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2-D & 3-D Calculations for the Effect of Guide Vane of Impulse Turbine

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ABSTRACT: This paper deals with the performance analysis of impulse turbine for OWC type wave energy conversion device. Numerical analysis was performed using a commercially-available software FLUENT. This parametric study includes the variation of the setting angle of guide vane. Since parametric study at various flow coefficients requires tremendous amounts of computing time, two-dimensional cascade flow approximation was employed to find out optimum principal particulars in rather simple manner. Full three-dimensional calculation was also performed for several cases to confirm the validity of two-dimensional approach. Results were compared to other experimental data, for instance Setoguchi et al (2001)'s extensive set of data, and found to be well demonstrating the usefulness of 2-D analysis. Advantages and disadvantages of each method were also evaluated.

1. Introduction

Because of its simple and efficient operation, a Wells turbine combined with OWC (Oscillating Water Column) has been widely applied for an ocean-wave energy absorption for almost 20 years. It's a self-rectifying turbine, providing a uni-directional torque in a reciprocating airflow inside a duct without any stator or valve to control the airflow direction. A little late, but almost in the same time, an impulse turbine has been developed and tested by several researchers. While each turbine has obvious advantages and drawbacks, an impulse turbine is getting supporters more and more, mainly because of its wide operating range of flow rates, low rotor speed, good self-starting characteristics and so on. For more information, see Setoguchi et al. (2001) for impulse turbine and Raghunathan et al. (1991) for Wells turbine. In fact, Setoguchi group have devoted to these studies for nearly 20 year since early 80th using Wells as well as impulse turbines, whose group led this area by finding many valuable information. Recently Thakker group also joined the people in wave energy conversion area by publishing a couple of key papers with numerical and also experimental methods. (for example, see Thakker et al., 2003a and 2003b)

The present study has been carrying out as a part of on-going project on the development of prototype OWC-Impulse turbine system commenced last year at Korea Research Institute of Ships and Ocean Engineering in Korea. In this paper the performance of impulse turbine at various design parameters as well as flow conditions was investigated using a commercially-avaliable software FLUENT. It consists of two parts in general; one being 3-D analysis to see the effects of flow coefficient, guide vane and to show the accuracy and usefulness of numerical method, and the other being 2-D method to see the possibility of utilizing very simple and quick calculation capability of 2-D analysis if it successfully provides dependability, at least qualitatively. To see the effectiveness of 2-D analysis, the variation of setting angle of guide vane was examined and compared with the available experimental data (Setoguchi et al., 2001) and also with 3-D calculation. It was found that the 2-D method provides more than enough and accurate information in selecting the design parameters, and consequently can be successfully utilized to predict the performance of impulse turbine in most cases, only with few exceptions.

2. TURBINE GEOMETRY, NUMERICAL METHOD AND TEST CONDITIONS

Impulse turbine rotor with the diameter of 38cm was designed for the present purpose and manufactured for experimental validation. It was designed very similar to that of Setoguchi et al. (2001). Number of rotor blades and guide vanes, and hub ratio are 30, 26 and 0.7, respectively. Blade axial chord and span are 6.84cm and 5.7cm. Figure 1 shows turbine geometry and some dimensions in brief. More details can be found in Hong et al (2003) and the present paper will only introduce the numerical side of study.

A commercial CFD code, FLUENT 5.4.8, was used in the present numerical analysis. The continuity and momentum equations were discretized by finite volume method, and solved by SIMPLE algorithm. Relative moving reference frame was adopted for the rotor blade. Axial reciprocating flow was considered as a constant axial velocity for numerical approach using steady analysis. In OWC-impulse turbine system, axial flow introduced into the duct with turbine rotor blades is generated by wave action inside OWC, and the frequency of wave is directly depending on wave period. Since its frequency is considerably low compared to harmonics of rotational blade-to-blade flow of turbine, the unsteadiness imposed by wave action could be considered by quasi-steady or steady analysis (Hyun et al, 1993). Although the size of turbine and the magnitude of axial velocity components were set to be 38cm in diameter and 15m/s for standard case, those values were not strictly controlled for calculation, since the effect of Reynolds number was found to be relatively minor based on the results of preliminary analysis.

