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Research and Design working characteristics of orthogonal turbine

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Abstract

For the orthogonal turbine runner, the geometry parameters such as runner diameter, blade height, number of blades, D/H ratio, blade profiles... have certain effects on the ability to working situation and the performance of the turbine. Therefore to give the runner samples with good efficiency and working ability, it need to design, simulate and test many runner samples, hence we can select the most optimal one.

This paper simulate the circumstances on changing D/H ratio of the runner samples, specific to the 03 cases that are D/H = 0.9, D/H = 1 and D/H = 1.1 to evaluate turbine working situation, based on we can select the optimum working runner sample.

Keywords: ***Simulation, orthogonal turbine, hydro power***

1. *Introduction*

Blade profiles much affect on power performance of the turbine. There are many kinds of aerodynamic profiles with high quality have been studied at famous research institutions about aero-dynamic on the world. The currently most popular that are NACA profiles of National Aeronautics and Space Administration (NASA) are used for orthogonal turbine. NACA profiles have more suitable characteristics of aero-dynamics as Reynolds number, angle of attack, chord length, coefficients of lift and thrust, sliding scale, largest and smallest pressure coefficient.

In addition, the geometry parameters of runner also affect the working characteristics of orthogonal turbine. One of geometric parameters such as D/H ratio of – the ratio between diameter and blade length of the runner. The energy characteristics evaluation of the turbine can be done by experiment or can use CFD fluid dynamics calculating software to simulate the turbine working process. The article contents presents some research results on effect of D/H ratio changing on turbine energy characteristics by using simulation software ANSYS – Fluent.

2. *The mathematical formula orthogonal turbine*

For the orthogonal turbine, H is height of turbine runner, U is the flow velocity, P is the axis mechanical power. Then we have the formula for orthogonal turbine as follows:

$$C_T = \frac{T}{\frac{1}{2}\rho ARU^2} \tag{2.1}$$

$$C_P = \frac{P}{\frac{1}{2}\rho AU^3} \tag{2.2}$$

With CP and CT, respectively are torque and power coefficients, ρ is the liquid density, A is the turbine swept area, R is the radius of turbine runner. For the straight balde vertical orthogonal turbine, A is defined as follows:

$$A = D.H \tag{2.3}$$

3. *Research Methodology*

The article using Ansys-Fluent software to calculate, simulate energy exchange process between the flow and turbine runner by 3D model with the changing cases of ratio as D/H = 0.9; D/H = 1 and D/H = 1.1. The computational models studied with flow velocites on changes in the range of 1.5 m/s - 3.5 m/s, is applied to free flow. The simulation results can evaluate working characteristics, from which based to select working turbine model with best energy characteristics

4. *Problem building*

4.1. Blade profile Selection

There are many NACA sample profiles, however in the research results shown that NACA0018 profile to be best suitable for orthogonal vertical axis turbines [1].

