

CFD BASED HILL CHART CONSTRUCTION AND SIMILARITY STUDY OF PROTOTYPE AND MODEL FRANCIS TURBINES FOR EXPERIMENTAL TESTS

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ABSTRACT

According to the growing energy demand in Turkey, Hydro Energy Research Laboratory (ETU Hydro) at TOBB University of Economics and Technology was established which is responsible for the Computational Fluid Dynamics (CFD) aided design, manufacturing and standardized performance tests of model hydraulic turbines. The hydroturbine design process at the test center is mainly composed of five steps: preliminary design, CFD aided design, model-prototype similarity analysis, model manufacturing and model tests according to International Electrotechnical Commission (IEC) standards. This study focuses on model-prototype similarity analysis for the turbines to be tested. By using reduced quantities, discharge factor and speed factor, hill chart diagram is obtained for the model using CFD techniques and compared with the hill chart obtained for the prototype. Overall hydraulic characteristics, hydraulic losses, cavitation characteristics are found out for the model. Scale effects on the model turbine are also investigated.

INTRODUCTION

With increasing computational power, CFD methods became a dominant tool in the design process of hydroturbines since 1990's [1]. On the other hand, sufficiently complete and reliable characteristics of turbines, which would cover a wide range of operating conditions can only be obtained by experimental methods [2,3]. In the model tests; performance measurements such as power, efficiency, flow rate, head, temperature, and torque are carried out in the sight of international standard of hydraulic turbine model tests, IEC 60193 [3,4].

In order to test hydroturbines, as the most turbine dimension, power, head and flow rate are over the capacity of test centers, model turbine tests are conducted. This procedure involves scaling down the prototype to a model based on IEC 60193 standart [3-5].

Model test conditions should provide a small scale of a real hydropower station, which has all the corresponding features of the actual power plant. For this purpose, model tests require geometric, kinematic and dynamic similitude between model and prototype [2,6,7]. In [2], similarity laws are given which are used to obtain the performance and dimensions of a model turbine if the performance of prototype turbine is given. The first step of the model test determines the operating conditions

and dimensions of the model turbine to ensure that the model and prototype modes of operation are similar.

In this study, model turbine dimensions and operating conditions of the KEPEZ 1 HEPP, in Turkey is determined which will be tested at the Center of Hydro Energy Research (TOBB ETU HYDRO Lab). Model tests will be performed for verification purposes. The overall hydraulic characteristics of the model are determined; several analyses are performed to be able to perform the CFD aided design and model tests of the designed turbine. Efficiency values over a wide range of operating conditions are obtained by CFD analyses for the model turbine by conducting eighty full turbine analyses and numerical hill charts are constructed. Flow characteristics, structures of losses in the actual turbine and in the model are compared. Scale effects between the model and prototype are investigated.

NOMENCLATURE

D	[m]	Runner Diameter
H	[m]	Head
n	[rpm]	Rotational Speed
Q	[m ³ /s]	Volumetric Flow Rate
v, u, w	[m/s]	Velocity components
P	[W]	Power
x	[m]	Cartesian axis direction
y	[m]	Cartesian axis direction
z	[m]	Cartesian axis direction
Special characters		
n_1'	[m ^{0.5} s ⁻¹]	Reduced speed parameter
Q_1'	[m ^{0.5} s ⁻¹]	Reduced Flow rate parameter
ψ	-	Energy Coefficient
ϕ	-	Discharge Coefficient
Subscripts		
1		Prototype
2		Model Turbine

AFFINITY LAWS

Generally, most turbines are too big to be tested at model test laboratories therefore prototype is scaled down to a model turbine [4, 5]. In the model tests, model turbine should reflect the full scale turbine for meaningful results. For this purpose, a scaled model should have fully similar geometric, kinematic

and dynamic attributes to eliminate scaling effects. According to IEC 60193 standards [3], to achieve hydrodynamic similarity between two hydraulic machines, these two machines should have geometrical similitude and all of the force ratios should be identical. Two turbines with different size of one type should meet the following conditions [2].