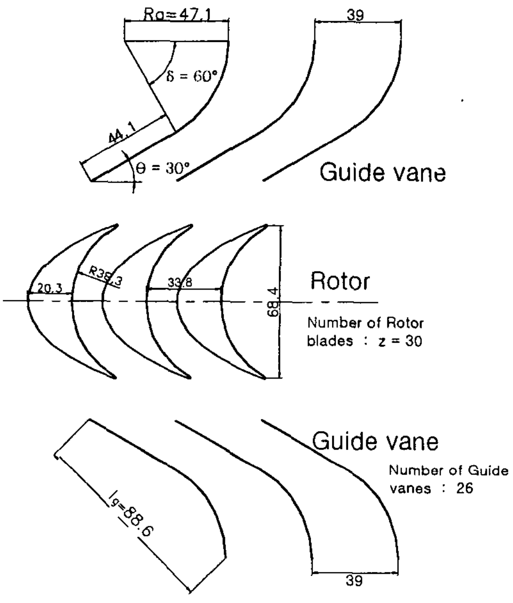
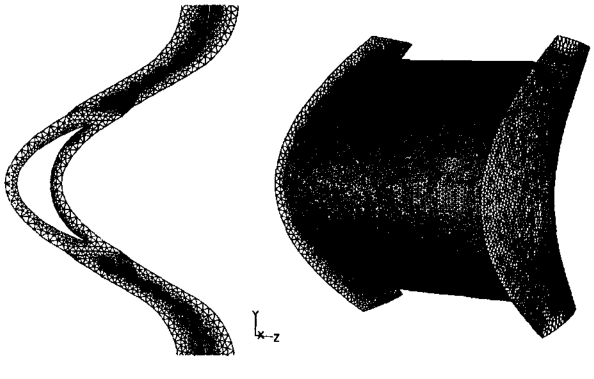


Fig. 1 Turbine Geometry

GAMBIT 1.3.0 was used for grid generation. Unstructured grid system was adopted; a triangle mesh for 2-D calculation and a tetrahedral mesh for 3-D calculation. Figure 2 depicts the example of meshes generated in 2-D and 3-D sense, respectively. For 2-D calculation the rotating motion of rotor blade was transformed into the transnational motion of 2-D blade in expanded rotation direction of infinite diameter, just like the horizontal direction in a cascade flow of Fig. 1. In order to examine the grid dependency on numerical accuracy, the number of meshes were varied from 1,000 to 10,000 for 2-D calculation to find out the optimum number of meshes of 3,600. For 3-D case, the number of meshes was decided by similar examination to choose the number of meshes of 190,000. This number for 3-D calculation is over 50 times more grids than for 2-D case, which is directly related to the computing time. In fact, it usually takes 30 seconds for one test condition in 2-D case, while it takes 30 minutes for 3-D case. It means a half day is more than enough with 2-D method to complete the performance curve of an impulse turbine, very efficient and convenient way for parametric study in preliminary design stage, provided the accuracy of numerical result is guaranteed mathematically and physically. The advantages and disadvantages of 2-D and 3-D calculations will be discussed later in details, and, in fact, this may be the major contribution of the present paper.

Although the numbers of rotor blade and guide vane differ from each other (30 and 26 respectively), it was assumed the number of guide vane to be 30 in order to apply the periodic boundary condition easily and to eventually save considerable amounts of computing time, this effect being certainly negligible. Several kinds of boundary conditions were applied; no-slip condition for blade and guide vane surfaces, uniform velocity inlet and pressure outlet boundary conditions, periodic boundary condition on boundary of a set of one blade and one guide vane, and relative velocities on the rotor. The range of blade Reynolds number varied from $5x10^4$ ~ $3x10^5$, where Reynolds number was defined based on blade chord and resultant flow velocity (vector sum of inflow and rotation speed of rotor blade). Axial velocity at inlet section was set to be 15m/s in most of cases. While flow was assumed to be either laminar or turbulent based on Reynolds number, the results showed a little difference regardless of flow condition. The k-e model was employed for turbulent flow. As already discussed previously only steady calculation was made.