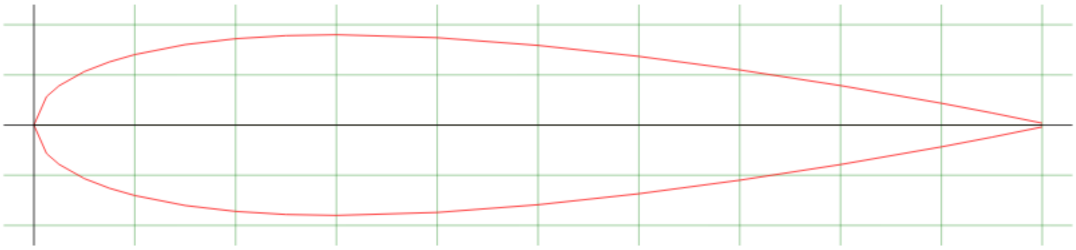


Figure 4.1. NACA 0018 profile

x	1	0,9500	0,9000	0,8000	0,7000	0,6000	0,5000	0,4000	0,3000
y	0,00189	0,01210	0,02172	0,03935	0,05496	0,06845	0,07941	0,08705	0,09003
x	0,2500	0,2000	0,1500	0,1000	0,0750	0,0500	0,0250	0,0125	0,0000
y	0,08912	0,08606	0,08018	0,07024	0,06300	0,05332	0,03922	0,02841	0,00000
x	0,0125	0,0250	0,0500	0,0750	0,1000	0,1500	0,2000	0,2500	0,3000
y	-0,02841	-0,03922	-0,05332	-0,06300	-0,07024	-0,08018	-0,08606	-0,08912	-0,09003
x	0,4000	0,5000	0,6000	0,7000	0,8000	0,9000	0,9500	1	
y	-0,08705	-0,07941	-0,06845	-0,05496	-0,03935	-0,02172	-0,01210	-0,00189	

Figure 4.2. Co-ordinates of NACA 0018 profile

4.2. Physical model building

This paper simulate the dynamic relationship between the changing cases on ratio D/H of orthogonal turbine runner to different velocities, specifically 03 cases of change that are D/H = 0.9; D/H = 1 and D/H = 1.1. From there we built the calculation physical model for 03 cases with the following parameters:

Desp \ Case	D/H = 0.9	D/H = 1	D/H = 1.1
Diameter D (mm)	1000	1000	1000
Height of blade H (mm)	1110	1000	910
Blade number Z	10	10	10
Profiles	NACA 0018	NACA 0018	NACA 0018
Chord length c (mm)	100	100	100

4.3. Calculation and simulation methods

Fluid dynamic calculation with computer support (CFD) is a useful tool to analyze flow energy characteristics. The CFD can predict properties of torque, pressure on turbine runner, from which we can predict the turbine energy characteristics. In this paper we use ANSYS - Fluent software to simulate problem.

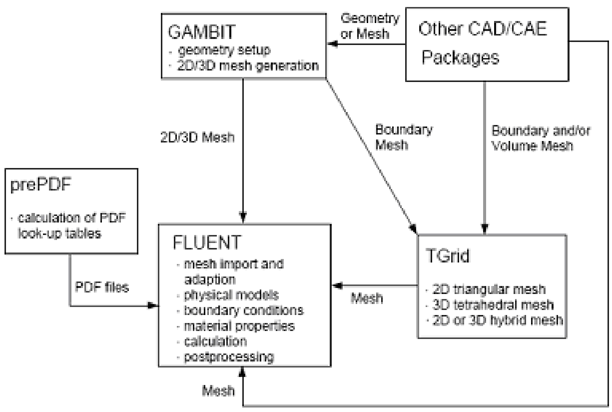


Figure 4.3. Structure of Ansys Fluent software

4.3.1. Turbulence model

There are many models of turbulence flow, including k-ω model and k-ε model is most commonly used currently. K-ω model used with the problem in boundary layer, near the wall while the more distant regions, need to use k-ε model. When using k-ε model, must use a wall function to resolve the boundary layer. This means that the mesh density near the wall should be included in order to obtain all of viscous boundary layer.

In the most cases, k-ε model give the same results as other models when use a coarser mesh density near the wall, which reduces the time to complete the calculations [2].

This paper uses k-ε turbulent flow model to simulation calculate, shown by 02 equations on kinetic energy k and ε turbulent loss rate as following [3]:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k u_t)}{\partial x_t}=\frac{\partial}{\partial x_t}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_t}\right]+G_k-\rho\varepsilon \tag{4.1}$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(\rho\varepsilon u_t)}{\partial x_t}=\frac{\partial}{\partial x_t}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\right]\frac{\partial\varepsilon}{\partial x_t}+C_1\frac{\varepsilon}{k}G_k-C_2\rho\frac{\varepsilon^2}{k} \tag{4.2}$$

With: C_1, C_2, σ_k are empirical constants, G_k is the quantity express formation of kinetic energy, it depends on the velocity gradients and turbulence viscosity.

$$G_k=\mu_t\left(\frac{\partial u_t}{\partial x_j}+\frac{\partial u_j}{\partial x_t}\right)\frac{\partial u_j}{\partial x_t} \tag{4.3}$$

Turbulent viscosity is derived from k and ε, including a constant is determined empirically by $C_\mu = 0.09$,

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \tag{4.4}$$

4.3.2. Meshing model

We mesh 03 models with same type of triangle mesh before we proceed to run the simulation. The more smoothy divided grid will obtain results more accurate simulation.

