DESIGN AND ANALYSIS OF HYDROKINETIC TURBINE USING SAVONIUS BLADE

A Dissertation submitted towards the partial fulfilment of
the requirement for the award of degree of

Master of Technology
in
THERMAL ENGINEERING

Submitted by
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2K14/THE/25

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(Formerly Delhi College of Engineering)

CERTIFICATE

This is to certify that the dissertation title "**Design and Analysis of Hydrokinetic Turbine Using Savonius Blade**" submitted by **Mr. SIDHANT MALLICK, Roll. No. 2K14/THE/25**, in partial fulfilment for the award of degree of Master of Technology in "THERMAL ENGINEERING**",** run by Department of MECHANICAL Engineering in Delhi Technological University during the year 2014-2016., is a bona fide record of student's own work carried out by him under my supervision and guidance in the academic session 2014-16. To the best of my belief and knowledge the matter embodied in dissertation has not been submitted for the award of any other degree or certificate in this or any other university or institute.

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DECLARATION

I hereby declare that all the information in this document has been obtained and presented in accordance with academic rules and ethical conduct. This report is my own work to the best of my belief and knowledge. I have fully cited all material by others which I have used in my work. It is being submitted for the degree of Master of Technology in Thermal Engineering at the Delhi Technological University. To the best of my belief and knowledge it has not been submitted before for any degree or examination in any other university.

SIDHANT MALLICK

M. Tech. (THERMAL)

2K14/THE/25

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ABSTRACT

Hydrokinetic energy is the energy of a water mass due to its flow. The greater the water velocity, the greater is the hydrokinetic energy it posesses. Hydrokinetic turbines are used to covert this kinetic energy to obtain electricity. Since the performance of these turbines played an important role in viability of hydrokinetic power generation projects hence different design parameters are studied to enhance its power generation capacity and performance.

In this project torque and rotational speed of simple Savonius turbine are obtained at different speeds. The values of stream velocity are taken as 0.8m/s and 2.0 m/s for 2D-transient analysis where variation of speed with time is studied. Diameter of the blade is taken as 0.266m and height of the blade is taken as 0.17m. For 3D-steady analysis stream velocities are taken as 0.3m/s, 0.65m/s and 0.9 m/s. The design of turbine blade is done on Solid Works, domain of the blade and meshing is done on ICEM CFD. CFD analysis is done with the help of ANSYS FLUENT. Using transient setup, a plot between time and rpm is plotted.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF NOMENCLATURE

A frontal area of the turbine = H × D (m^2)

Re Reynolds number

T torque generated by turbine (N m)

V free stream velocity (m/s)

v_1 velocity at upstream of the turbine (m/s)

v_2 velocity at downstream of the turbine (m/s)

W width of the channel (m)

D diameter of the turbine (m)

H height of the turbine blades (m)

I turbulence intensity

k turbulent kinetic energy (J/kg)

P_{max} maximum power available in the fluid stream (W)

P_{rot} power extracted by the turbine (W)

C_p coefficient of power

LIST OF ABBREVIATIONS

Rpm rotational speed of turbine

TSR tip speed ratio

LIST OF GREEK SYMBOLS

ρ density of water (kg/m3)

μ dynamic viscosity of water (kg/m s)

ω angular velocity of the turbine (rad/s)

$\in$ rate of dissipation (J/kg s)

1. INTRODUCTION

1.1 ENERGY AND HYDRO-POWER

Energy is one of the most basic requirements of our universe. Energy has come to be known as a strategic commodity and any uncertainty about its supply can affect the functioning of the whole economy, particularly in developing economies. The substantial and sustained economic progress of India is placing enormous demand on its energy resources. The supply and demand imbalance in energy sources is pervasive requiring effective efforts by Government of India to fulfil energy supplies as India faces possible very severe energy supply constraints. Energy requirement in our country is increasing at an enormous rate. Achieving energy security in this strategic sense is of basic importance not only to India's economic development rate but also for the human development objectives that focuses at alleviation of unemployment, poverty and aims at meeting the Millennium Development Goals. Holistic planning to attaining these objectives requires quality energy statistics that is capable to address the issues related to energy demands, energy, poverty and environmental effects of energy growth.