(a) 2-D Grid (b) 3-D Grid

Fig. 2 Grid Generations

Performance of the impulse turbine in steady flow condition is expressed in terms of input coefficient C_A and torque coefficient C_T as follows

$$C_A = \frac{\Delta pQ}{\frac{1}{2}\rho_a(v_a^2 + U_R^2)bl_r z v_a}$$

$$C_T = \frac{T}{\frac{1}{2}\rho_a(v_a^2 + U_R^2)bl_r z r_m}$$

Here Δp, Q, T represent pressure drop, flow rate and torque, and v_a, U_R, b, l_r, z, r_m are axial mean velocity, rotational velocity of rotor blade at $r = r_m$, blade span, chord, number of blade and radius at mid-span respectively. The efficiency of turbine η and the flow coefficient ϕ can be expressed as,

$$\eta = \frac{T_\omega}{\Delta pQ_d} = \frac{C_T}{C_A \phi}$$

$$\phi = v_a / U_R$$

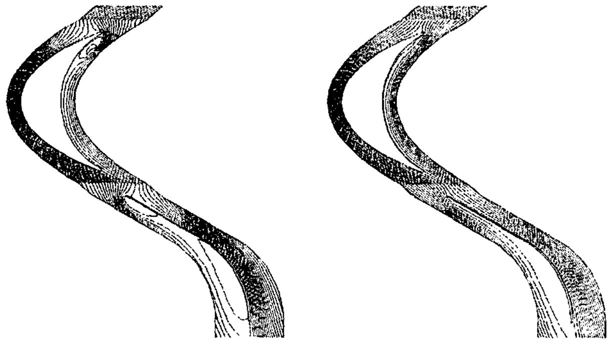
where flow coefficient ϕ has a physically equivalent meaning with the angle of attack in wing theory.

3. RESULTS AND DISCUSS

2.1 Comparison Between 2-D and 3-D Analyses

To demonstrate the accuracy and general tendency of 2-D and 3-D calculations, calculated results of FLUENT were reprocessed to obtain input coefficient C_A, torque coefficient C_T, and turbine efficiency η. That is, C_A can be obtained from calculated pressure and velocities at inlet and outlet sections. Torque can be calculated by integrating the pressure and frictional stresses on blade surfaces into rotational direction. The 2-D calculation was made at mid-span of rotor blade r_m. Tip clearance of 1mm was assumed in 3-D calculation because 1mm allowance is practically the most common situation.

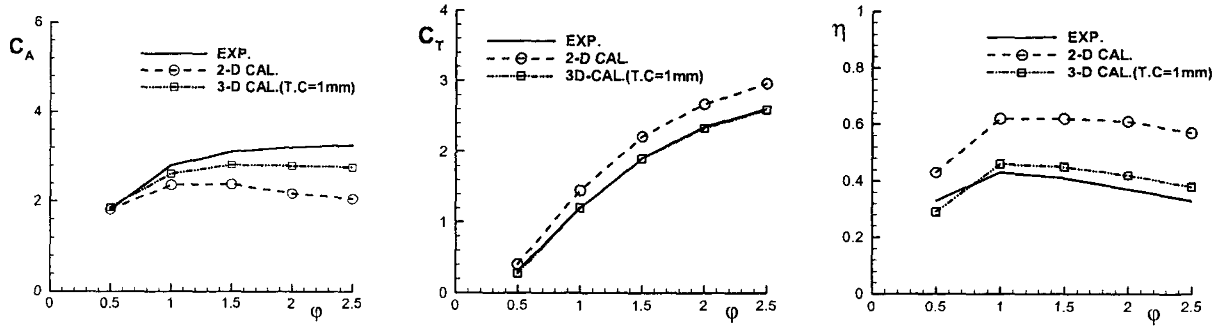
Fig. 3 shows the computed results together with Setoguchi et al. (2001)'s experimental data at various flow coefficients. Calculated results generally provide lower pressure drop (lower C_A) and higher torque (i.e. higher C_T) although 3-D calculation gives better agreements with experimental data. Especially C_T in 3-D calculation is surprisingly well coincident with the experimental result. Discrepancies between calculation and experiment increases with increasing flow coefficient, supposedly due to the three-dimensionality of flow field whose effect more prominent on input coefficient. It will be discussed later when we discuss on the effect of tip clearance. The present results were considered satisfactory, and even the 2-D result shows qualitatively same trend with experimental data even though its quantity is rather different from experiment.