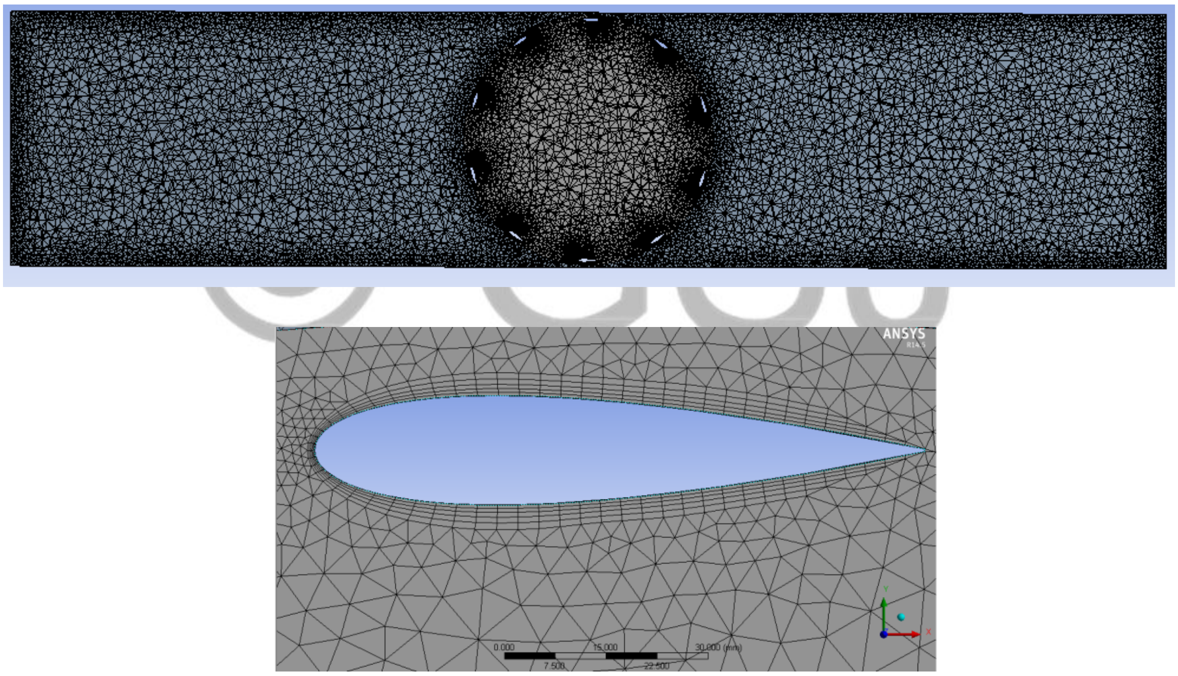


Figure 3.4. Meshing on orthogonal turbine model

After meshing, we achieved the following results:

Desp \ Cases	D/H = 0.9	D/H = 1	D/H = 1.1
The node number on meshing model	2184799	1934879	1706923
The element number on meshing model	9323392	8125898	7232985

4.3.3. Solution method

The selection of solver, viscous model, material properties, boundary conditions are carried out as follows:

Solver:

Interpolation method: Implicit; Direction: 3D; Time: Steady; Turbulence model: k-ε; Heat exchanger: None; Material: salt water with ρ=1000 kg/m3.

To set up conditions for problem calculation: Velocity, to ignore the influence of gravitational acceleration

Boundary condition:

Inle condition: inlet velocity in range 1.5 – 3.5 m/s

Blade condition: Wall

Symmetrical surfaces

Outlet condition: outlet pressure

5. ***Results and Discussion***

5.1. ***Turbine energy characteristics Comparison when changing D/H for 03 study cases***

After running the above cases simulation, we obtained the results of velocities and pressure distribution on runner and around blades.

