Journal of the Korean Society of Marine Engineering, Vol.32, No.7, pp.1044~1050, 2008(ISSN 1226-9549)

Internal Flow Analysis of a Tubular-type Small Hydroturbine by Runner Vane Angle

Sang-Hyun Nam* · You-Taek Kim† · Young-Do Choi** · Young-Ho Lee***

(Received April 2, 2008 ; Revised July 16, 2008 ; Accepted July 16, 2008)

Abstract : Most of developed countries, the consumption of fossil fuels has been serious problems that cause serious environment pollution like acid rain, global warming. Also, we have faced that limitation fossil fuels will be exhausted. Currently, small hydropower attracts attention because of its small, clean, renewable, and abundant energy resources to develop. By using a small hydropower generator of which main concept is based on using the different water pressure levels in pipe lines, energy which was initially wasted by use of a reducing valve at the end of the pipeline, is collected by turbine in the hydropower generator. A propeller shaped hydroturbine has been used in order to use this renewable pressure energy. In this study, in order to acquire basic design data of tubular type hydraulic turbine, output power, head, efficiency characteristics due to the flow coefficient are examined in detail. Tubular-turbine among small hydraulic power generation can be used at low-head. The purpose of this study is to research turbine's efficiency due to runner vane angle using CFD analysis.

Key words : Tubular-type hydroturbine, Runner vane, Velocity distribution, Pressure distribution, CFD

Nomenclature

Cp : pressure coefficien($(P_s - P_{t2})2/\rho U^2$)
Cp_p : Cp at pressure surface
Cp_s : Cp at suction surface
P : output power
P_s : blade surface static pressure
P_{t2} : runner outlet total pressure
R : variable position from leading edge to trailing edge at the blade center line
R_{LT} : length from leading edge to trailing edge at the blade center line
U : circumferential velocity at the outer tip of runner blade
β_s : stagger angle (runner vane opening)
$\triangle H$: differential head
$\triangle Cp$: differential $Cp(Cp_p - Cp_s)$
η : efficiency
ρ : density of working fluid
ϕ : flow coefficient

† Corresponding Author(Korea Maritime University, Dept. of Marine System Engineering, E-mail:kimyt@hhu.ac.kr, Tel: 051)410-4258)
* DNDE Co. Ltd.
** Industry-Academic Cooperation Foundation, Korea Maritime University
*** Division of Mechanical & Information Engineering, Korea Maritime University

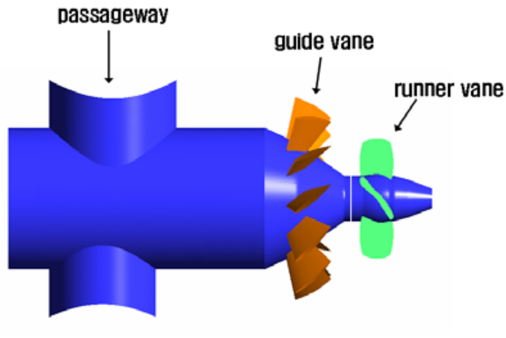


Fig. 1 Tubular-type turbine model

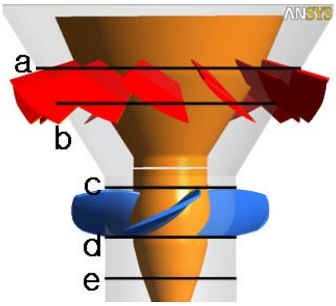


Fig. 2 Arrangement of guide vane and runner vane

1. Introduction

Mass consumption of fossil fuel such as oil and coal causes global warming and the environmental problems due to acid rain. Because natural resources are limited, a serious problem of the exhaustion of fossil fuel is expected in the near future around the world. As a countermeasure, alternative energy such as solar thermal, biomass, wind power, small hydraulic power, geothermal power, and marine energy is being drawn to attention. The small hydraulic power generations have power output of small(1,000kW~10,000kW), mini (100kW~ 1,000kW), and micro(less than 100kW) classes. Micro hydraulic power generation of which the output is less or equal to a 100kW is attracting considerable attention.