(a) ϕ = 0.5 (b) ϕ = 1.0

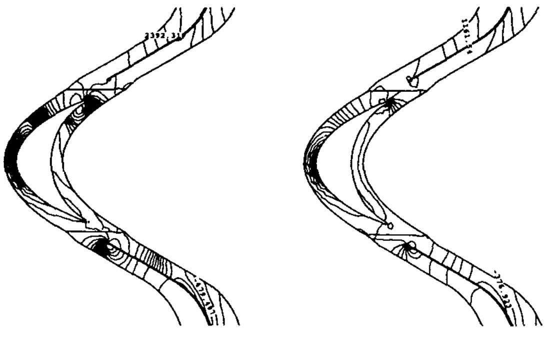
Fig. 4 Comparison of Streamlines

More detailed flow field can be found in Fig. 4 and Fig. 5. In Fig. 4, streamlines at mid-span of rotor blade were shown for ϕ = 0.5 and ϕ=1 to clearly demonstrate the differences. Flow separations are shown near the leading edge of pressure side of rotor blade and at left side of outlet guide vane when ϕ=0.5. As a result, more unnecessary pressure drop and less torque are expected. On the other hand for ϕ=1 flow pattern looks much more favorable, and weaker separation only occurs at left side of outlet guide vane, yielding better efficiency than for ϕ=0.5.



(a) Input Coefficients (b) Torque Coefficients (c) Turbine Efficiency

Fig. 3 Comparison at Standard Condition Calculations



(a) ϕ = 0.5 (b) ϕ = 1.0

Fig. 5 Comparison of Pressure Distributions

Because impulse turbine is operated in reciprocation flow condition, this separation is almost unavoidable unless self-controlled guide vane is adopted. The pressure distributions at mid-span of rotor blade were shown in Fig. 5, where more contour lines imply steeper pressure gradient. For ϕ = 0.5, more pressure drop in axial direction (vertical direction in this figure) is expected from the figure and unfavorable flow situation is also anticipated from the steeper pressure variations near leading edge of blade and outlet guide vane. These are the reasons why this impulse turbine becomes more effective at ϕ = 1.

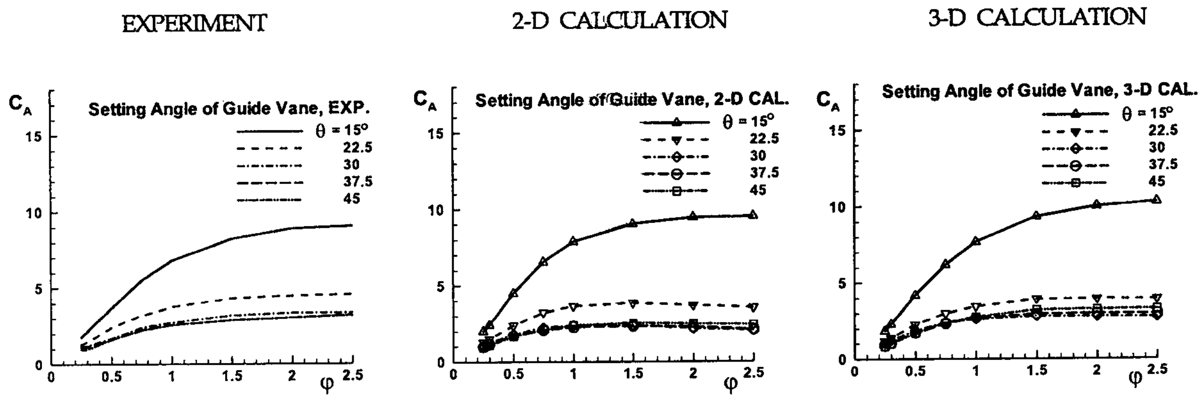
2.2 Effect of Guide Vane (Effectiveness of 2-D Analysis)

