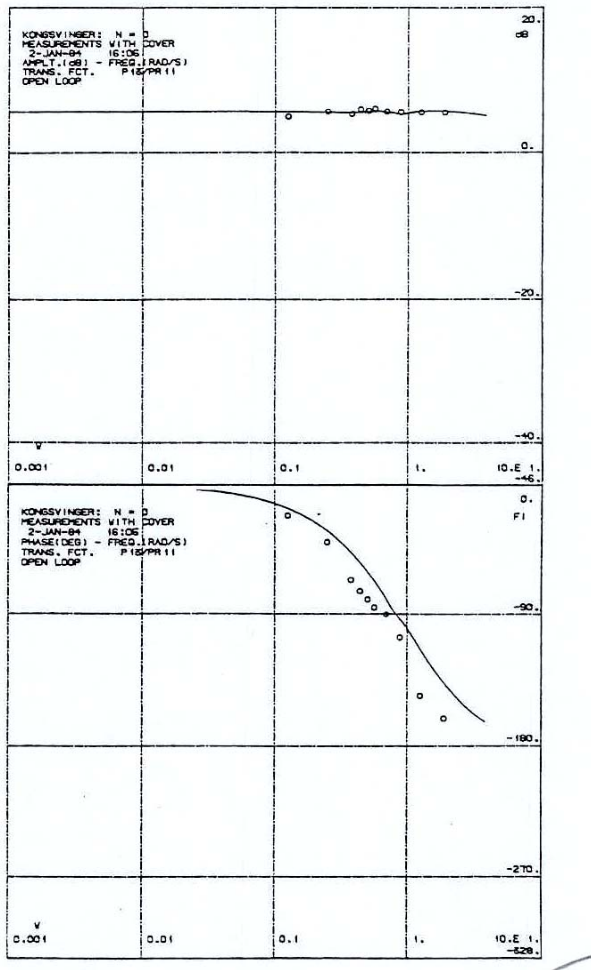

Fig 11 Closed loop analysis of the Bulb Turbine Kongsvinger compared with frequency response test.(Measurement shown by dots)

The result of the analysis in open loop is shown in fig. 10. In fig. 11 the analysis in closed loop is shown. The theoretical analysis was verified by frequency response measurements at site which is shown by the measuring points plotted. There is an offset of about 10^{O} of the phase angle at high frequencies which was explained by off set of the measuring signal and the small time constant Ty=0.2 used in the analysis because the real value was probably higher. However, the gain proved the validity of the theory.

6. MATERIAL QUALITY AND STANDARDS, BRAZILIAN PRODUCTION.

Material quality.

For material specification the European EN standard, the German DIN standard, the American ASTM or the British GB may be used by the bidders. (Also other standards may be used if applicable.) These standards are well known and must be accepted.

EXAMPLE:

STAINLESS STEEL CASTING. 13%Cr, 4% Ni, NORMALLY USED FOR BLADES.

EN 10283	DIN 17445	ASTM	GB
GX4CrNi 13-4+QT1	G-X 5 CrNi 13 4 V1	A 743 Grade CA-6NM	GB 6967 Grade ZG06Cr13Ni4Mo

Requirements for the foundry.

The foundry must fulfill the requirements in CCH 70-3 for the high stressed parts of the runner blades and runner hub, if the hub is cast**.**
The foundry must also prove that a control of the content of N_2, O_2 and H_2 is made before casting. This is because of danger of material defects and brittleness in the steel structure are triggered by these elements together with the content of S is the root for creating weakness flaws and cracks caused by MnS in the structure.
Further the experience and skill in repair welding and also experience in spraying hard surface coating should be required from the successful bidder.

Standards.

Guidelines for technical specifications for tubular turbines IEC 61366-5 is recommended.
The foundry must fulfill the requirements in CCH 70-3 for both casting and repair. Specially the requirement for Magnetic Particle test MT 70-3 and Ultra Sonic test UT 70-3 must be fulfilled referring to the strict requirements. Proposals from the bidders are expected.

Forgings.

The runner shaft must also fulfil the requirement of quality forgings with special attention to the fillet radius against the flange. It is not expected that the shafts of the required sizes can be forged in Brazil. Machining close to finished value is required for inspection before shipping if the final machining shall be made in Brazil. The shaft core must be bored for inspection to a dimension sufficient for the oil piping for runner blade control.

Brazilian production.

To the author’s knowledge Brazilian foundries have proven the ability to produce large runners for Francis turbines cast and NC machined in Brazil. Then the runner blades and

runner hub could be produced in Brazil. Ref Voith Siemens work shop and foundry in Sao Paulo.
Further the runner chamber, guide vanes could be produced in Brazil for the same reason.
The only question is if the shaft could be forged in Brazil.
Further the journal bearing and shaft seal and the thrust bearing (normally part of the generator) require special competence, but may also be made in Brazil. (I do not have information about this.)
The heavy parts of the governors such as servomotors, oil pressure system control valves are normally standardized products for high pressure systems that probably will be used.
Some of these parts may be imported.

The computer based speed control and control system for the plant will be part of the electromechanical system, but the location for production will depend of the experience of the successful bidder. (May be produced in Brazil?)
The generator is not part of my evaluation so further evaluation of the electro mechanical system will not be given.

The welded structure of steel linings must be made by local companies supervised both by the structural engineers and the turbine manufacturer.

CONCLUSION

It is important to receive an updated Characteristic Diagram (Hill Chart Diagram) from potential bidders as soon as possible for review.
This is because the efficiency will drop both towards maximum and minimum head i.e. at minimum and maximum flow in the river.

Cavitation may occur especially at low tail race level where the production is small but valuable. A possible restriction of production at low flow must be evaluated.
A possibility may be to install a small number of the units designed for operation at high head and low tailrace level i.e. reduced specific speed.
For operation at maximum flow a few fixed blade cheap turbines designed for large capacity and low net head may be installed.

THE ECONOMY OF THE PROJECT IS DEPENDING ON THE POWER PRODUCTION AND IT IS IMPORTANT TO GET THE CHARACTERISTIC DIAGRAM (HILL CHART DIAGRAM) FROM THE POTENTIAL BIDDERS FOR VERIFICATION OF THE PERFORMANCE OF THE OFFERED TURBINES IN ORDER TO CALCULATE THE ANNUAL POWER PRODUCTION CORRECTLY.

A preliminary study of the forces from the turbines on the power house structure is given in this report, because it is important to avoid distortion of the runner chamber and connection between the guide vane apparatus and the joint to the draft tube. This is to avoid damage of this structure and the maximum net head for Santo Antõnio is on the limit for Bulb Turbines.

Sand erosion problems have also been discussed with advises and proposals of hard surface coating.

Further studies can be made in my office in Norway as soon as the necessary Characteristic Diagrams (Hill Chart Diagrams) are available.
I am aware of the confidentiality of this matter, but it is vital to analyse the possibility of operation because of the extremely wide range of head and flow in Santo Antõnio Project.

Report handed over BRASILIA 30.11.2006

Pelo Consultor

Assinado por HERMOD BREKKE