- All turbine angles should be alike:
 $\delta_{i1} = \delta_{i2}$
- The ratio of dimensions should be constant:

$$\frac{D_{11}}{D_{12}} = \frac{D_{21}}{D_{22}} = \frac{b_{01}}{b_{02}} = \dots$$

- Ratios of all velocities should be similar to have similar liquid passage:

$$\frac{v_{i1}}{v_{i2}} = \frac{u_{i1}}{u_{i2}} = \frac{w_{i1}}{w_{i2}} = \text{constant}$$

Velocity triangles and runner angles are shown in Figure 1. Based on the definitions given above, similarity equations of turbines of one type under conditions of similar modes of operation can be written (equation 2-5) [2].

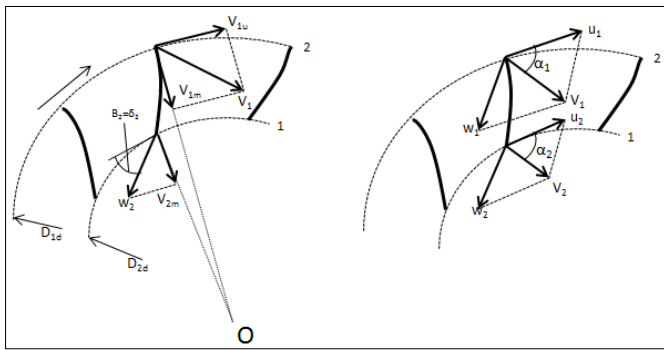


Figure 1. Velocity parallelograms and triangles of a Francis turbine runner [adapted from ref. 2]

Kinematic condition for the similarity of turbine modes of operation are given as:

$$\frac{Q_1}{n_1 D_1^3} = \frac{Q_2}{n_2 D_2^3} \quad (1)$$

where; N is rotational speed, D is runner diameter, and Q is the flow rate. Subscript 1 denotes prototype properties and subscript 2 denotes model properties. By modifying the discharge similarity, power and rotational speed ratios are formulated as in equations (4) and (5):

$$\frac{n_1}{n_2} = \frac{D_2}{D_1} \sqrt{\frac{H_1}{H_2}} \quad (2)$$

$$\frac{P_2}{P_1} = \left(\frac{N_1}{N_2}\right)^3 \left(\frac{D_1}{D_2}\right)^5 \quad (3)$$

where; H is the head and P is the power. The efficiency terms are neglected, due to the difference between efficiencies being smaller than 1%. With satisfying the geometrical and kinematic similarity principles, dimensionless terms are defined which determine the hydraulic characteristics of the machine and given in equations (4) and (5). Values of the dimensionless parameters remain constant under similar modes of operation; therefore reduced parameters are widely used to see the relationship between the prototype and the reduced scale model [3].

$$\varphi = \frac{4Q'_1}{\pi n'_1} \quad (4)$$

$$\psi = \frac{2g}{\pi^2 (n'_1)^2} \quad (5)$$

Using known parameters, discharge and speed, Q'_1 and n'_1 can be defined which are used to provide the definition of φ and ψ , the dimensionless discharge and the energy coefficients.

$$n'_1 = \frac{nD}{\sqrt{H}} \quad \& \quad Q'_1 = \frac{Q}{D^2 \sqrt{H}} \quad (6)$$

Model hydraulic efficiency measured at the test will be scaled up to prototype efficiency with the equation given below.

$$(\Delta\eta_h)_{M^* \rightarrow P} = \delta_{ref} \left[\left(\frac{Re_{ref}}{Re_{M^*}} \right)^{0.16} - \left(\frac{Re_{ref}}{Re_P} \right)^{0.16} \right] \quad (7)$$

$$\Delta\eta_{hP} = \eta_{hM^*} + (\Delta\eta_h)_{M^* \rightarrow P} \quad (8)$$

Here, Re is Reynolds number, Re_p is prototype Reynolds number, Re_M is model Reynolds number, Re_{ref} is 7×10^6 and δ_{ref} is defined as $\delta_{ref} = \frac{1 - \eta_{opt}}{\left(\frac{Re_{ref}}{Re_{opt}} \right)^{0.16} + \frac{1 - V_{ref}}{V_{ref}}}$.

According to IEC, both model and prototype machines should have same discharge, cavitation and energy coefficient. The equality of these coefficients characterizes the hydraulic similitude of the machines [3].

