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EFFECT OF TIP GROOVE ON BIPLANE WELLS TURBINE

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ABSTRACT

Oscillating Water Column (OWC) is one of the most used wave energy conversion devices around the world. OWC uses a Wells turbine as Power Take-Off (PTO) device. Previous research on monoplane Wells turbine shows that tip groove on the rotor blade improves the performance of the turbine. In the present work, tip groove casing treatment is applied to the biplane Wells turbine. Three different casing grooves having groove depth (GD) 1%, 3% and 5% of chord length are studied. A numerical analysis of the flow through the turbine is carried out by solving Reynolds averaged Navier-Stokes equations using ANSYS CFX® v16.0 commercial code. The performances of the biplane Wells turbine with and without tip groove are compared. It is found that circumferential tip groove improves the turbine operating range and the torque coefficient. Turbine with tip groove 3% of chord length produced better performance among all the cases studied. Using circumferential tip groove, 16.6% improvement is achieved in maximum torque coefficient for a particular operating speed.

Keywords: *Wells turbine, wave energy, tip groove, CFD, OWC*

NOMENCLATURE

ρ	Air density (kg/m^3)
ω	Angular velocity (rad/s)
C	Chord length
η	Efficiency
φ	Flow coefficient
u_a	Inlet air velocity (m/s)
Q	Mass flow rate (kg/m^3)
ΔP_0^*	Pressure drop coefficient
R	Rotor tip radius(m)
T	Torque(N-m)
C_T	Torque coefficient
ΔP_0	Total pressure drop(Pa)
u_t	Tip speed velocity (m/s)

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1.0 INTRODUCTION

Ocean waves are a huge source of energy and this energy can be harvested using Oscillating Water Column (OWC) type wave energy converter. The OWC can be installed near shore or away from the shoreline depending on the location. The energy conversion in OWC happens in three phases: first, the wave energy is converted to pneumatic energy inside the OWC using the oscillatory motion of the water. The pneumatic energy is converted to mechanical energy with the help of a turbine. Finally, a generator connected to the turbine converts the mechanical energy into useful electrical energy. The turbine used in OWC is a special type of turbine that can rotate in one direction irrespective of the direction of the airflow. Both impulse turbine and reaction turbines have been used for this purpose. Wells turbine is one type of reaction turbine suitable for OWC devices.

The Wells turbine consists of symmetrical aerofoil blades with 90-degree stagger angle. The blades are placed perpendicular to the hub with a tip clearance suitable for the purpose. The main parameters affecting the performance of Wells turbine are the solidity, tip clearance, aspect ratio and hub to tip ratio. Raghunathan [1] analytically studied the effect of different parameters on the performance of Wells turbine and suggested suitable range of values for the same. Brito-Melo et al. [2] carried out a numerical simulation to study the different parameters of Wells turbine. Raghunathan and Tan [3] suggested using Wells turbine with blades in two different planes (biplane Wells turbine) for better performance characteristics. Gato and Curran [4] carried out an experimental investigation to study the performance of a biplane Wells turbine. Halder et al.[5] studied the effect of circumferential tip groove on a monoplane turbine and achieved 26% increase in turbine power for a particular operating point.

In the present work, a numerical study of the effect of tip groove on biplane Wells turbine has been reported. The numerical results are compared with available experimental results. Biplane Wells turbine with different groove depths (GD) are numerically simulated and detail analysis has been presented in this paper.

2.0 COMPUTATIONAL METHODOLOGY

Figure 1 shows a biplane Wells turbine with tip groove having 4 blades in each plane. The turbine has an overall solidity of 0.64 (0.32 per plane). The blades have NACA0015 profile with chord length of 0.125 m. The gap between the two planes is 1.5 times chord length. The blades are positioned at 45-degree circumferential angle between the planes. As the turbine is symmetrical around the axis of rotation, only one-fourth section of the full turbine is taken as the computational domain. The tip clearance is taken as 1 mm throughout the computational domain and groove depth is varied for different cases. The computational domain consists of one blade from both the planes. Table 1 shows the geometric parameters of the reference turbine considered for analysis.