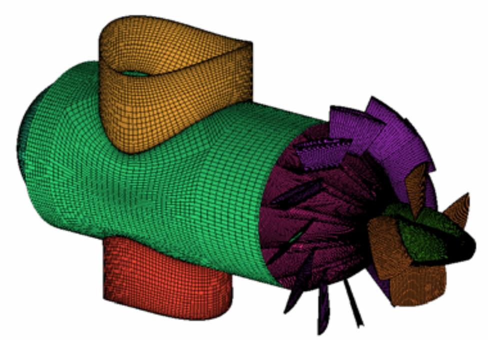


Fig. 3 Computational domain of turbine model

This is because of its small, simple, renewable, and abundant energy resources. By using a small hydropower generator of which main concept is based on using the different water pressure levels in pipe lines, energy which was initially wasted by use of a reducing valve at the end of the pipeline, is collected by turbine in the hydropower generator. A propeller shaped hydro turbine has been used in order to use this renewable pressure energy. Previous studies by researchers for the propeller-type turbine have been tried to determine the optimum configuration of the turbine by experimental and numerical methods. For example, Ales, S. et al[1]. had tried to researched performance for change of blade geometry., Arno, G. et al.[2] had demonstrated optimization of numerical algorithm.

In this study, in order to acquire basic design data of tubular type hydraulic turbine[3], output power, head, efficiency characteristics due to the flow coefficient are examined in detail. Moreover the relationship not only between the internal flow and performance, but also between pressure and velocity distributions with the variations of flow coefficient and runner vane opening angle are investigated by using a commercial CFD code.

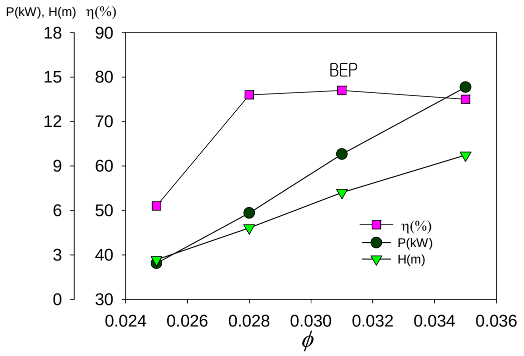


Fig. 4 Performance characteristics curves of turbine model by flow coefficient

2. Modeling and numerical analysis

Tubular-type hydro turbine using a differential pressure in water pipe line is shown in Fig. 1. Tubular total length from inlet to outlet is 1.5m. Output power generated by runner is transferred to a generator, which is installed outside the turbine, by use of belt pulley. The runner shape of the tubular-type hydro turbine is shown in Fig. 2. There are 4 movable runner blades and 12 fixed guide vanes. Outer diameter of the runner blade is 0.197m.

The total number of nodes for casing, guide vane and runner vane is 2,271,351 as shown in Fig. 3. The CFD grid for tubular-type hydroturbine had been constituted hexahedral that consider y+, non-dimensional distance from the wall. A commercial code of CFX-10 is adopted to conduct CFD simulation to solve incompressible turbulent flow. In order to solve accurately the viscous sublayer, SST(Shear Stress Transport) is applied as a turbulent model. SST turbulence model and constant pressure at inlet and averaged outflow at outlet are used boundary conditions. All the calculations for the test cases by the variations of runner blade opening are conducted under the condition of steady state. Flow coefficient as shown in Fig.4, is adopted to calculate the internal flow of the turbine model. Four movable runner blade angle, which are defined as Case A, B, BEP, and, C are used for the simulation of internal flow of the turbine model.

