CONCEPTUAL DESIGN OF A HYDROELECTRIC POWER PLANT FOR A REHABILITATION PROJECT

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ABSTRACT

This study presents the conceptual design of a hydroelectric power plant, as a part of a large scale rehabilitation project for an existing power plant in Antalya, Turkey. The aim of the rehabilitation project is to increase the power and efficiency of the plant and its scope includes CFD aided turbine design, model production and tests, the design, production and implementation of the turbine, generator and the SCADA system. This study is the first attempt, as a preliminary study, to handle the problem and perform a conceptual design of the hydroelectric power plant. The existing plant is modeled to estimate the head and flow rate characteristics at various sections of the system. The net head and flow rate of the turbine are estimated. Transient analyses of the system are also performed to evaluate water hammer characteristics. The results of the transient analyses provide the inputs for the design of by-pass pipeline and pressure relief valve. The estimated net head and flow rate from the simulations are used as inputs for the preliminary design. The dimensions of the spiral case, the diameter of the stay vanes and guide vanes, wicket gate heights, runner diameter and rotational speed, runaway characteristics and preliminary output power are determined. The best efficiency point and the design point of the turbine are also obtained as the net head versus the flow rate. These results provide an idea on the feasibility of the increase in power.

INTRODUCTION

Hydroelectric power is one of the renewable energy sources for electricity generation. Hydroelectric power plants produce approximately 20% of the total electricity in the world [1]. Francis et al. developed the reaction turbine in 1848 [2]. Although the design methodology changed significantly, working principle of a Francis turbine is the same as in the past. Design of turbines was solely based on experimental research using scaled turbines or model tests in the past [3]. This design method was of course expensive and it was restricted. It mostly depended on the experience of the engineers and researchers [4]. In the last decades, the improvement of computational power led to use it for turbine design. The first application of Computational Fluid Dynamics (CFD) for turbine design was in two dimensions in 1970's [5]. The development of numerical methods allowed the 3D Reynolds averaged Navier-Stokes (RANS) equations for hydraulic turbines to be solved [6]. Nowadays, the improvement of computational power and numerical methods let CFD results predict the turbine performance accurately. CFD methods are also used in rehabilitation projects because of high accuracy level of results [7].

Kepez-I Hydroelectric Power Plant that is located in Antalya, Turkey was set into operation in 1962. Water goes to Kepez from a water source through a conveyance channel. The water is carried to the power plant by the penstock that consists of two parts. The first part of the penstock is made of concrete and has a length of 650 m. The second part that has a length of 770 m and it is made of steel. A surge tank is located between the concrete and the steel penstock. Three vertical Francis type turbines are used to operate the power plant. Each turbine produces 8.8 MW power; therefore the power plant has a power capacity of 26.4 MW. A general view of the power plant and elevations of water source, surge tank, power plant and tail water are shown in Figure 1.

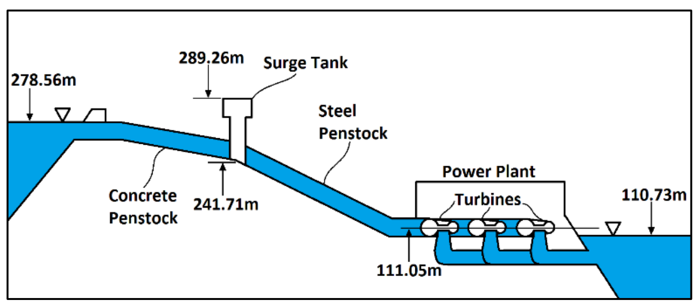

Figure 1.General view of the hydroelectric power plant

Penstock is used to connect the power plant to the water source. The thickness and diameter of the penstock are determined according to the water pressure. The surge tank regulates the pressure fluctuations so it allows the penstock to remain in compact sizes and allows the turbine to work efficiently. The effect of these pressure fluctuations which constitute an unsteady hydraulic problem is called water hammer [8]. Water hammer can occur in different situations such as load acceptance, load rejection and instant load rejection conditions. The pressure drops in the penstock in load acceptance condition when the guide vanes open suddenly. On the other hand, pressure increases in the penstock in load rejection condition when the guide vanes close suddenly. Water hammer can cause major damage to the turbines, valves or the penstock. The surge tank is sufficient to protect the penstock that is between the water source and itself. However the penstock that is between the surge tank and the power plant should also be protected; therefore a pressure relief valve (PRV) can be used in these conditions [8]. PRV is a valve that opens in a defined pressure. It opens when the pressure exceeds the limit value in the pipeline and high pressure values are avoided in the system.