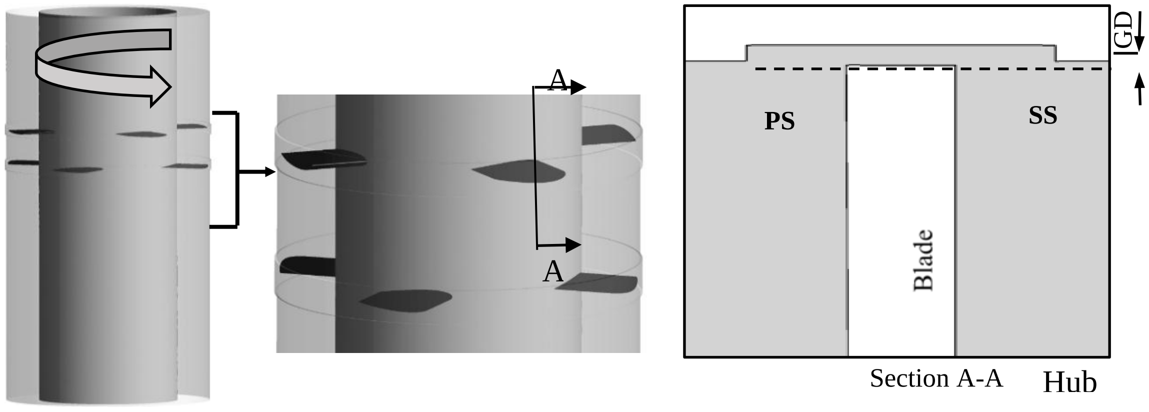
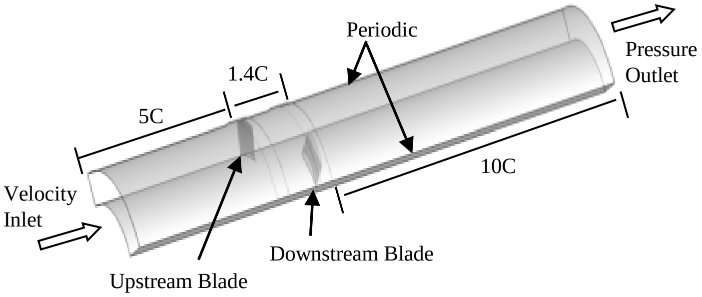


Figure 1: Biplane Wells turbine with tip groove

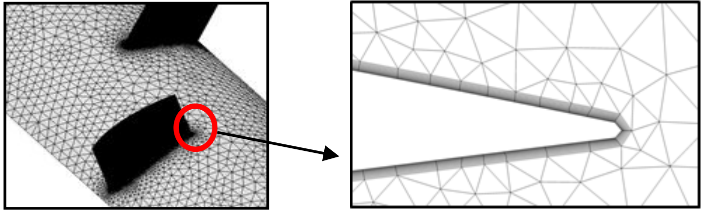
Table 1: Geometric parameters

Parameter	Specification
Blade profile	NACA0015
Blade chord length	0.125 m
Hub radius	0.2m
Tip radius	0.294m
Tip clearance	0.001m

Figure 2(a) shows the computational domain considered for analysis. The upstream side is taken as five times the chord length and the downstream side is ten times of chord length. The gap between the planes is 1.5 times the chord length. Figure 2(b) and 2(c) show the unstructured mesh with prism layers near the boundary of blades. Figure 2(d) shows the mesh in the tip groove region. The unstructured mesh is created using ANSYS ICEM CFD v16.0. The prism layer has 12 layers and the distance of the first layer from the boundary of the blade is 0.000012 m. The height of the first layer is calculated keeping the Y+ value less than 1. The fine boundary layers are used to capture the effect near blade wall more accurately.

