RAPEL POWER PLANT

DESIGN; MANUFATURE AND ERECTION OF PIECES FOR THE REHABILITATION OF THE TURBINE N°5

GUARANTEED CHARACTERISTICS

2.1.2 The runner, bushings of wicket gates and actuator system of distributor, stationary and runner seals, air injection system (if applies) and associated pieces will be manufactured in (country, city and address):

B VALUES OF DESIGN AND CHARACTERISTICS OF THE TURBINE AND ASSOCIATED EQUIPMENT

1. GUARANTEED PERFORMANCE

Nominal Net Head H1 =75.5 m,
Nominal Guaranteed Power =__**73.68 mw**_ (> 73.55 MW), that corresponds to $Q_{nominal}$= 107 m^3/s,with 100% opening

Percentage of guaranteed Nominal Power	Power	Guaranteed efficiency	Weight coefficient
%	**MW**	**%**	**k**
95	70.00	94.0	0.1
90	66.31	94.0	0.3
80	58.94	92.2	0.25
65	47.89	88.6	0.2
50	36.84	83.6	0.15

The Nominal 100% opening will be the opening with flow of Q = 107m^3/s and head H1 = 75.5m.

2. POWER; FLOW AND EFFICIENCY

2.1 Efficiency curves, power and flow as function of opening of wicket gates , with indication of upper and lower limits of permanent operation because of cavitation for each of the following net heads Ho =78.7 m; H1 =75.5 m y H2 = 72.3 m

GRAPHS N^{ros} : ______________________**None** _______________________________________
Included in the attachment

2.2 Efficiency curves and flow as function of power with indication of upper and lower limit of permanent operation because of cavitation for each fo net head H0, H1 y H2.

NAME AND SIGNATURA OF MANUFACTURER

Date : . . . .

NAME AND SIGNATURA OF DESIGNER

Date: . . . .

GUARANTEED CHARACTERISTICS

GRAPHS N[ros] : ____________________ **Proposal Section 6_** ____________________
Included in the attachment.

2.3 Hill diagram

2.3.1 Turbine: Net head vs flow with parametrics curves of:

2.3.1.1 Efficiency.
2.3.1.2 Gates openingof distributor. (from "idle" to "max opening").
2.3.1.3 power. (from "0" to "max").
2.3.1.4 Incipient cavitation coefficient.
GRAPHS Nº : ___Proposal Section 6__________________________
(Include as reference the limits because of cavitation for net head and flow).

NOTE: Normal range of operation should be stated. Zones beyond the range will be just for reference.

2.4 Weight guaranteed performance as stated in 7.2 of terms of reference.

2.4.1 $(\xi\ pg)$: ________**90.9**____________%

2.5 Tables (Charts) of efficiency, power and flow, as function of the position of the wicket gates of distributor.

GUARANTEED CHARACTERISTICS

2.5.1 For Ho = 78.7 m

NET HEAD (H) (m)	~~Ao/Aon~~ Flow (%)	POWER (P) (MW)	FLOW (Q) (m^3/s)	EFFICIENCY (ɖ) (%)	LIMITS OF OPERATION(MW)
Ho = 78.7	__(Máx)				
	100	76.89	107	93.2	XXXXXXXXXXXXX
	95	73.70	101.7	94.0	XXXXXXXXXXXXX
	90	69.72	96.3	93.9	XXXXXXXXXXXXX
	85	65.32	91.0	93.1	XXXXXXXXXXXXX
	80	60.52	85.6	91.7	XXXXXXXXXXXXX
	75	56.09	80.3	90.6	XXXXXXXXXXXXX
	70	51.34	74.9	88.9	XXXXXXXXXXXXX
	60	42.32	64.2	85.5	XXXXXXXXXXXXX
	50				XXXXXXXXXXXXX
	40				XXXXXXXXXXXXX
	30				XXXXXXXXXXXXX
	20				XXXXXXXXXXXXX
	__(en vacío)				XXXXXXXXXXXXX

NOTES: Aon =Nominal opening of 100%. Corresponds to opening with flow Q = $107 m^3/s$ and net head H1 = 75.5 m.