Change in climate is the most urgent and biggest environmental threat in the world. The greenhouse effect which is produced by human activities, by burning of fossil fuels such as gas, oil and coal for energy produced greenhouse gases, cause increase in global temperatures, leading to more severe weather patterns such as droughts, floods and storms, rising sea levels and threats to the whole ecosystems. To avoid inconsistent environmental condition rising global emissions must decelerate within the next 10 years. This means we need to adopt those forms of energy that do not emit GHGs and as such we go for renewable sources of energy which are also green energy sources. Demands for renewable energy sources are accelerating with the recent deficit in fossil fuels, and as such hydro energy conversion has become a most reliable technology for power generation.

Ministry of New and Renewable Energy has developed a new data base of potential sites of small hydro and more than 6000 potential sites with an aggregate capacity of around 20,000 MW for projects up to 25000 KW capacity have been identified. From the above data we can conclude that in India we have scope to construct small hydro power station at remote area to make life of people better and fulfill their requirement. After this reassessment had been done and the total potential of India is given below the table no 1 till 2014:

Hydroelectric power (hydropower) is a renewable source, but it has many social and environmental impacts. Large hydropower projects involve the use of dams, which by their nature dramatically alter an otherwise free-flowing river by storing water and releasing it (either through generation or spill), usually in a controlled manner. Power is obtained by extracting the kinetic energy of water running through a turbine, and converting it to electricity by coupling the turbine to an electric generator. While these dams do not consume

fossil fuels, they alter ecosystems and affect wildlife and the people who depend on that water. The effects include, but are not limited to, changes to water temperature, level of dissolved oxygen and other water-quality parameters; decline in river species; increased erosion along riverbanks and altered sediment transport.
Hence from water sources hydrokinetic turbine is an alternative power extraction way to utilise energy.

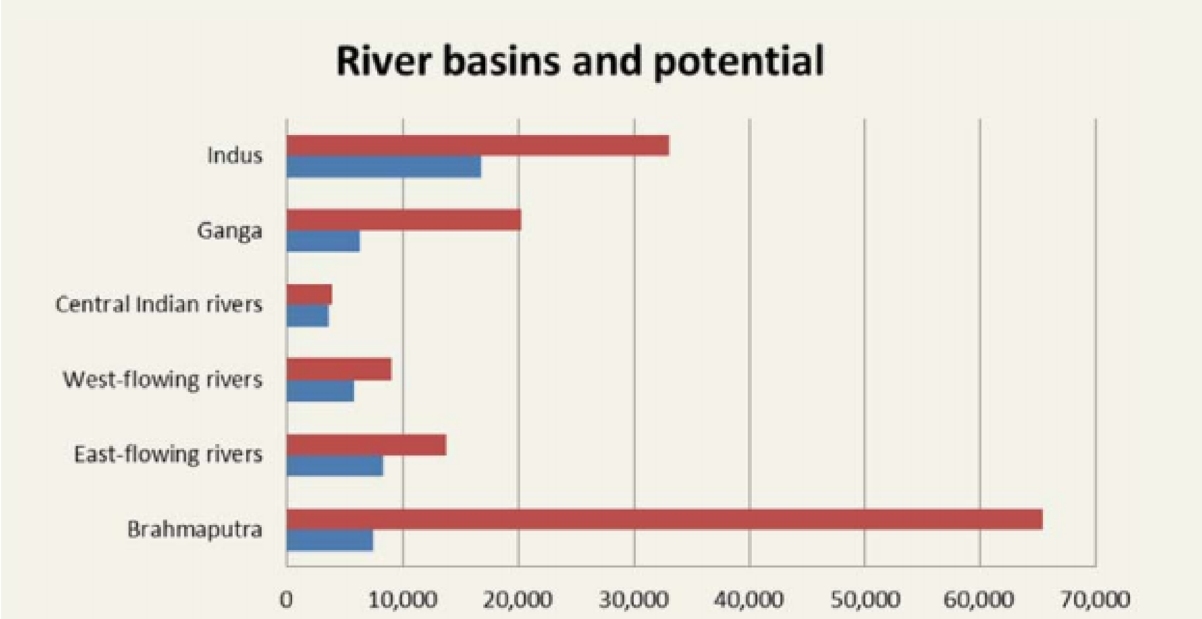
Table 1.

S. NO.	ASSESSEMENT YEAR	TOTAL SATAE COVER UNDER ASSESSEMENT	POTENTIAL MW
1	2002-2003	8	4326
2	2003-2004	6	2696
3	2004-2005	5	1387
4	2005-2006	3	913
5	2006-2007	10	4855
6	2007-2008	4	3470
7	2008-2009	4	860
8	2009-2010	3	4570
9	2010-2011	7	2510
10	2011-2012	6	3120
11	2012-2013	6	2516
12	2013-1014	7	6188
13	2014-2015	1	93
	GRAND TOTAL	70	37540

1.2. HYDROKINETIC TURBINES

The hydrokinetic conversion device is an emerging class of renewable energy technology and it offers ways to extract the energy of flowing stream without the impoundment or diversion of the conventional hydropower facilities which is based on dams.