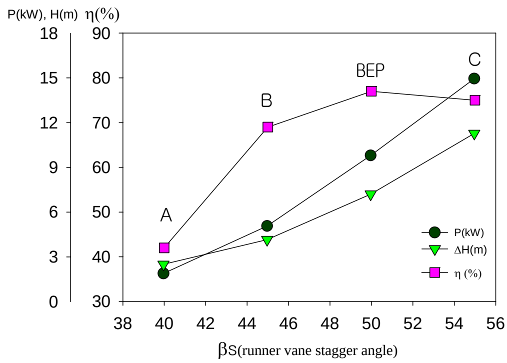


Fig. 5 Performance characteristics curves of turbine model due to runner vane angle

3. Results and discussion

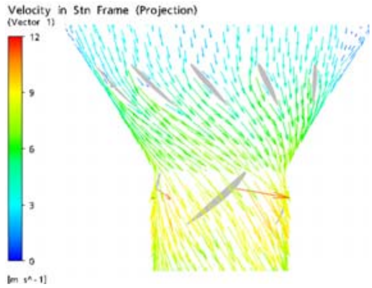
Figure 4 shows performance characteristics of turbine model by flow coefficient in the case of runner vane stagger angle, β_s=50° at runner hub. The best efficiency(C point), η_{max} is 0.76 at ϕ=0.031. At that point P=8.2kW, differential head, $\triangle H$=6.1m.

Figure 5 shows turbine model characteristics by different stagger angle of runner blade in case of flow coefficient of ϕ=0.031. In this figure, Case BEP point means a BEP point in Fig. 4. It can be seen that the best efficiency point occurs at β_s=50°.

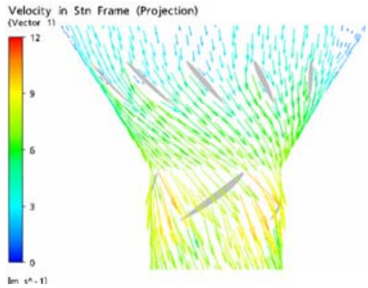
Figure 6 shows the absolute velocity vectors on mid section(in Fig. 2) at Case A, B, BEP and C points of Fig. 5, respectively. Figure 7 shows circumstantially the velocity vectors between the runner blades. At BEP point, it is considered that the good efficiency was caused by the straight flow without rotating after passing through the runner vane. On the other hand, as β_s becomes larger, it can be seen that the counterclockwise rotating component are gradually increase. On the contrary, as β_s becomes smaller, clockwise rotating component is occurred.

Figure 8 shows the pressure distributions around the runner vane on mid section(in Fig. 2) at Case A, B, BEP, and C points of Fig. 5. Also, Fig. 9 shows the pressure coefficient, Cp, distributions along the upper and lower blade center line. As β_s decrease, $\triangle Cp$ becomes larger. $\triangle Cp$ has a good value near the blade leading edge at all of the stagger angle, however, reverse pressure gradient occurs at β_s=40°. In addition, maximum pressure occurs at front of leading edge and minimum pressure occurs 17% from the leading edge at the suction side.

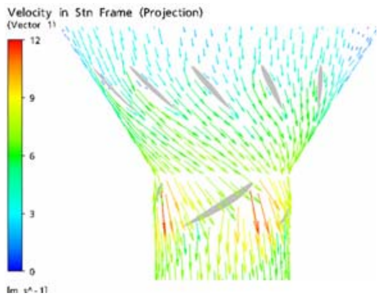
Figure 10 shows tangential velocity distributions according to flow direction from guide vane inlet(a)to runner vane exit region(e), respectively. At the inlet region of the guide vane(a), the fluctuations of tangential velocity distributions are high, and at the inlet



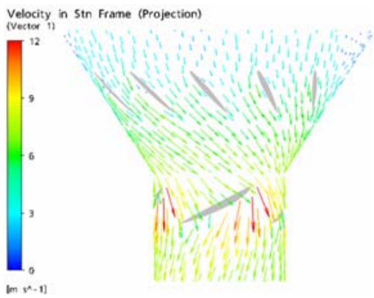
(a) β_s=40° (Case A)



(b) β_s=45° (Case B)