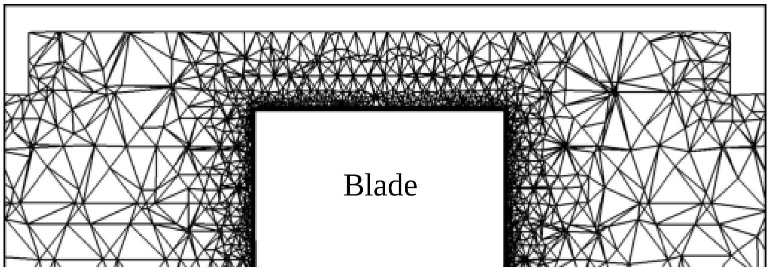


(a) Computational domain



(b) Mesh in the hub

(c) Mesh near blade surface



(d) Mesh in the tip groove region

Figure 2: Computational domain and mesh

Reynold’s Averaged Navier-Stokes (RANS) equation is discretized using finite volume method to solve the continuity, momentum and energy equations. A finite volume based commercial code ANSYS CFX v16.0 is used for the numerical simulation. SST k-Ω turbulence model is used as it captures the near wall effects more accurately. The convergence criteria are considered as RMS residual less than 1x10^{-6} and mass imbalance less than 0.001. The fluid (air) is incompressible and steady flow is considered. The Periodic boundary condition is imposed on the circumferential side and the hub, casing and rotor blade surface have no-slip boundary condition. A steady velocity is considered at the inlet and ambient pressure at the outlet. The rotational speed of both the rotor is fixed at 2500 rpm. Due to the computational requirement, the numerical simulation is done using a supercluster having a total computational power of 97 TFlops. Using this high-performance computing system, the average time for each simulation is 10 ~ 12 hours.

In the present work, three different turbines with a groove depth of 1%, 2% and 3% of chord length are simulated and compared with reference turbine without tip groove. The turbine solidity, rotational speed, and other boundary conditions remain constant for all the cases studied.

3.0 RESULT AND DISCUSSION

As per the literature available on Wells turbine, the performance is expressed using four nondimensional parameters:

i. The torque coefficient:

$$C_T = \frac{T}{\rho\omega^2 R^5} \tag{1}$$

ii. The pressure drop coefficient:

$$\Delta P_0^* = \frac{\Delta P_0}{\rho\omega^2 R^2} \tag{2}$$

iii. The efficiency:

$$\eta = \frac{T\omega}{\Delta P_0 Q} \tag{3}$$

iv. Flow coefficient:

$$\varphi = \frac{u_a}{u_t} \tag{4}$$

The turbine is simulated at five different flow coefficient from 0.075 to 0.225. For each flow coefficient, the inlet velocity is different. However as the rotational speed is constant, the tip velocity remains constant for any change in flow coefficient. The numerical results are first validated with experimental results of Gato and Curran [4]. Figure 3 shows the present numerical results are in good match with experimental results and the deviation is within ±5% for all three parameters.

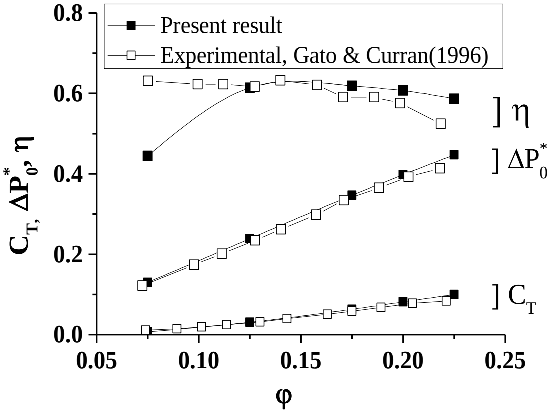


Figure 3: Validation with experimental result

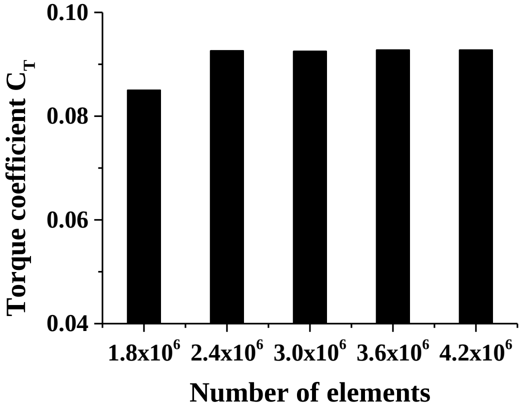
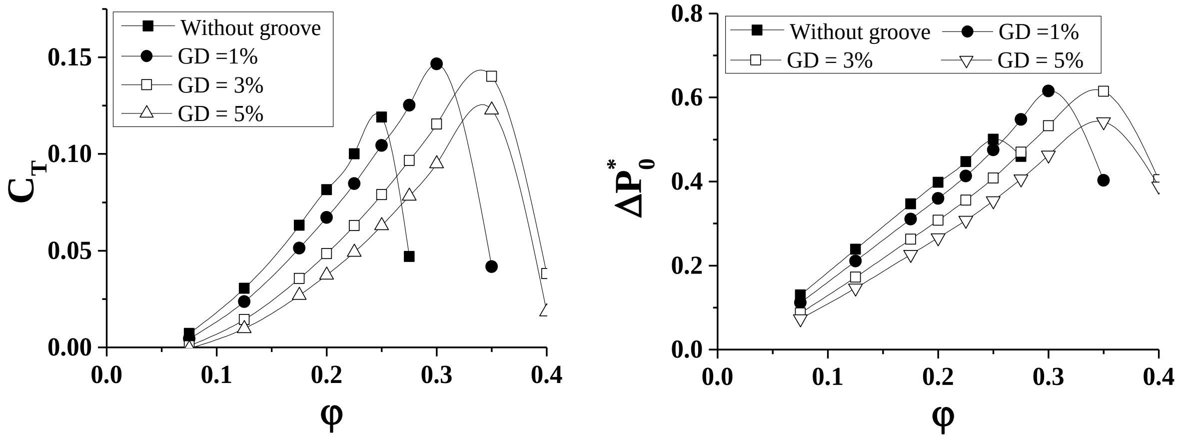