Table 2.



	Brahmaputra	East-flowing rivers	West-flowing rivers	Central Indian rivers	Ganga	Indus
Assessed (MW)	65,400	13,775	8,997	3,868	20,252	33,028
Developed and under construction (MW)	7,412	8,253	5,761	3,548	6,294	16,840

1.3 CONCEPT OF HYDROKINETIC TURBINES

Hydrokinetic energy conversion is process of utilization of kinetic energy contained in river streams or other artificial waterways for generation of electricity. The devices which are used for the harnessing of kinetic energy of free stream flow of river or canal are called hydrokinetic turbines. These turbines are also known as water current turbine, free flow/stream turbine zero head hydro turbine [1] and Ultra low head hydro turbine,. This innovative kind of renewable energy is being considered as unconventional and unique technology, which shows a large application possibility to use river currents. In contrast to conventional hydroelectric plants, hydrokinetic converters are made without significantly changing the natural pathway of the water stream.

1.4 WORKING PRINCIPLE OF HYDROKINETIC TURBINE

These turbines work on same principle, on which wind turbines work where kinetic energy of the streaming fluid is utilized to rotate an electromechanical energy converter and subsequently generate electricity. The output power of hydrokinetic turbine is directly dependent on the density of fluid, cross-sectional area, and fluid velocity cubed.

Governing equation is similar to wind turbine and as follows:

$$P = (1/2) \times \rho \times A \times v^3 \quad (2)$$

Hydrokinetic turbines are designed for water velocity of 0.8m/s to 2.5m/s which is much lower than wind speed. Water is almost 835 times denser than wind. Considering these facts, water current has higher energy capacity than the wind [2]. Water current turbines extract energy from the fluid by reducing flow velocity.

1.5 CLASSIFICATION OF HYDROKINETIC TURBINE

This list is by no means exhaustive and many of the concepts are adopted from the wind engineering domain. Based on the alignment of the rotor axis with respect to water flow, two general classes could be formed, namely, the axial and cross flow turbines. The axial turbines have axes parallel to the fluid flow and employ propeller type rotors. On the other hand, the cross flow types encounter water flow orthogonal to the rotor axis and mostly appear as cylindrical rotating structures [3]. The classification of hydro kinetic turbines has shown in fig.