Now we see the dependability and practicality of 2-D analysis by examining the effect of setting angle θ of guide vane. Five different setting angles were chosen following Setoguchi et al (2001); 15, 22.5, 30, 37.5 and 45 degrees. Input coefficient, torque coefficient and efficiency are plotted in Fig. 9. The results of Setoguchi (2001) are also plotted as a solid line. It is shown from experimental data that the value of C_A is largest at θ = 15 degrees, and decreasing sharply with increasing θ and almost invariant after 30 degrees. The value of C_T is also largest at θ = 15 degrees and decreases gradually with increasing q. On the other hand turbine efficiency gave somewhat interesting result, the optimum case being θ = 30 degrees. It was followed by 37.5, 22.5, 45 in sequence and the worst case turned out to be θ = 15 degrees. (However, at lower θ, the turbine efficiency in case of 22.5 degrees is better than in 37.5 degrees.) These sequences can be predicted by both 2-D and 3-D analyses surprisingly well. Only difference is that absolute values are overestimated consistently in 2-D calculation, for which we already accepted those discrepancies. Order between 22.5 degrees and 37.5 degrees is reversed in the 3-D calculation although difference is relatively small. Since 3-D calculation requires at least 50 times more computing time than 2-D case and moreover a lot of additional time for grid generation, 2-D analysis can be a perfect answer when we want to perform parametric study in preliminary design stage requiring tremendous amounts of computing efforts. Of course we should use 3-D method if flow three-dimensionality is expected to play an important role, a typical example being the effect of tip clearance. Also 3-D analysis must be utilized for the performance analysis in final stage of turbine.

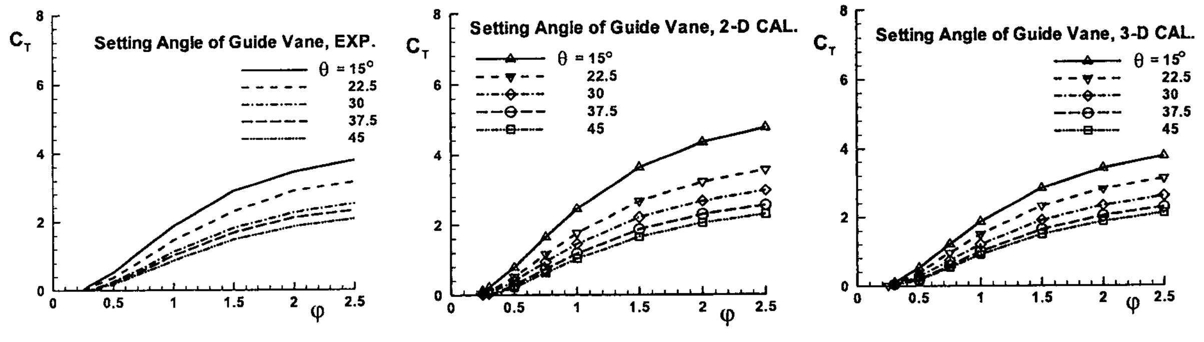
In order to interpret the flow physics embedded in Fig. 6, pressure distributions at mid-span and streamline patterns are shown in Figs. 7 and 8 in case of ϕ = 1, where only three setting angles of 15, 30 and 45 degrees are shown for brevity. At first for θ = 15 degrees, more pressure drop in axial direction is expected from the fact of obvious trend expressed by the higher pressure gradient in axial direction. Flow approaching the rotor blade was increased and the flow is deflected more in circumferential direction. This steep change of flow direction both inlet and outlet passages creates the acceleration of flow in front of rotor blade (aft-part of inlet guide vane) as well as through downstream guide vane. It could reduce the higher torque, but very unfavorable flow pattern at outlet was unavoidable. Guide vane at outlet passage acts like blockages, the main reason of high pressure loss. In fact the loss by pressure loss produced due to this separated downstream flow far exceeds the gain of torque produced by the acceleration of flow in rotor blade. Overshoot of flow direction at outlet was also observed although it is not clearly shown here