This study presents the conceptual design of Kepez 1 HEPP as a part of a large scale rehabilitation project. The aim of the rehabilitation project is to increase the power and efficiency of the plant and its scope includes CFD aided turbine design, model production and tests, the design, production and implementation of the turbine, generator and the SCADA system. This study is the first attempt, as a preliminary study, to handle the problem and perform a conceptual design of the hydroelectric power plant that is going to be rehabilitated.

NOMENCLATURE

A	[m²]	Flow Area
C	-	Hazen-Williams Roughness Coefficient
g	[m/s²]	Gravitational acceleration constant
H	[m]	Head
h_L	[m]	Head loss
h_m	[m]	Minor loss
h_T	[m]	Turbine loss
I	[kgm²]	Moment of Inertia
k	-	Hazen-Williams Eqn. Constant
K_m	-	Minor loss coefficient
n	-	Safety Factor
P	[Pa]	Pressure
R	[m]	Hydraulic radius
S	[m/m]	Friction slope
r_{in}	[m]	Inlet radius of the pipe
t	[m]	Thickness of the pipe
Q	[m³/s]	Flow rate
V	[m/s]	Velocity
z	[m]	Elevation

Special characters

ω	[rpm]	Rotational speed of the turbine runner
σ_{allow}	[Pa]	Allowable yield strength
γ	[N/m³]	Specific weight

Subscripts

in	inlet

METHODOLOGY

The existing plant is modeled using WaterCAD software [9] to estimate the head and flow rate characteristics at various sections of the system. The water network is controlled by two fundamental physical laws, Conservation of Mass and Energy principle. This software solves for the distributions of flows and hydraulic grades using the Gradient Algorithm. This algorithm is used for analysis of pipe flows. The model includes the water source and tail water as reservoir, the penstock, surge tank, junctions and turbines. The dimensions and elevations of the components are defined according to the technical drawings of the actual plant. The model is simulated to obtain the major and minor losses of the pipes and junctions, pressure losses and head losses. These changes in head values, based on friction (major losses) and specific shape of the fitting (minor losses), obtained with the energy equation [9];

$$\frac{P_1}{\gamma} + z_1 + \frac{V_1^2}{2g} = \frac{P_2}{\gamma} + z_2 + \frac{V_2^2}{2g} + h_T + h_L \tag{1}$$

The major losses of the system is calculated as;

$$Q = k * C * A * R^{0.63} * S^{0.54} \tag{2}$$

The minor losses of the system is calculated as;

$$h_m = K_m * \frac{V^2}{2 * g} \tag{3}$$

Consequently, the net head and flow rate of the turbine are estimated. The dimensions of the components that are used in the model, are given in Table 1 and the details of the turbines are given in Table 2.

Table 1. Dimensions of the components

Part	Diameter (mm)	Material	Length (m)
Concrete Penstock-1	2500	Concrete	340
Concrete Penstock-2	2500	Concrete	320
Penstock-1	2400	Steel	73.2
Penstock-2	2400	Steel	158
Penstock-3	2400	Steel	95.1
Penstock-4	2130	Steel	121
Penstock-5	2130	Steel	64.7
Penstock-6	2130	Steel	149
Penstock-7	2130	Steel	98
Penstock-8,9	1800	Steel	7.5
Branch 1,2,3	1100	Steel	9.12
Tailwater 1,2,3	1500	Concrete	5

Table 2. Details of the turbines

Turbine	Elevation (m)	Efficiency (%)	Rotational Speed - ω (rpm)	Moment of Inertia - I (kgm^2)
Turbine 1, 2, 3	111.05	92	750	190.345

Several parts of the model of the power plant are given in Figure 2, Figure 3 and Figure 4. The full model is given in Annex A.