NAME AND SIGNATURA OF MANUFACTURER

NAME AND SIGNATURA OF DESIGNER

Date : . . . .

Date: . . . .

GUARANTEED CHARACTERISTICS

2.5.2 For H1 = 75.5 m

NET HEAD (H) (m)	~~Ao/Aon~~ Flow (%)	POWER (P) (MW)	FLOW (Q) (m^3/s)	EFFICIENCY (ɖ) (%)	LIMITS OF OPERATION(MW)
H1 = 75.5	__(Máx)				
	100	73.68	107	93.1	XXXXXXXXXXXXX
	95	70.63	101.7	93.9	XXXXXXXXXXXXX
	90	67.02	96.3	94.1	XXXXXXXXXXXXX
	85	62.73	91.0	93.2	XXXXXXXXXXXXX
	80	58.25	85.6	92.0	XXXXXXXXXXXXX
	75	60.64	80.3	90.8	XXXXXXXXXXXXX
	70	49.42	74.9	89.2	XXXXXXXXXXXXX
	60	40.60	64.2	85.5	XXXXXXXXXXXXX
	50				XXXXXXXXXXXXX
	40				XXXXXXXXXXXXX
	30				XXXXXXXXXXXXX
	20				XXXXXXXXXXXXX
	___(en vacio)				XXXXXXXXXXXXX

NOTE: Aon =Nominal opening of 100%. Corresponds to opening with flow Q = 107m^3/s and net head H1 = 75.5 m.

NAME AND SIGNATURA OF MANUFACTURER NAME AND SIGNATURA OF DESIGNER

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GUARANTEED CHARACTERISTICS

2.5.3 For H2 = 72.3 m

NET HEAD (H) (m)	~~Ao/Aon~~ Flow (%)	POWER (P) (MW)	FLOW (Q) (m^3/s)	EFFICIENCY (η) (%)	LIMITS OF OPERATION(MW)
H2 = 72.3	_____(Máx)				
	100	70.25	107	92.7	XXXXXXXXXXXXX
	95	67.42	101.7	93.6	XXXXXXXXXXXXX
	90	64.12	96.3	94.0	XXXXXXXXXXXXX
	85	60.14	91.0	93.3	XXXXXXXXXXXXX
	80	55.78	85.6	92.0	XXXXXXXXXXXXX
	75	51.76	80.3	91.0	XXXXXXXXXXXXX
	70	47.37	74.9	89.3	XXXXXXXXXXXXX
	60	38.92	64.2	85.6	XXXXXXXXXXXXX
	50				XXXXXXXXXXXXX
	40				XXXXXXXXXXXXX
	30				XXXXXXXXXXXXX
	20				XXXXXXXXXXXXX
	___(en vacio)				XXXXXXXXXXXXX

NOTAS: Aon =Nominal opening of 100%. Corresponds to opening with flow Q = 107m^3/s and net head H1 = 75.5 m.

2.6 Tables (Charts) of efficiency and flow as function of power.

NAME AND SIGNATURA OF MANUFACTURER NAME AND SIGNATURA OF DESIGNER

Date : . . . . Date: . . . .

RAPEL POWER PLANT

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GUARANTEED CHARACTERISTICS

2.6.1 For Ho = 78.7 m

NET HEAD (H) (m)	POWER (P) (%)	POWER (P) (MW)	FLOW (Q) (m^3/s)	EFFICIENCY (ɖ) (%)	LIMITS OF OPERATION(MW)
Ho = 78.7	____(Máx)				
	100 (1)	76.89	107	93.2	XXXXXXXXXXXXX
	95	73.05	100.8	94.0	XXXXXXXXXXXXX
	90	69.2	95.7	93.8	XXXXXXXXXXXXX
	85	65.36	91.2	93.0	XXXXXXXXXXXXX
	80	61.51	86.7	92.0	XXXXXXXXXXXXX
	75	57.67	92.2	91.0	XXXXXXXXXXXXX
	70	53.82	77.7	89.8	XXXXXXXXXXXXX
	60	46.13	68.8	87.0	XXXXXXXXXXXXX
	50	38.45	59.7	83.5	XXXXXXXXXXXXX
	45				XXXXXXXXXXXXX
	40				
	30				
	20				
	__(idle)				

NOTES: (1) = power at ~~100 % of nominal opening.~~ Cavitation Limit

2.6.2 For H1 = 75.5 m

NAME AND SIGNATURA OF MANUFACTURER

Date : . . . .