Figure 4: Grid independence test

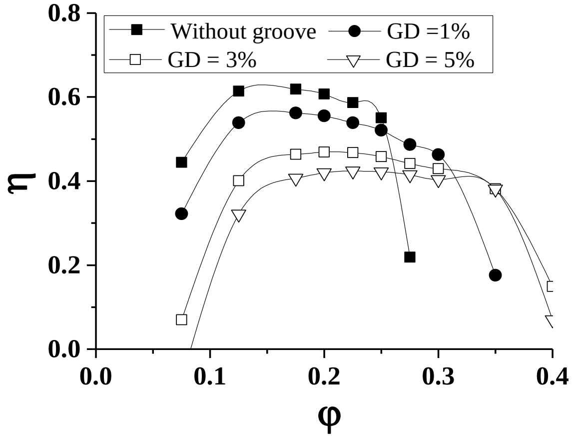
A grid independence test is carried out to find out the optimum number of elements of the computational domain. Figure 4 shows the grid independence test result. The number of elements is varied from 1.8 to 4.2 million. It can be seen that the torque coefficient does not vary after 2.4 million elements. So the optimum element number is taken as 2.4 million for this numerical study.

Figure 5 shows the comparison of different tip groove depths with reference turbine having no tip groove. It can be seen that torque coefficient and pressure coefficient increases significantly when tip groove is used. The torque coefficient increases for groove depth of 1% and 3% of chord length; however it decreases when groove depth is increased more. The stall point is moved from φ = 0.225 for no tip groove to φ = 0.350 for tip groove with groove depth equals to 3% of chord length. As a result, the turbine with groove depth has a significantly wide operating range. However, it can be seen that the efficiency is decreased for turbines when tip groove is employed. Efficiency is a function of both torque coefficient and pressure coefficient. As the increase in pressure coefficient is more compared to torque coefficient, the efficiency decreases for the use of tip groove. However, the operating range remains wider for all the cases with tip groove. It can be seen that tip groove with groove depth 3% of chord length gives the highest torque coefficient and wider operating range among all the cases considered here.



(a) Torque coefficient

(b) Pressure Coefficient

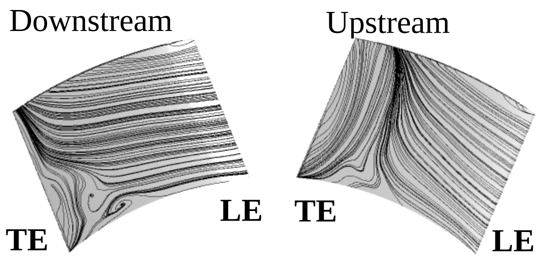
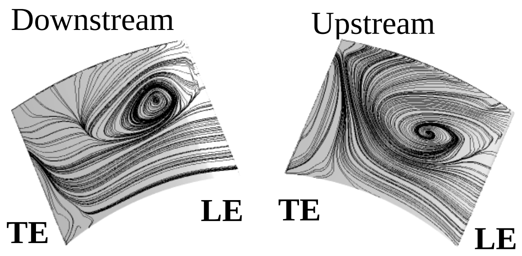


(c) Efficiency

Figure 5: Comparison of different groove depths

Figure 6 shows the streamlines on the suction surface of the blade for lower (φ = 0.225) and higher (φ = 0.275) flow coefficient. At φ = 0.225, the streamlines are attached to blade suction surface for all the cases. No flow separation occurs for low flow coefficient. However, at higher flow coefficient φ = 0.275, a strong vortex structure is seen on the suction surface of the blade for the reference turbine without any tip groove. As a result, the torque coefficient, as well as efficiency, drops significantly beyond this flow coefficient. However, at φ = 0.275, turbines with a tip groove, do not show any significant flow separation on the suction surface of the blade. The flow is attached till the trailing edge of the blade. As a result, the stall phenomenon does not occur at this point and the turbine with groove depth can operate for a higher range of flow coefficient.