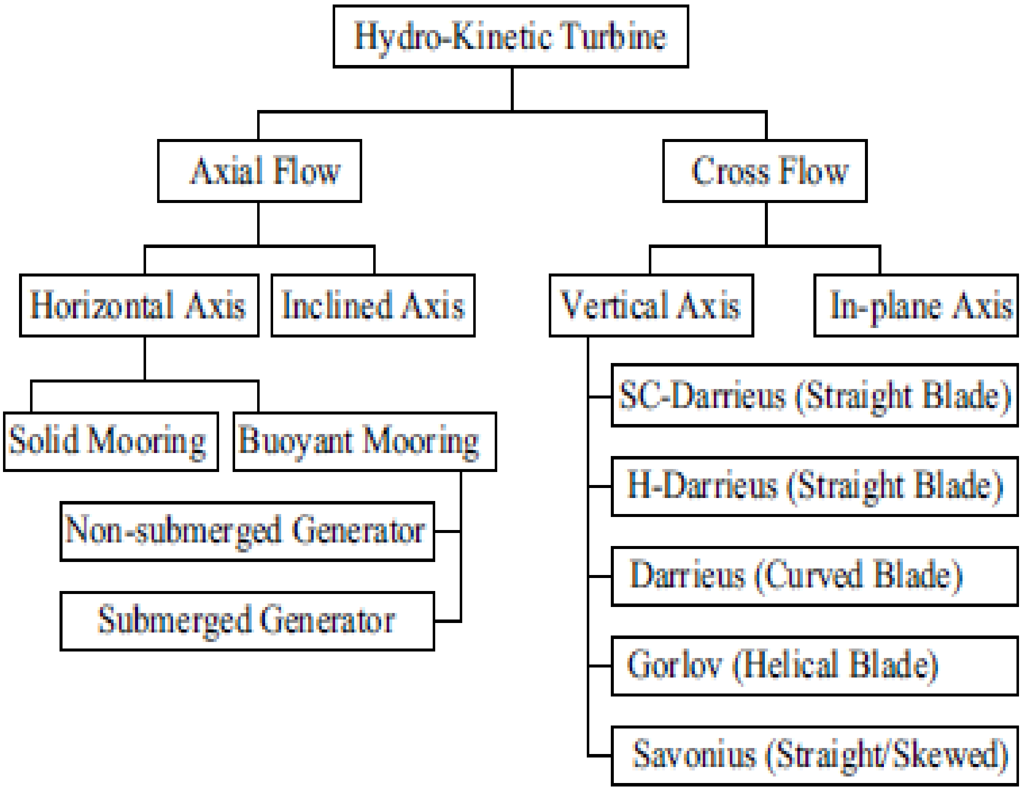


Fig. 1 Classification of hydrokinetic turbines

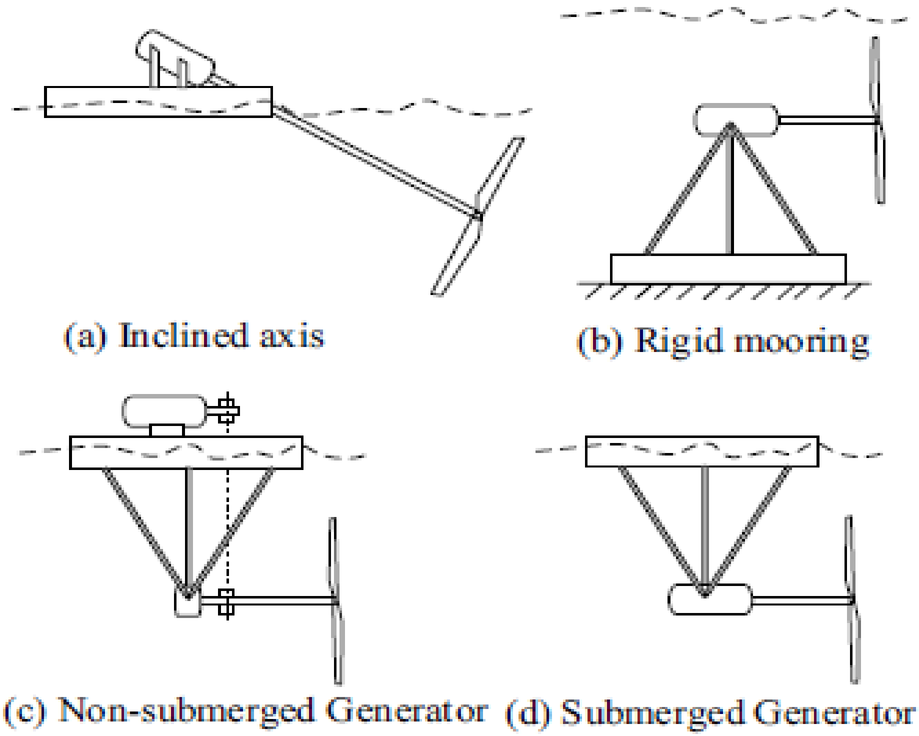


Fig.2 Axial flow turbines

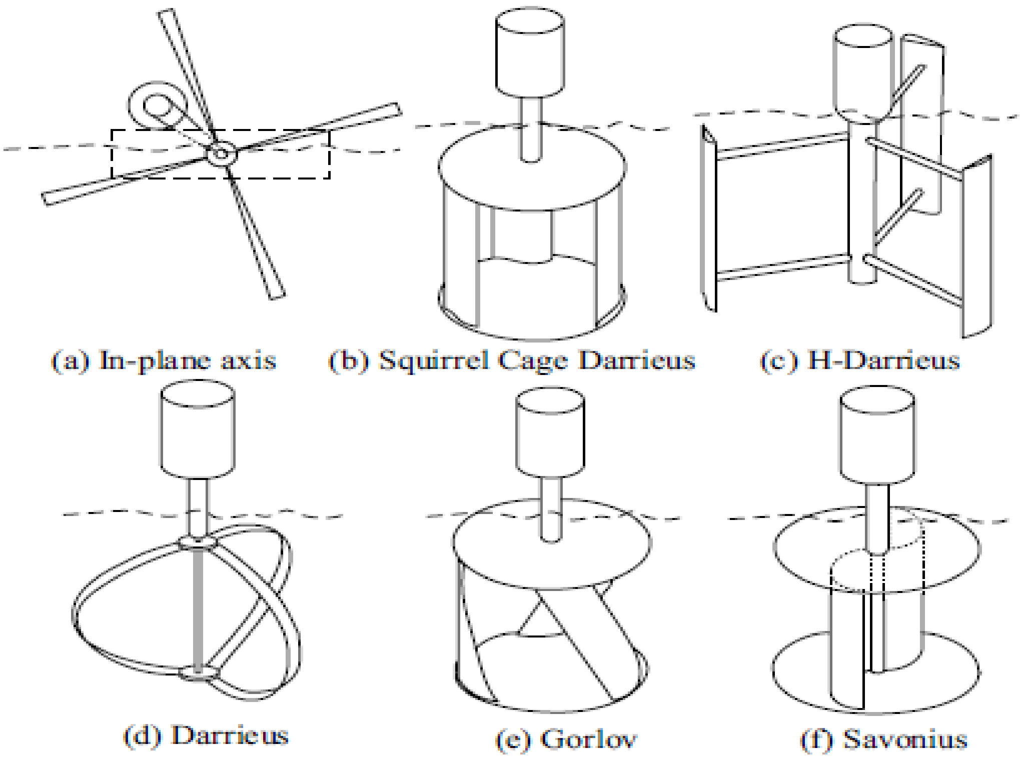


Fig.3 Cross-flow turbines

Betz's law indicates the maximum power that can be extracted from the wind, independent of the design of a wind turbine in open flow. It was published in 1919, by the German physicist Albert Betz. The law is derived from the principles of conservation of mass and momentum of the air stream flowing through an idealized "actuator disk" that extracts energy from the wind stream. According to Betz's law, no turbine can capture more than 16/27 (59.3%) of the kinetic energy in wind. The factor 16/27 (0.593) is known as Betz's coefficient. Practical utility-scale wind turbines achieve at peak 75% to 80% of the Betz limit.

The Betz limit is derived on the basis of an open disk actuator. If a diffuser is used to collect additional water flow and direct it through the turbine, more energy can be extracted, but the limit is applied to the cross-section of the complete structure. The Betz limit is not dependent on the geometry of the water extraction system, hence it may take any form condotioned that the flow travels from the entrance to the control volume and to the exit, and the control volume has uniform entry and exit velocities. Any extra effects can only reduce the act of the system (usually a turbine) since this analysis was idealized to disregard friction. Any non-ideal effects would deviate from the energy available in the incoming fluid, lessening the overall efficiency. Some