NAME AND SIGNATURA OF DESIGNER

Date: . . . .

RAPEL POWER PLANT

DESIGN; MANUFATURE AND ERECTION OF PIECES FOR THE REHABILITATION OF THE TURBINE N°5

GUARANTEED CHARACTERISTICS

NET HEAD (H) (m)	POWER (P) (%)	POWER (P) (MW)	FLOW (Q) (m^3/s)	EFFICIENCY (d) (%)	LIMITS OF OPERATION(MW)
H1 = 75.5	____(Máx)				
	100 (1)	73.68	107	93.1	XXXXXXXXXXXXX
	95	70.00	100.7	94.0	XXXXXXXXXXXXX
	90	66.31	95.4	94.0	XXXXXXXXXXXXX
	85	62.63	90.9	93.2	XXXXXXXXXXXXX
	80	58.94	86.4	92.2	XXXXXXXXXXXXX
	75	55.26	81.9	91.2	XXXXXXXXXXXXX
	70	51.58	77.5	90.0	XXXXXXXXXXXXX
	60	44.21	68.6	87.1	XXXXXXXXXXXXX
	50	36.84	59.6	83.6	XXXXXXXXXXXXX
	45				XXXXXXXXXXXXX
	40				
	30				
	20				
	__(vacio)				

NOTAS: (1) = power at ~~100 % of nominal opening~~ Cavitation Limit

2.6.3 For H2 =72.3 m

NAME AND SIGNATURA OF MANUFACTURER

Date : . . . .

NAME AND SIGNATURA OF DESIGNER

Date: . . . .

GUARANTEED CHARACTERISTICS

NET HEAD (H) (m)	POWER (P) (%)	POWER (P) (MW)	FLOW (Q) (m^3/s)	EFFICIENCY (ɖ) (%)	LIMITS OF OPERATION(MW)
H2 = 72.3	____(Max)				
	100 (1)	70.25	107	92.7	XXXXXXXXXXXXX
	95	66.74	100.5	93.8	XXXXXXXXXXXXX
	90	63.23	95.1	93.9	XXXXXXXXXXXXX
	85	59.71	90.5	93.2	XXXXXXXXXXXXX
	80	56.2	86.1	92.2	XXXXXXXXXXXXX
	75	52.69	81.7	91.1	XXXXXXXXXXXXX
	70	49.18	77.2	90.0	XXXXXXXXXXXXX
	60	42.15	68.2	87.3	XXXXXXXXXXXXX
	50	35.13	59.2	83.8	XXXXXXXXXXXXX
	45				XXXXXXXXXXXXX
	40				
	30				
	20				
	__(idle)				

NOTES: (1) = power at ~~100 % of nominal opening~~ Cavitation Limit

2.7 Max Efficiency of the turbinewith net head of:
H1 = 75.5 m, ɖ = _**94.1**_ __ % for POWER of ___**68.5**_____ MW

2.8 Characteristics curves of the turbine ("Universal Chart"):
$Q_{11}(Q/D\sqrt{H})$ *versus* $\xi_{11}:(\xi x\, D/\sqrt{H})$.1
for openings (Ao), from "opening for idle" to "max opening".

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Date : . . . .

NAME AND SIGNATURA OF DESIGNER

Date: . . . .

GUARANTEED CHARACTERISTICS

(Zones beyond the guaranteed range (normal operation) will be just for reference.)