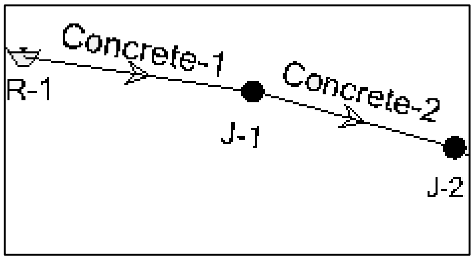

Figure 2. Water source and the concrete part of the penstock

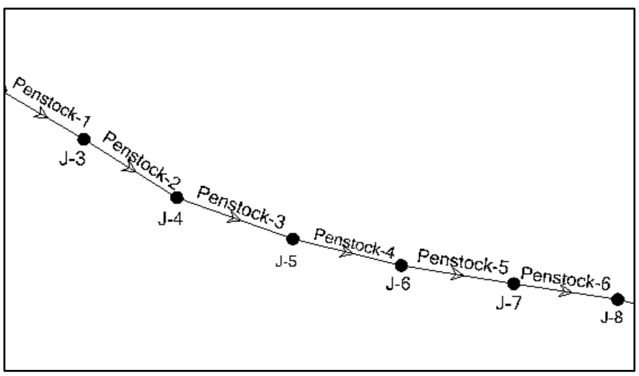

Figure 3. Steel part of the penstock

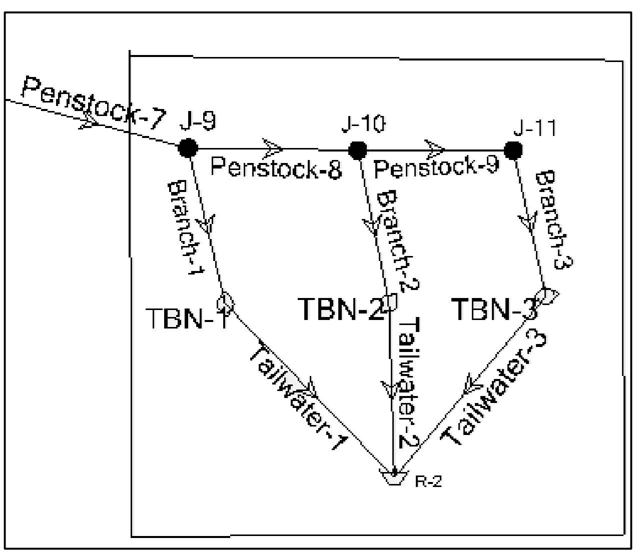

Figure 4. Branches, turbines and the tail water

The results of the steady-state analyses show the minor and major losses of the pipes and junctions, pressure losses and head losses. Therefore, the net head and flow rate of the turbine are estimated and these values determine the design point of the turbine.

The same model is also used in the transient analyses of the system to evaluate water hammer characteristics. The results of the transient analyses provide the inputs for the design of by-pass pipeline and pressure relief valve.

Water hammer can occur in load acceptance, load rejection or instant load rejection conditions. The higher pressure values occur in the load rejection condition among the other cases so the load rejection case is studied in this study. To illustrate, the guide vanes should be closed in a short time that could be called emergency shut-down time, without causing damage to the penstock and the pipeline in case of power cut-off. Different from the steady-state analyses, the wave speed is determined.

The maximum allowable pressure in the pipeline should be determined according to Hoop stress and longitudinal stress [10]. The Hoop stress for a cylindrical vessel is calculated as;

$$\sigma_{allow} = \frac{P * r_{in}}{t} \tag{4}$$

The longitudinal stress for a cylindrical vessel is calculated as;

$$\sigma_{allow} = \frac{P * r_{in}}{2t} \tag{5}$$

This pressure value is the limit of the pipeline that the resulting maximum pressure of the transient analysis should not exceed. The shut-down time of guide vanes is given randomly as a starting point and the pressure value is checked in order to determine whether the pipeline has enough strength. If the pipeline does not have enough strength, the time is increased and iterative process is repeated.