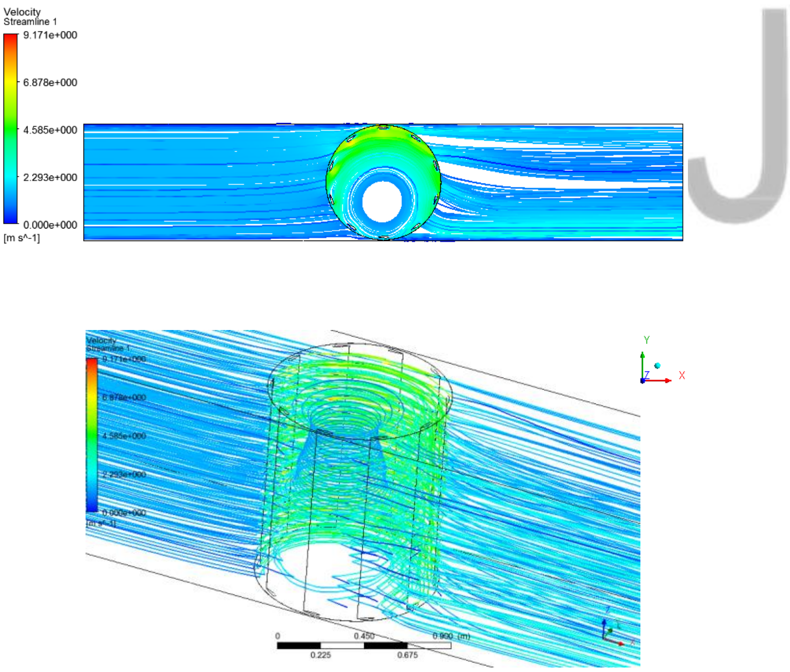
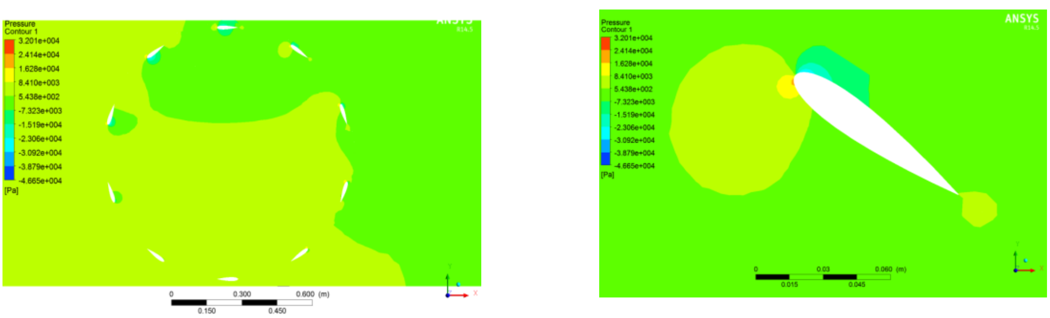