As parametrics curves should be detailed the following:

2.8.1 equal EFFICIENCYs (ɖ constant).
2.8.2 equal opening of distributor (Ao constant).
2.8.3 equal cavitation coefficient (ů constant).

GRAPHS Nº: ____

3. **SPEEDS**

3.1 Nominal rotation speed: 187.5 rpm
3.2.1 Max original runaway speed: 358 rpm
3.2.2 Max proposed runaway speed: ______**350**_____________rpm (78.8 m)
3.3.1 First critical speed (original) of set shafts turbine-generator: 465 rpm (1.3 x runaway speed)
3.3.2 First critical speed (proposed) of set shafts turbine-generator with new runner : ___________________ rpm
3.4.1 Specific speed (original) for nominal power and net head H1 = 75.5 m (system rpm, kW, m) : 229
3.4.2 Proposed specific speed for the nominal POWER nominal and net head H1 = 75.5 m (system rpm, kW, m) : __________**237**___________

4. **HIDRAULIC PUSHMENT**

4.1 Max hydraulic pushment over the runner with un-wear seals: proposed: 1,290 kilo Newtons
4.2 Max hydraulic pushment over the

NAME AND SIGNATURA OF MANUFACTURER

Date : . . . .

NAME AND SIGNATURA OF DESIGNER

Date: . . . .

6. HYDRAULIC DESIGN AND PERFORMANCE

6.1 Runner Design Features

The Rapel turbine runners have unusually short bands. This style of design has very little blade surface to develop the proper torque with reasonable cavitation resistance. Modern designs utilize deeper runners which develop high torque over larger blades. This approach provides high efficiency with light blade loading, thereby eliminating damaging cavitation.

American Hydro has designed and computer tested a range of runner designs to optimize the Rapel design. To extend the runner band two approaches are possible. The runner can be extended into the draft tube as shown in Drawing No. 23634-D. This is a very effective approach which will work well at Rapel. Although there is a slight decrease in efficiency due to the expansion of the water leaving the runner, the substantial benefit of this design is that it fits the existing waterpassages with no modifications. This is very helpful with fully embedded draft tubes.

Because the upper draft tube section is removable at Rapel, American Hydro concentrated on runner designs which exactly fit a redesigned draft tube. Drawing No. 23689-D shows the final design. By replacing the upper draft tube section we not only provide the highest possible efficiency, but also change the draft tube aeration, removing piping from below the runner.

The existing Rapel runner has two sealing surfaces on the band (top and bottom). This design, while slightly reducing the leakage flow, can lead to damaging runner pulsations and oscillations. As shown on Drawing No. 23634-D American Hydro has eliminated the lower seal. The single seal design is consistent with modern design practice.

The extensive aeration piping used below the current Rapel runners is highly unusual for a moderate head Francis turbine. American Hydro does not anticipate that the new runner will need aeration operation over the normal operating range. Field tests have demonstrated that similar pipes below the runner do reduce efficiency.

To be conservative American Hydro has chosen to maintain aeration below the runner through the walls of the new upper draft tube section. The primary aeration will, however, be provided through the runner nose cone. American Hydro has extensive experience in the CFD analysis and design of aeration systems which provide large volumes of aspirated air. Field tests have shown that aeration through the nose cone is effective in eliminating low load pressure pulsations and improving low load efficiencies.

The attached Appendix 1 displays American Hydro's work with aerating turbines. For Rapel, American Hydro will design a new aeration system and provide the necessary passages through the runner and nose cone. Modifications to the headcover, etc. are not included as part of this proposal. We will work with ENDESA to provide the best overall aeration system.

6. HYDRAULIC DESIGN AND PERFORMANCE (continued)

6.2. Detailed Runner Design

The state of the art replacement runner will be designed using the **American Hydro Runner Design System ("AHRDS")**. Developed by American Hydro engineers, **AHRDS** provides a computer interactive design environment with complete flexibility for runner shape definition.