Figure 7 shows the streamlines at different blade span of the turbine for the reference turbine and the one with 3% groove depth. At φ = 0.225, there is no flow separation occurring at any blade span for both the reference turbine as wells as a turbine with 3% groove depth. At φ = 0.275, for turbine without tip groove, flow separation occurs near the leading edge of the blade and a vortex structure is seen at 75% of blade span. The separation starts occurring near the hub and it propagates towards tip region. At 95% of blade span, i.e, near the tip, the reverse flow on the surface of the blade can be observed if no tip groove is used. However, when tip groove is used, the flow is always attached with blade surface at this flow coefficient. So the turbine with tip groove has a smooth operation at higher flow coefficient without any stall phenomenon.

	φ = 0.225	φ = 0.275
Without groove		

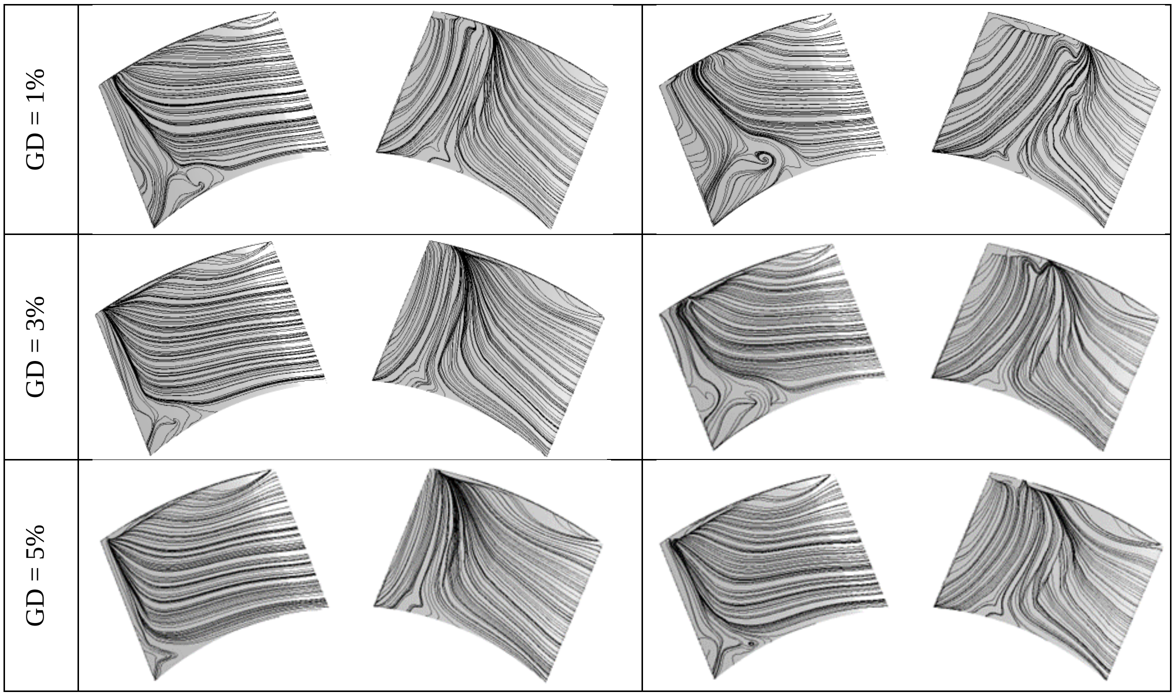


Figure 6: Streamlines on the suction surface of the blade

Figure 8 shows the contour of pressure coefficient at φ = 0.275 where the stall occurs for turbine without any tip groove. A comparison of pressure contours at a plane through mid-chord shows the variation in pressure from hub to tip region when the tip groove is used. When the tip groove is used, the low-pressure region is concentrated only near the tip region. However, in case of reference blade without any groove, the low-pressure region is spread across the suction surface of the blade.