Figure 5.1. Velocity distribution, stream through runner



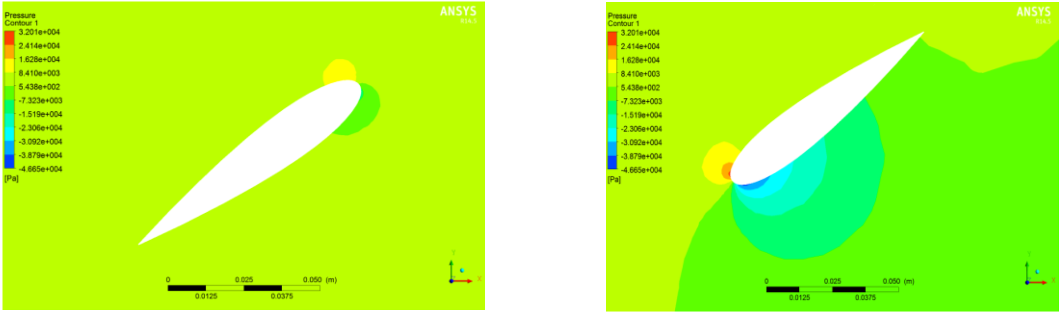


Figure 5.2. Pressure distribution around blade profile for case of D/H = 0.9

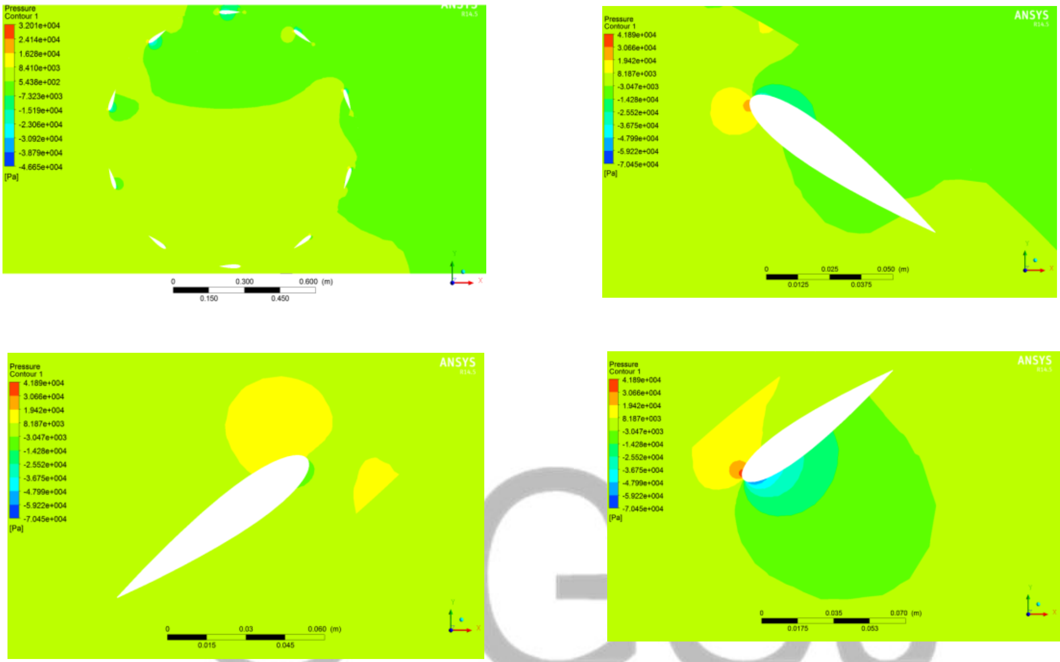


Figure 5.3. Pressure distribution around blade profile for case of D/H = 1