Situation becomes a little better, but opposite in nature for θ = 45 degrees, where flow is not effectively accelerated. Smooth transition of flow into rotor blade was not effectively achieved either, indicated by a small separation region on the pressure side near leading edge of blade. Flow at outlet was pretty smooth though because guide vane at outlet didn't block the flow passage. As a result, the least amount of torque, i.e. least C_T, was produced by rotor, even though the serious pressure loss at outlet was avoided. Again the loss in torque by ineffectiveness of inlet guide vane exceeds the gain of avoiding additional pressure loss by downstream guide vane. While either 15 or 45 degrees has some advantages as well as disadvantages, the most adequate flow field could be achieved by setting θ = 30 degrees. Flow was moderately accelerated and aligned good enough to increase flow velocity in front of blade and avoid flow separation on rotor blade. Also it could minimize the effect of pressure loss produced by unavoidable separation on outlet guide vane.

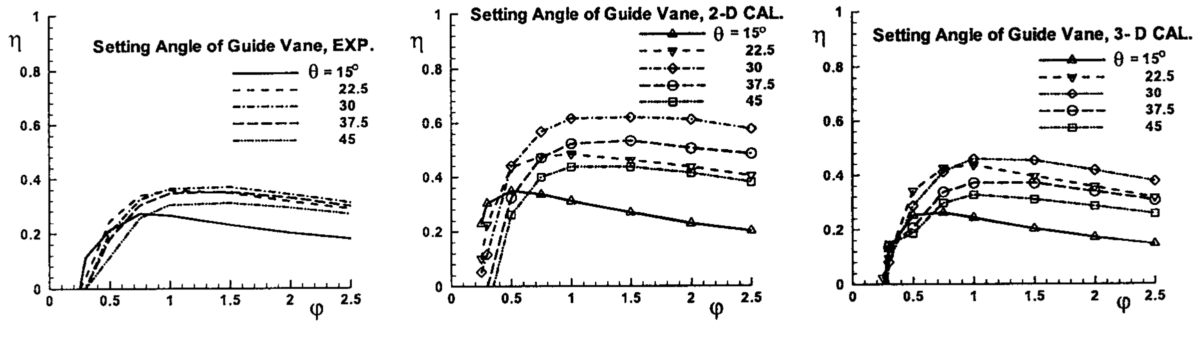
In summary the simple 2-D analysis was found to be very successful in estimating the effect of guide vane, and evaluated that it can be widely used for the parametric study in qualitative sense. No more result is provided here due to the space limitation, but in fact parametric study to see the effects of the number of rotor blade and the hub ratio has been successfully carrying out using 2-D method. Hopefully we'll have a chance to present those results later.