Preliminary design of the turbine components are based on the statistical data of the existing hydraulic turbines in the literature [11]. Maximum, minimum and gross head values, the flow rate, water temperature and system frequency are the inputs for the preliminary design. Different sizes of runner diameters are obtained due to their specific speed and rotational speed. The most convenient size of the turbine runner is chosen due to the settlement plan of the power plant and the dimensions of the other components such as the diameters of stay vanes and guide vanes, wicket gate height, spiral case size and draft tube size are determined. These dimensions are the starting point of the CFD aided design. The working range of the turbine is also determined for various flow rate values.

RESULTS AND DISCUSSION

As a result of the steady-state analyses the major and minor losses of the pipes and junctions, pressure losses and head losses; therefore the net head and flow rates are estimated. The minor, head and pressure losses of components are given in Table 3. The total pressure loss is calculated as 85.3 kPa and total head loss is calculated as 7.88 m. The flow rate of the system is calculated as 18.03 m^3/s. The turbine properties are given in the Table 4.

Table 3. The results of the steady-state analyses

Part	Minor Loss (m)	Head Loss (m)	Pressure Loss (kPa)
Concrete Penstock-1	0.34	1.28	12.5
Concrete Penstock-2	0	0.88	8.6
Penstock-1	0.02	0.27	2.6
Penstock-2	0	0.53	5.2
Penstock-3	0	0.32	3.1
Penstock-4	0.09	0.82	8
Penstock-5	0	0.39	3.8
Penstock-6	0	0.9	8.8
Penstock-7	0.78	1.37	13.4
Penstock-8	0.77	0.82	8
Penstock-9	0.27	0.34	3.3
Branch-1	0.63	0.81	8
Branch-2	0.51	0.69	6.7
Branch-3	0.2	0.38	3.7
Tailwater 1,2,3	0	0.03	0.3

Table 4. Turbine properties

Turbine	Rotational Speed - ω (rpm)	Head loss (m)	Flow rate (m3/s)
Turbine-1	750	160.1	6.03
Turbine-2	750	159.3	6.00
Turbine-3	750	159.2	6.00

Guide vane closing time is decided by the strength of the pipeline. The critical value is obtained on the branch pipe so the highest pressure occurs before the turbine. Hoop stress is calculated as 1.90 MPa and the longitudinal stress is calculated as 3.80 MPa. The yield strength of St50 steel is 295 MPa. Allowable yield strength is calculated as 147.5 MPa for a safety factor of 2. The inner radius of the branch is 543 mm and the thickness is 7 mm.

Allowable pressure equals to 1.90 MPa and guide vane closing time is calculated using this pressure. Firstly, the guide vanes are fully open for the first period that is for 10 seconds. The guide vanes start to close in the second period for 10-40 seconds. They are fully closed after 40 seconds. The time schedule of the guide vane closing is shown in Figure 5.

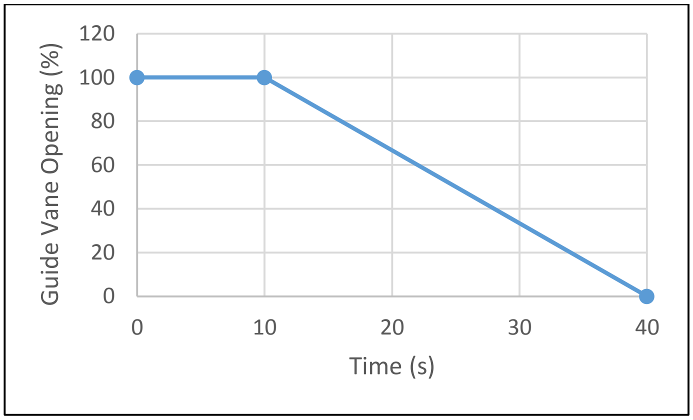

Figure 5. First scenario for guide vane closure

As a result of this scenario, the maximum pressure occurs on the branch and equals to 2.007 MPa that exceeds the allowable pressure of the pipeline so a second scenario is developed. The guide vanes are open for 10 seconds in the first period. They start to close in the second period that is for 10 – 55 seconds. The second scenario is shown in Figure 6.