The **AHRDS** system accomplishes two major functions. The geometry generation portion provides rapid computer-interactive runner hardware design. The fluid flow analysis "tests" the hydraulic performance of the runner geometry to determine power, efficiency, and cavitation resistance. The three-dimensional finite element flow analysis provides a fully three-dimensional calculation of water velocity, pressure, and energy for the wicket gates or for the runner. Boundary layer calculations are used to evaluate the fluid losses near the surface of the blades. This style of analysis has proven to be very accurate in determining turbine power and cavitation performance as shown by extensive model testing.

Once the hydraulic design is completed, the **AHRDS** program provides the numerical design data necessary for the mechanical design, structural analysis and for computer numerically controlled manufacturing of the runner. The quality and conformance to design of the manufactured runner are thereby assured through the single computerized geometry definition that is used in all phases of engineering and fabrication.

Figures 6.1 and 6.2 present the new higher power design for Rapel. Our design studies have demonstrated that a best efficiency flow of 107 cms and a full load flow of 115 cms can be accommodated by a modern, high efficiency design. Naturally, a lower capacity design with 99 cms design flow and 107 cms full load flow is very comfortable. (The low capacity design is in accordance with the specification and represents the base case proposal.) Because the high power design is more challenging, we have concentrated on this design for the CFD performance analysis.

6.3 Cavitation Performance

Every new runner produced by American Hydro is a new custom design. While previous model tests provide guidance for our engineers, each design is unique. The runner performance is not estimated from older designs. It is calculated using state-of-the art computational fluid dynamics (CFD). The AHRDS CFD analysis has been particularly effective in predicting the onset of cavitation. The predicted onset correlates well with the beginning of visual cavitation (sigma-begin or sigma-incipient) as shown in laboratory testing. However, the CFD analysis can only approximately estimate the sigmas (Thoma cavitation coefficients) at 0% and 1% change in efficiency. For this proposal we only present the calculated values of sigma-incipient. (Note that this is the most conservative sigma value.)

For the high power runner design Figures 6.3 and 6.4 present the suction side pressure patterns at 107 and 115 cms respectively. This CFD analysis has been completed at the normal tailwater of 24.5 m. These analyses show only a small region of low pressure at

6. HYDRAULIC DESIGN AND PERFORMANCE (continued)

the leading edges. This will not result in damaging cavitation. These figures demonstrate excellent cavitation performance for the new runner, even at 115 cms.

Figure 6.5 shows 115 cms analyzed at a tailwater of 23.5 m. Here the blade shows the start of cavitation. Sigma-begin for this runner, at full load, is calculated to be 0.143. Figure 6.6 shows the suction side pressure pattern for 117 cms at normal tailwater. This is the maximum flow allowed for normal tailwater. This flow is noted on the performance curves.

6.4 Pressure Pulsations

Pressure pulsations occur for part load operation with any Francis turbine. The magnitudes of the pulsations depend on draft tube design, runner design, and on the interaction of the pressure fluctuations with the rest of the electrical and hydraulic system. It is not currently possible to predict the magnitude of pressure pulsations with CFD analysis.

Model testing of the new runner can provide some qualitative effects of pressure pulsations. However, the magnitudes of the pulsations are not directly scalable. Furthermore, total system interactions are not modeled in the laboratory.

No levels of pressure pulsations are guaranteed with this proposal. However, American Hydro recently model tested a runner design for a turbine very similar to Rapel. Section 6.5 presents results from these tests. Model test pressure pulsations are included.

6.5 Performance Analysis

To establish the performance of the new custom designed runner in the existing wheelcase each of the turbine components must be evaluated using CFD to predict the component loss in efficiency. By comparing the Rapel waterpassage losses with those of a modern design the overall turbine efficiency is estimated.

For this analysis the design head is 75.5 m. The design flow is 107 cms.

Spiral Case – Detailed drawings of the spiral case were not available. However, based on scaled drawings the size of the case is ample for the design flow. No additional losses were calculated.