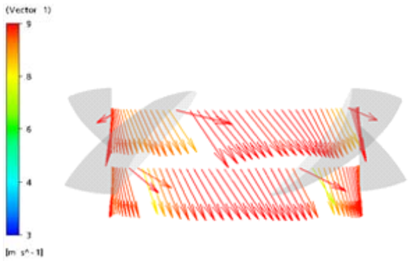
(c) β_s=50° (Case BEP)



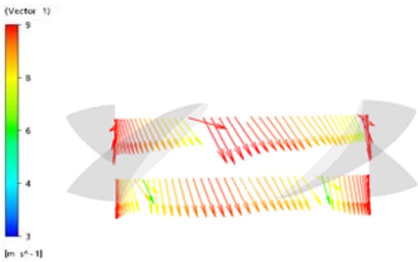
(d) β_s=55° (Case C)

Fig. 6 Velocity vectors of tubular-type hydro turbine by stagger angle of runner blade

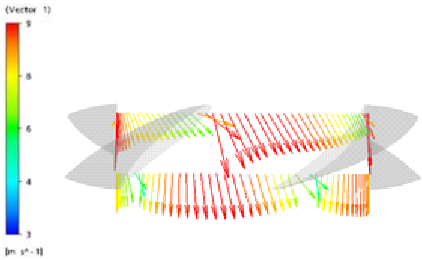
region of runner vane(c),the tangential velocity distributions have their maximum values. we can see the guide vanes have little effect on velocity distributions at inlet of the runner vane, both tangential velocity distributions.



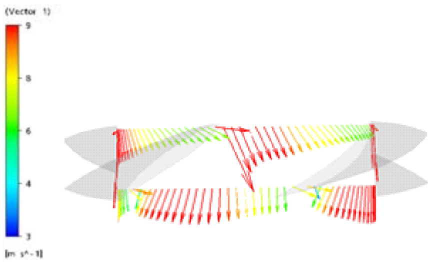
(a) β_s=40° (Case A)



(b) β_s=45° (Case B)

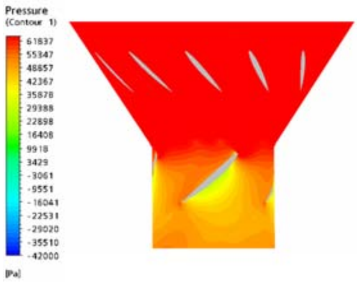


(c) β_s=50° (Case BEP)

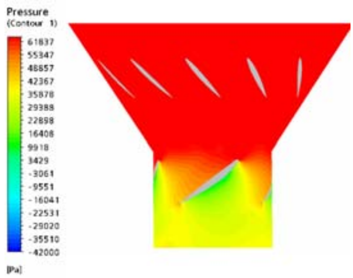


(d) β_s=55° (Case C)

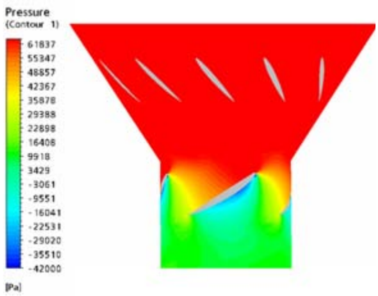
Fig. 7 Circumstantial velocity vectors by stagger angle of runner blade



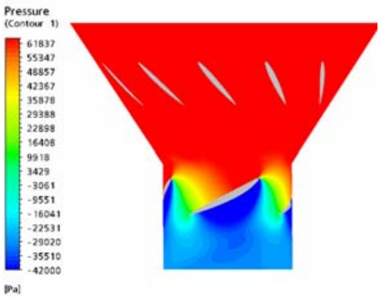
(a) β_s=40° (Case A)



(b) β_s=45° (Case B)



(c) β_s=50° (Case BEP)



(d) β_s=55° (Case C)

Fig. 8 Pressure distributions around the runner vane by stagger angle of runner blade