Theoretically, the prototype and model should have identical ratio of forces acting between the fluid and the components of the machine. Major similitude numbers are Reynolds, Euler, Thoma, Froude and Weber number [3]. In general, it is impossible to satisfy similitude of these ratios in the same test therefore, according to IEC 60193 [3], the similitude condition which will be considered must be the one with the greatest influence on the test results.

EXPERIMENTAL FACILITY

The test rig, used to test model turbines, has a maximum pumping power of 2 MW, test heads up to 160 m and a maximum flow rate of $2 \text{ m}^3/\text{s}$. The set-up, as shown in Figure 2, is capable of performing the performance and cavitation tests of Francis type turbines, pump/turbines and prototype turbines.

The rig can be operated in three different modes. Details of the set-up are given in [8] and [9].

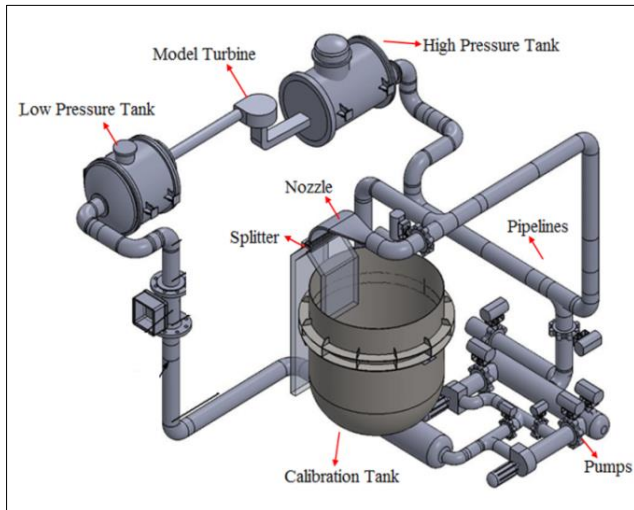


Figure 2. TOBB ETU Hydro Turbine Test Rig [9]

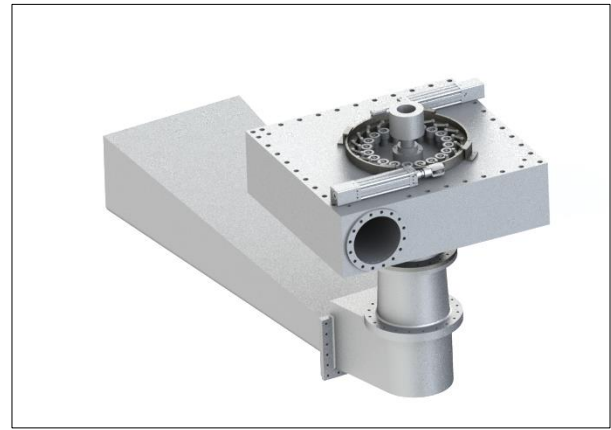


Figure 3. Solid model of the model turbine

In order to achieve an efficient similarity between prototype and model, the minimum values for model size, Reynolds number and test specific hydraulic energy necessities are defined in IEC 60193 [3]. All of the minimum criteria should be satisfied independently from each other and model size should be as large as possible, because decreasing model size means increasing the scale effects which results as a deviation from prototype observations. In Table 2, minimum criteria which are mentioned IEC 60193 [3] and model size and parameters which are calculated are given.

Table 2. Prototype and model turbine parameters for Kepez H.E.P.P.

Parameter	Francis Type Turbine	
	IEC 60193	Model Turbine(Kepez HEPP)
Reynolds Number (Re)	$>4 \times 10^6$	1.36×10^7
Specific Hydraulic Energy (J/kg)	>100	982
Reference Diameter (m)	>0.25	0.6

CFD RESULTS FOR MODEL TURBINES AND COMPARISON WITH THE PROTOTYPE ANALYSIS

Scale effects result as a deviation between the model and prototype due to the prototype parameters which are not identical with the model [10]. As Reynolds number and the diameter of the model are usually smaller than the prototype, friction losses are larger than the prototype. Due to viscous effects model efficiency is usually lower than the prototype efficiency. In [11], composition of the individual losses of Francis model machines is given. According to [11], main losses occur in the runner. Using equation 7 and 8, scalable losses from model to prototype is calculated as 0.689%.