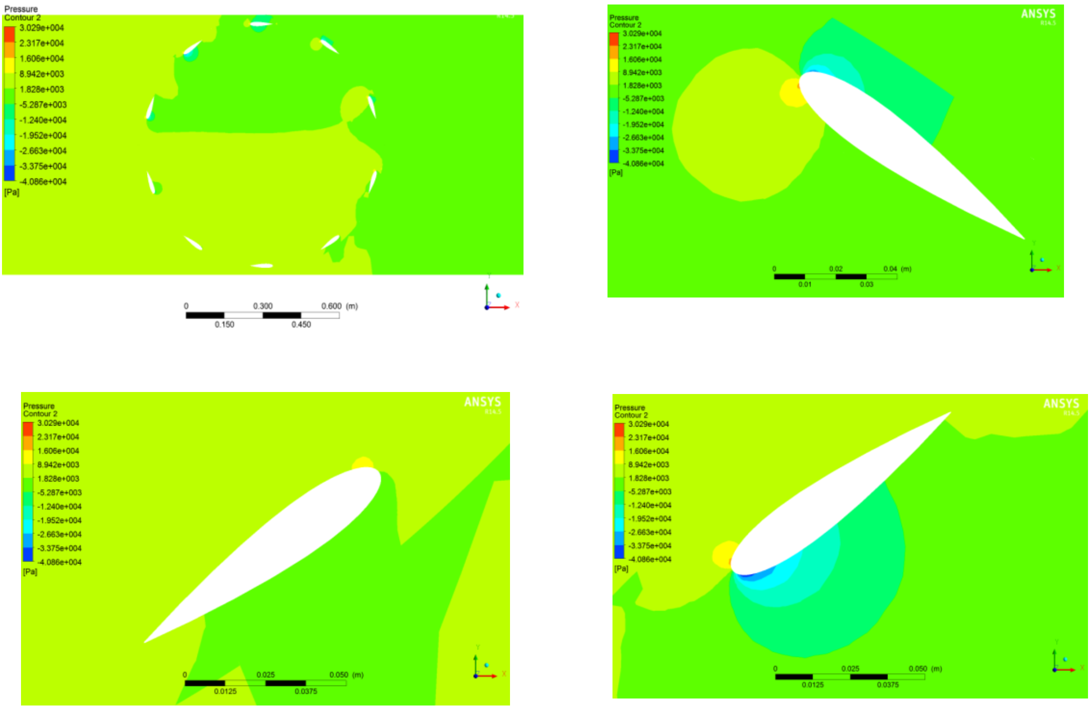


Figure 5.4. Pressure distribution around blade profile for case of D/H = 1.1

From the calculated results, we build the relation characteristic lines of turbine power, torque and efficiency with velocities for 03 studied cases.