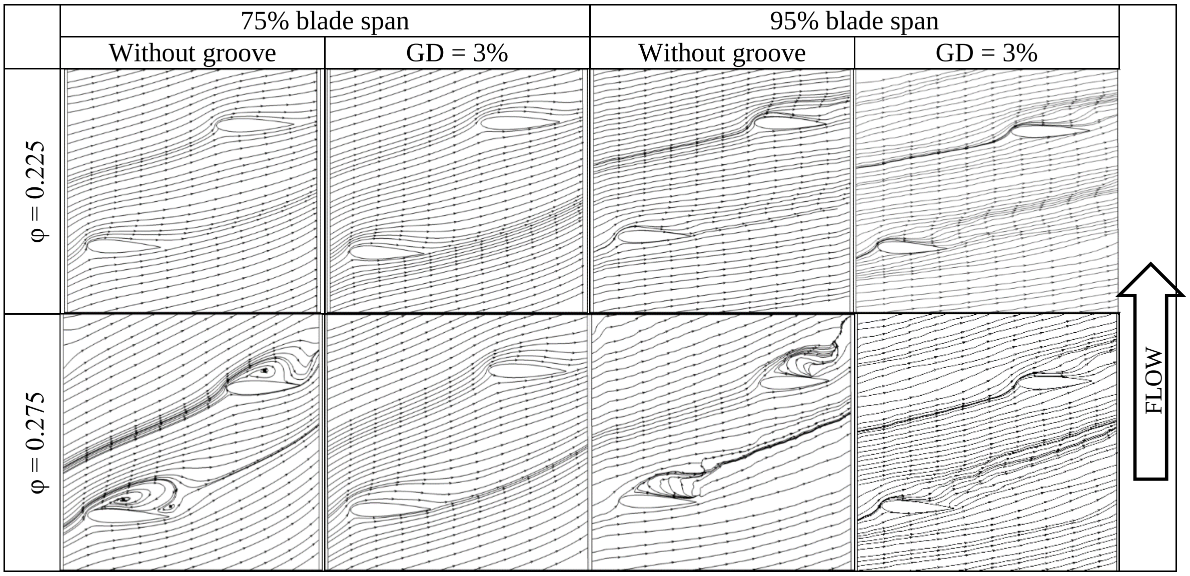


Figure 7: Streamlines at different blade span

Without groove		GD=3%	
Upstream	Downstream	Upstream	Downstream

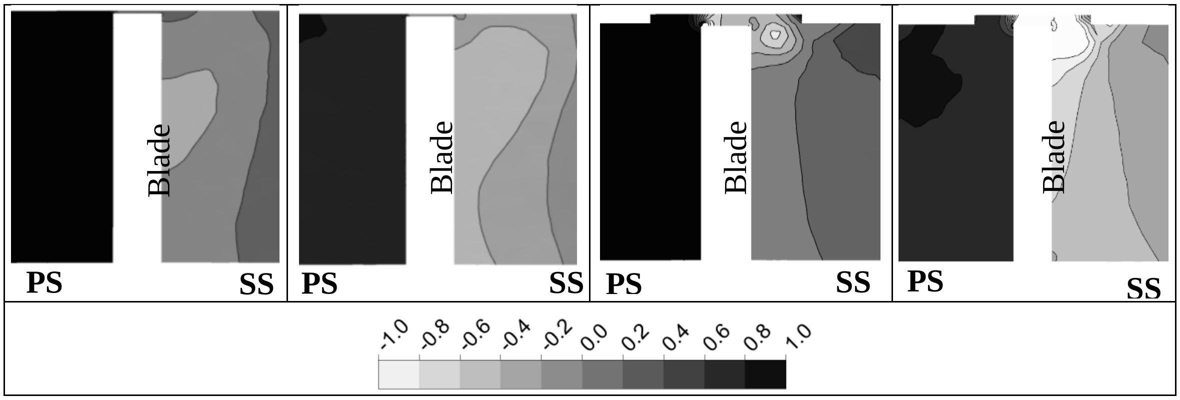


Figure 8: Contour of pressure coefficient at mid-chord plane (φ =0.275)

4.0 CONCLUSION

The effect of tip groove for a biplane Wells turbine for wave energy conversion has been studied numerically for different groove depths and the results are compared with a reference turbine without any tip groove. The use of tip groove delays the stall phenomenon and gives the turbine a wider operating range. The torque coefficient increases till a certain groove depth. Among the different cases studied, it is seen that turbine with a groove depth of 3% chord length gives the highest torque coefficient and maximum operating range. An improvement of 16.6% in torque coefficient is achieved when the tip groove is used with a groove depth of 3% chord length.

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