In Figure 4, pressure contours obtained from the CFD analysis of prototype and model are given. When pressure distribution in the runner is examined it is seen that in model turbine water enters to the runner with a smaller pressure than the prototype which causes more friction losses. Also the pressure difference between the runner inlet to outlet is smaller in the model

As mentioned, model tests require the geometric, kinematics and the dynamic similitude principles implemented between model and prototype. Model dimensions should be the range of the manufacturing capacity, but also it should be larger than 0.25 m according to IEC 6193[1]. Model turbine power, head and flow rate values should not exceed the facility capabilities but should ensure equation (1), equation (2) and equation (3). In-house codes are developed based on the similarity laws and the set-up capacity, to obtain the model turbine parameters. Using these in house codes, model turbine parameters, considering both similitude equations and set-up capacity constraints, are obtained for KEPEZ H.E.P.P in Turkey, which will be tested (Table 1).

Table 1. Prototype and model turbine parameters for Kepez H.E.P.P.

	KEPEZ HEPP	
	Prototype	Model
D(mm)	1234	600
Q(m ³ /s)	6.05	1.12
H(m)	162	100
N(rpm)	750	1211.9
P(kW)	8942	1025

The prototype to model scale ratio is $\lambda = \frac{D_1}{D_2} = 2.019$. Here D_1 is the prototype runner diameter, D_2 is model diameter. Solid model of the model turbine is given in Figure 3.

turbine than the prototype which means, in the model turbine smaller angular momentum is generated in the runner. Therefore, power output is also smaller in the model turbine.

In order to observe the behavior of the flow due to the scaling effects, hill chart is obtained for the model and compared with the hill chart of the prototype using CFD cases in a series of guide vane openings (Figure 5). Model hill chart was generated by scaling the prototype turbine ($\lambda = 2.019$). For model turbine, the design point has 6.1 kg/m^3 discharge and 165 m head which correspond energy coefficient to 0.000382 energy coefficient and 0.005511 flow coefficient.

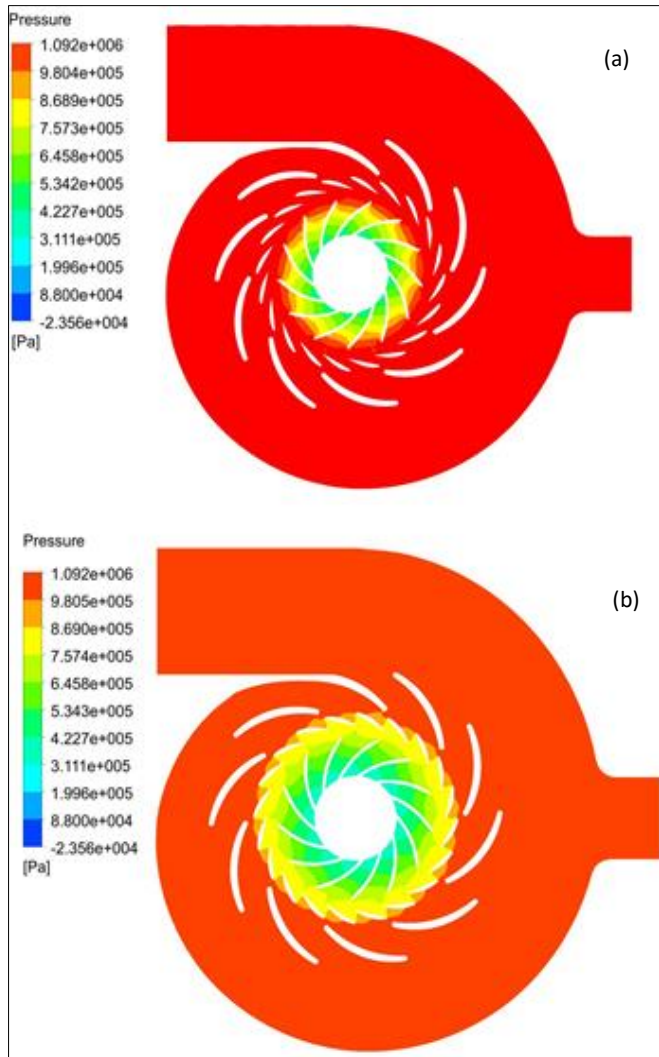


Figure 4. Pressure distribution for (a) prototype turbine (b) model turbine

Discharge and energy coefficients of the designed model turbine are plotted in the hill chart and compared with the hill chart of the prototype. The graph shows the isolines of the efficiency (η). In partial loads because of incidence losses, viscous effects, draft tube swirl structure increase, turbine efficiency drops both in scaled and real turbine. Maximum