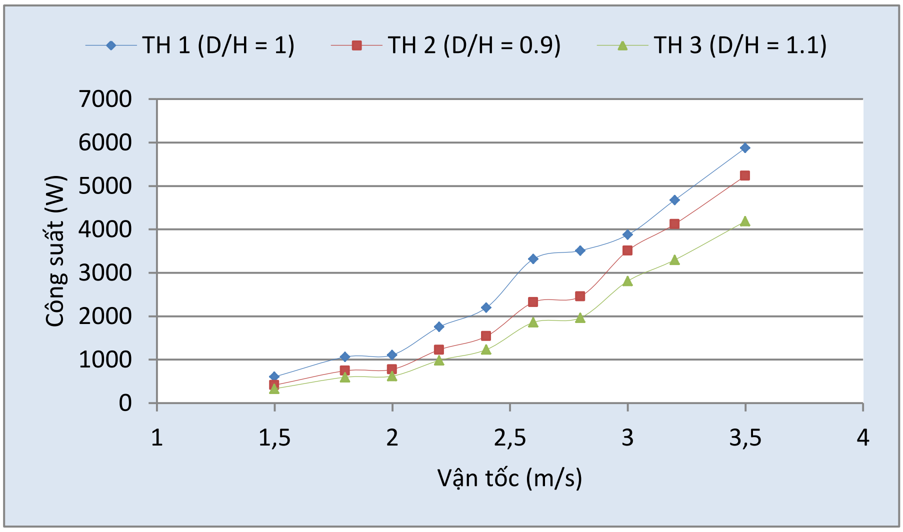


Figure 5.6. Relationship curve between capacity and velocity

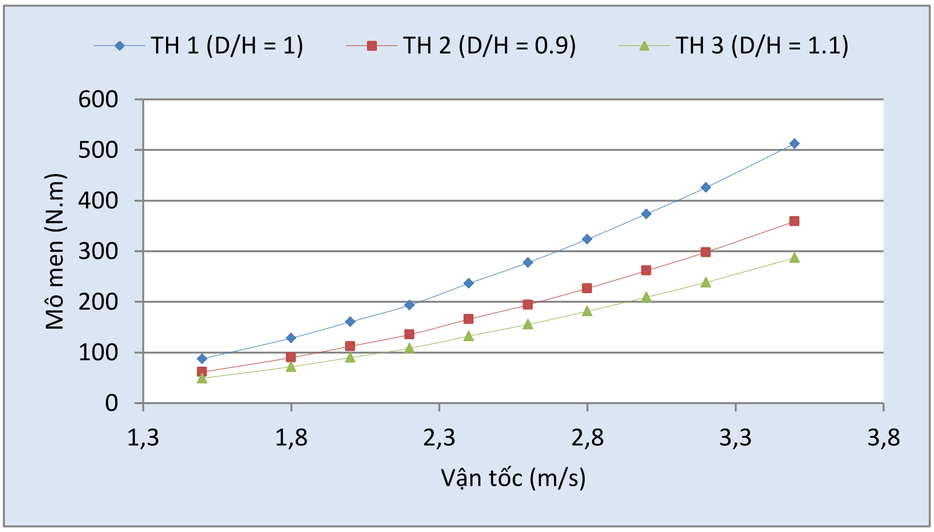


Figure 5.7. Relationship curve between moment and velocity

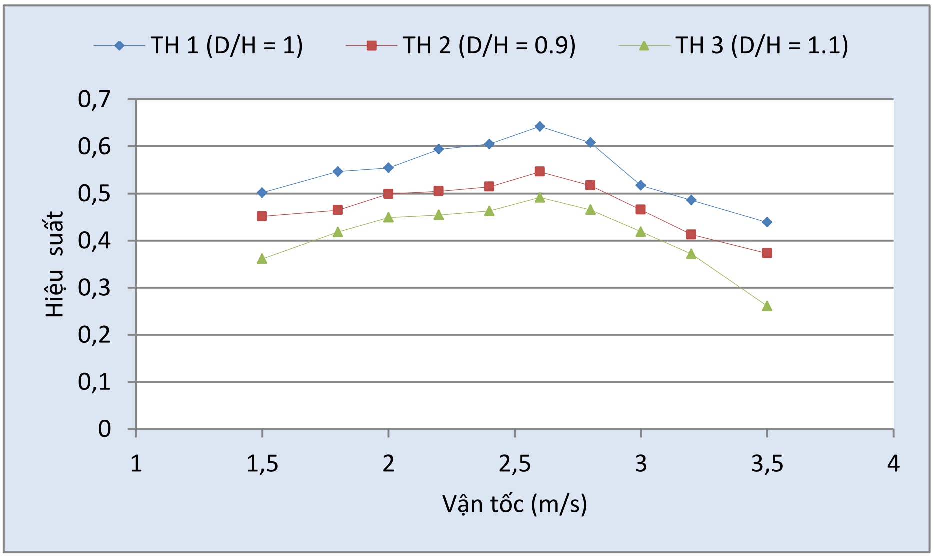


Figure 5.8. Relationship curve between efficiency and velocity

Discussion:

Based on the curve in Figure 5.6 we see when velocity increases, the turbine capacity also increases. In 03 cases, the power curve of case D/H = 1 is located on the top, next by the curve of the case D/H = 0.9 and finally the relationship curve of case D/H = 1.1. This means that, the same velocity value, the capacity value of case D/H = 1 is the largest, followed by the remaining 02 cases. So, the case D/H = 1 is maximum exploitation of flow energy more than the remaining 2 cases

Similarly, for the relationship curves as turbine torque and efficiency, we also found that the relationship curve of the case D/H = 1.1 is optimal than the remaining 02 cases.

6. Conclusion and recommendation

1) The research results on changing energy characteristics of orthogonal turbine when changing D/H ratios for the 03 cases mentioned above and preliminary study on changing velocity, pressure through runner showed that when changing the ratio D/H, pressure and velocity distribution also change, leading to energy characteristics also change.

2) Base on the calculated parameters and relationship curves between power, efficiency and velocity after simulated for 3 cases above. We see that, turbine runner with case of D/H = 1 is the optimized one more than 02 remaining cases with the same flow conditions

3) The article also provides research method by simulations for the problem models, and method of results building into relationship curves that describe the working characteristics of orthogonal turbine.

7. Refenrences

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