Stay Vanes – Flow analysis of the stay vanes demonstrated that the vane angles are too radial. The flow from the spiral case is much more tangential. The calculated additional loss is 1.07% in efficiency. American Hydro has had good success recontouring stay vanes to improve efficiency. (One example is shown in Section 6.6.) For Rapel, we estimate that the efficiency could be improved about 0.3% with a modest recontouring of the vanes. We are willing to work with Endesa to design and implement this modification. However, for this proposal we have guaranteed efficiencies with no change to the stay vanes.

6. HYDRAULIC DESIGN AND PERFORMANCE (continued)

Wicket Gates – The wicket gates are of modern design and should perform well.

Runner – The runner is a fully modern design. Total skin friction loss for this runner is only 1.2% in efficiency.

Draft Tube – The Rapel turbine has a moderately high specific speed. Such designs have relatively high flows with moderate to lower net heads. Because fluid velocities entering the draft tube are relatively high, draft tube performance is critical to unit efficiency.

To evaluate draft tube performance American Hydro uses the Fluent code to calculate a fully three-dimensional Navier-Stokes analysis of the flow. Our analysis has been developed in coordination with Pennsylvania State University to best evaluate draft tube losses. Appendix 1 presents typical Fluent analyses.

For Rapel the calculated draft tube efficiency loss is 2.6%. This is a relatively low loss. The additional loss compared with a fully modern draft tube is just 0.5%.

For an all modern model turbine designed for Rapel we would expect a model efficiency of 93.72%. (This model efficiency is referred to a model Reynolds number of 8×10^6 based on the adjustments in IEC 60193.) Based on the above analysis the expected Rapel model efficiency is calculated as:

Modern Model -	93.72%
Spiral Case -	0.0%
Stay Vanes -	-1.07%
Wicket Gates -	0.0%
Runner -	0.0%
Draft Tube -	-0.5%
Rapel Model -	92.25%

With an expected model efficiency of 92.25% the step-up formulas of IEC 60193 are used to calculate an expected prototype (peak) efficiency of 94.11%. Figures 6.7 and 6.8 present expected performance curves for Rapel for the base case runner (99 cms best efficiency, 107 cms full load) and for the high power runner (107 cms best efficiency, 115 cms full load). Note that all efficiencies presented in this proposal are based on net head measured in accordance with IEC 60193. All measured efficiencies are subject to test tolerances in accordance with IEC the applicable IEC test codes.

6. HYDRAULIC DESIGN AND PERFORMANCE (continued)

6.6. Comparable Model Test Results

To further support the expected and guaranteed performance presented in this proposal American Hydro can present model results from an independent test lab. The tested turbine was upgraded with a custom runner designed and produced by American Hydro. The results presented here are for the turbine with no modifications except for runner replacement. Additional testing showed that stay vane modifications improved efficiency by an additional 0.3%. (Figure 6.9 shows the stay vane modifications.)

	Rapel	**Prototype of Tested Model**	**Tested Model Scaled to Rapel**
Runner Blade Inlet Diameter - m	3.646	4.50	3.498
Runner Discharge Diameter – m	3.482	4.48	3.482
Wicket Gate Height – m .	0.974	1.219	0.947

The Rapel full load condition can be transferred to prototype conditions for the tested model as follows:

	Rapel	Prototype for Tested Model	Rapel Adjusted to Prototype of Tested Model
Speed	187.5	105.9	105.9
Diameter – m	3.482	4.48	4.48
Head – m	75.5	39.63	39.87
Flow – cms	115	143	138

Figure 6.10 shows the prototype performance at a head very close to the adjusted Rapel head of 39.87 m. Here we see that the adjusted Rapel flow of 138 cms is easily exceeded by this high power design.

Figure 6.11 shows the model hill curve. The peak model efficiency of 92.3% would reach 92.6% with a stay vane modification. Note also that Rapel is a lower capacity design. Slightly higher efficiencies can be expected.

Figures 6.12 and 6.13 show sigma breaks for two heads at the flow equivalent to 115 cms at Rapel. The Rapel plant sigma of 0.156 is above 10%.

Figure 6.14 presents pressure pulsation results indicating pulsations at low loads and no significant pulsations at moderate and high loads.