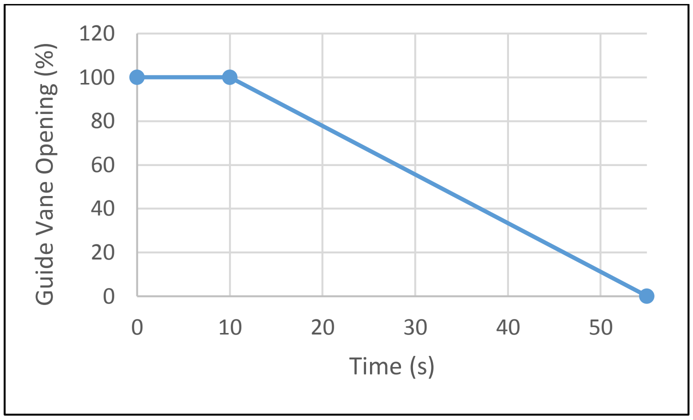

Figure 6. Second scenario for guide vane closure

As a result of this scenario, the maximum pressure occurs on the branch and it is equal to 1.864 MPa that does not exceed the allowable pressure of the pipeline. The maximum pressure values of the pipeline is given in Table 5. Consequently, the guide vane closing time is calculated as 45 seconds so that water hammer does not damage the penstock and the pipeline.

Table 5. The maximum pressure values on the branch pipe for the second scenario

End Point	Max Pres. (MPa)
Branch 1	1.881
Branch 2	1.882
Branch 3	1.884

The results of the steady-state analysis are the inputs of the preliminary design. The head losses and pressure losses are calculated so the net head and flow rate are estimated with the help of steady-state analysis. The final specifications of the power plant are given in Table 6.

Table 6. Specifications of the power plant

Rated discharge (m3/s)	6.1
Net head (m)	162
Site gross head (m)	170
Site elevation (m)	111.05
Water temperature	200 C
Minimum net head (m)	155.05
Maximum net head (m)	167.75

CONCLUSION

It is decided based on the preliminary design that, the guide vanes, turbine runner and conical part of the draft tube have to be redesigned. The diameter of the new turbine runner must be the same as the old one because the spiral case and the stay vanes are under concrete and cannot be changed. Preliminary design results provide an idea on the feasibility of the increase in power. According to the results, new turbine runner can produce 8.9 MW of power with an efficiency of 93% These dimensions will also be the starting point of the CFD aided design that will be used to increase the efficiency of the power plant.

ACKNOWLEDGMENTS

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REFERENCES

[1] World Energy Council Turkish National Committee Energy Report, 2006.
[2] Francis J. B., Lowell Hydraulic Experiments, 5th edition, Van Nostrand, *Princeton*, New Jersey, 1909.
[3] Ardizzon G., Cavazzani G., and Pavesi G., A new generation of small hydro and pumped-hydro power plants: Advances and future challenges, *Renewable Sustainable Energy Reviews*,vol. 31, 2014, pp. 746-761.
[4] Carija Z., Mrsa Z., and Fucak S., Validation of Francis water turbine CFD simulations, *Strojarstvo*, vol. 50,2008, pp. 5-14.
[5] Roache P. J., Computational Fluid Dynamics,*Hermosa Publishers,* Albuquerque, 1972, New Mexico.
[6] Keck H., and Sick M., Thirty years of numerical flow simulation in hydraulic turbomachines, *Acta Mechanica*, vol. 201, 2009, pp. 21-229.
[7] Muntean S., - Numerical Analysis of the flow in the old Francis runner in order to define the refurbishment strategy, *U.P.B. Scientific Bulletin*, Series D,2010, pp.117-124.
[8] Calamak M., and Bozkus Z., , Protective measures against waterhammer in run-of-river hydropower plants, 10th *International Congress on Advances in Civil engineering*, Ankara.,2012.
[9] Bentley WaterCAD Getting Started Guide, 2010
[10] Hibbeler, R.C. Strength of Materials 8th Edition, *Prentice Hall*, New Jersey., 2008.
[11] Siervo, F, Leva, F, Modern trends in selecting and designing Francis turbines, *Water Power & Dam Construction* 28(8),1976, pp. 28-35.

ANNEX A

MODEL OF THE POWER PLANT