(a) Input Coefficients

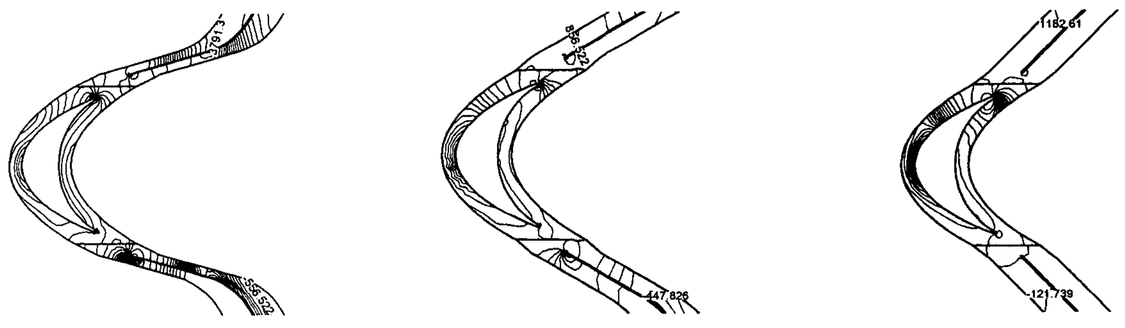


(b) Torque Coefficients



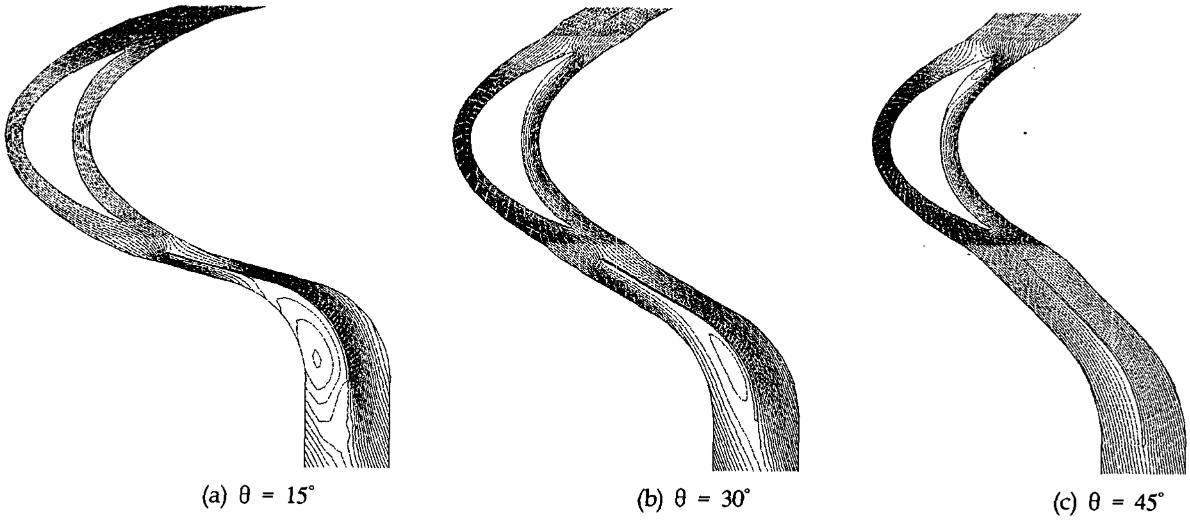
(c) Turbine Efficiency

Fig. 6 Comparison to See the Effect of Setting Angle of Guide Vane



(a) θ = 15° (b) θ = 30° (c) θ = 45°

Fig. 7 Pressure Distributions at Several Setting Angles of Guide Vane



(a) θ = 15° (b) θ = 30° (c) θ = 45°

Fig. 8 Steamlines at Several Angles of Guide Vane

4. SUMMARY AND CONCLUSIONS

Numerical approach was made by a commercial CFD software. The performance of impulse turbine designed almost same as Setoguchi et al. (2001) was systematically investigated in various flow situations. Calculations were made using 2-D and 3-D methods, each having advantages and disadvantages. After testing grid dependency, total number of meshes were chosen to be 3600 in 2-D case and 190,000 in 3-D case, meaning the 3-D method is at least 50 times more time-consuming. Results were compared with experimental data available in Setoguchi et al (2001). The followings are summary and findings obtained through the present study.

(1) Calculated results generally agreed well with experimental data, while 3-D method showing better agreement. Discrepancies between calculation and experiment increased with increasing flow coefficient due to the three-dimensionality of flow field. The 2-D method was found to be acceptable in qualitative sense.

(2) The dependability and practicality of 2-D analysis was investigated by examining the effect of setting angle of guide vane. Both 2-D and 3-D analyses could predict the results surprisingly well. It was concluded that the simple 2-D analysis could be successfully utilized for parametric study in qualitative sense. More numerical and experimental evaluations will solidify the present conclusions.

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