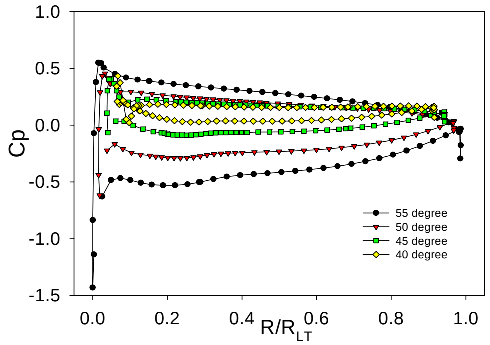


Fig. 9 Pressure coefficient distributions along the upper and lower blade center line by stagger angle of runner blade

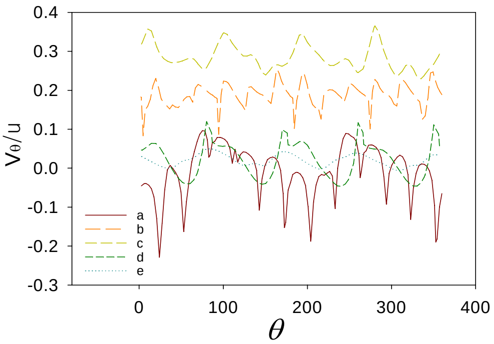


Fig. 10 Velocity distributions according to flow direction(from a to e)

4. Conclusion

In order to acquire basic design data, we carried out the CFD analysis for tubular-type hydroturbine due to flow coefficient and runner vane opening angle.

(1) The best efficiency occurs at ϕ =0.031 and β_s=50°.

(2) The efficiency is affected by rotating components at a runner outlet region.

(3) The flow coefficient and β_s becomes smaller, the clockwise rotating flow occurs, on the contrary, the flow coefficient and β_s becomes larger, the counter clockwise rotating flow appears.

(4) As flow coefficient increase and β_s decrease, $\triangle Cp$ becomes larger.

5. Acknowledgments

The authors express their appreciation to Korea Energy Management Corporation (KEMCO) and Ministry of Commerce, Industry and Energy for their grant of research and development project of new & renewable energy Technology. (Project No. 2006-N-SH02-P-02-0)

6. References

[1] Ales, S., and Jiri, O., 2006, "Shape Optimization of a Kaplan Turbine Blade," 23rd IAHR Symposium on Hydraulic Machinery and Systems Yokohama, Japan, October

[2] Arno, G., Robin, S., and Domink, S., 2006, "Kaplan Turbine Runner Optimization by Numerical Flow Simulation (CFD) and Evolutionary Algorithm," 23rd IAHR Symposium on Hydraulic Machinery and Systems Yokohama, Japan, October

[3] S. H. Nam, Y. T. Kim, Y. D. Choi, C. D. Nam, Y. H. Lee, 2007, "Influence of the Number of Blades on Small Hydroturbine," The Korean Society of Marine Engineering pp.211-212

Author Profile

You-Taek Kim

Sang-Hyun Nam

He received his M. Eng. in 2008 from Korea Maritime University, Korea. He is currently an engineer in DNDE Co. Ltd., Korea. His research interests include CFD analysis on the cavitation of pump, hydraulic turbine and tidal current turbine.

He received his B.Eng. and M. Eng. degrees from Korea Maritime University, and his Dr. Eng. from Kyushu Institute of Technology, Japan. He is currently an associate professor in the dept. of Marine System Engineering at Korea Maritime University, Korea. His research interests include new & renewable energy, fluid machinery, two-phase flow, cavitation, and CFD.

Young-Do Choi

He received his B. Eng. and M. Eng. degrees from Korea Maritime University, and his Dr. Eng. from Yokohama National University, Japan. He is currently a researcher in the Industry-Academic Cooperation Foundation at Korea Maritime University, Korea. His research interests include new & renewable energy, fluid machinery, PIV, CFD and flow visualization.

Young-Ho Lee

He received his B. Eng. and M. Eng. degrees from Korea Maritime University, and his Dr. Eng. from the University of Tokyo, Japan. He is currently a professor in the Division of Mechanical and Information Engineering at Korea Maritime University, Korea. His research interests include new & renewable energy (ocean, wind power and small hydropower energies), PIV & CFD, optimum design of pump and environmental fluid machinery.