efficiency value in the model hill chart is smaller than prototype as it is expected. Optimum design point corresponds to same energy and discharge coefficient in both situations. According to the comparison of hill charts, it is seen that scaling affects primarily the hydraulic efficiency.

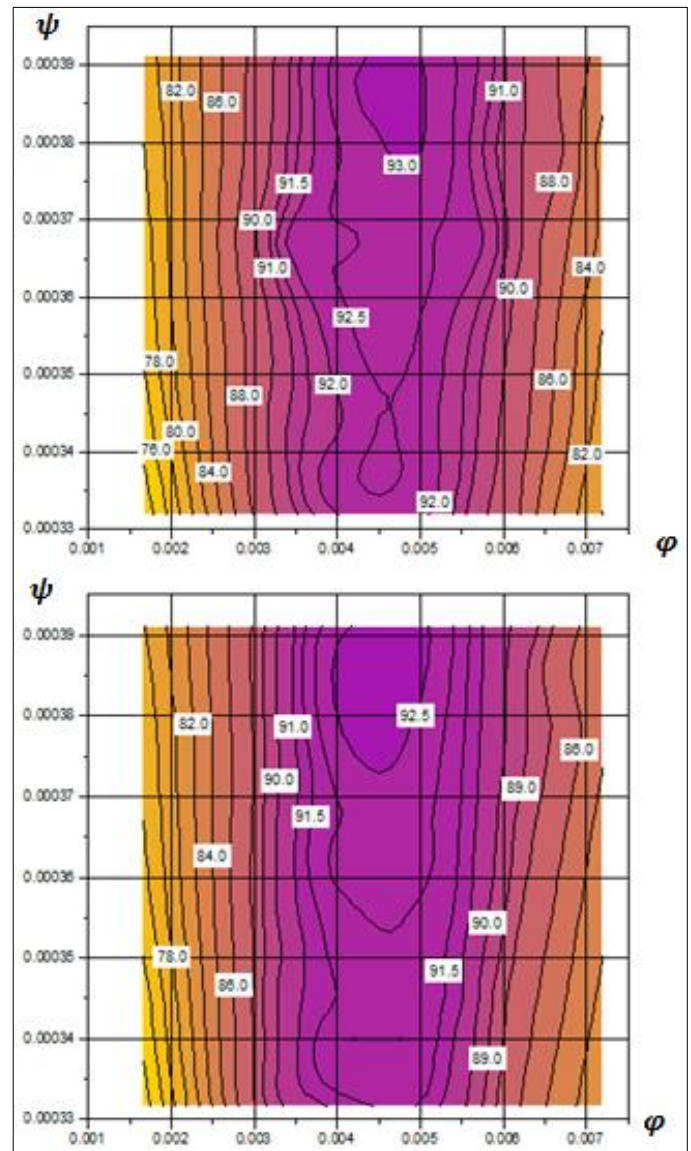


Figure 5. Hill chart for (a) prototype turbine (b) model turbine

CONCLUSION

Traditional way of designing hydroturbines is model tests. In the model tests, performance measurements such as power, efficiency, flowrate, head, temperature, and torque are carried out. Generally, most turbines are too big to be tested at model test laboratories therefore prototype is scaled down to a model turbine. At TOBB ETU Center of Hydro Energy Research, before model tests, model dimensions and parameters are obtained by in-house codes. The system developed at the center, is used for a real power plant (Kepez HEPP) in this study. According to numerical hill charts, model turbine

represents real-world prototype efficiently. According to the numerical results, passing from model to prototype can change the efficiency which is a crucial parameter.

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