inventors manufacturers and have claimed that of exceeding the Betz' limit by using nozzles and other water diversion devices, generally by misrepresenting the Betz limit and manipulating only the rotor area and not the total input of water contributing to the hydrokinetic energy extracted from the system.

1.6 SAVONIUS ROTOR

Savonius rotor is a vertical axis turbine which is principally a drag type rotor and was proposed by Finnish Engineer Savonius. The fundamental shape of this rotor is an ‘S’ form, which has a small overlap amid of two semi-circular blades. Savonius rotor is very simple construction with low noise and low cost. It has good starting characteristics and an ability to accept fluid from any direction. The working theory of Savonius rotor is based on the variation of the drag force between the convex and the concave parts of the rotor blades when they rotate about a vertical shaft. The drag coefficient for convex surface is less than concave surface. So the returning blade with concave side facing the water flow would experience less drag force than the advancing blade.

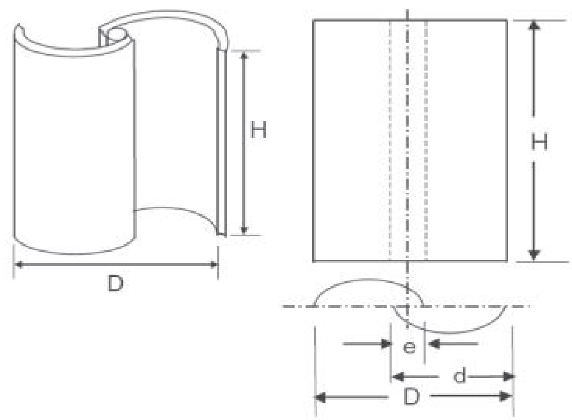


Fig.4 Schematic diagram of Savonius turbine

1.7 CHANNEL DESIGN

A channelling device can be used to modify flow conditions in the vicinity of the rotor. This systems developed from theoretical modelling and small-scale model tested in a hydrodynamic test channel. The basic advantages of this kind of channel include little need for civil works , relocation, ease of transport, self-regulated operation, and it is to be a low-cost long-lifetime system and improving the load factor.

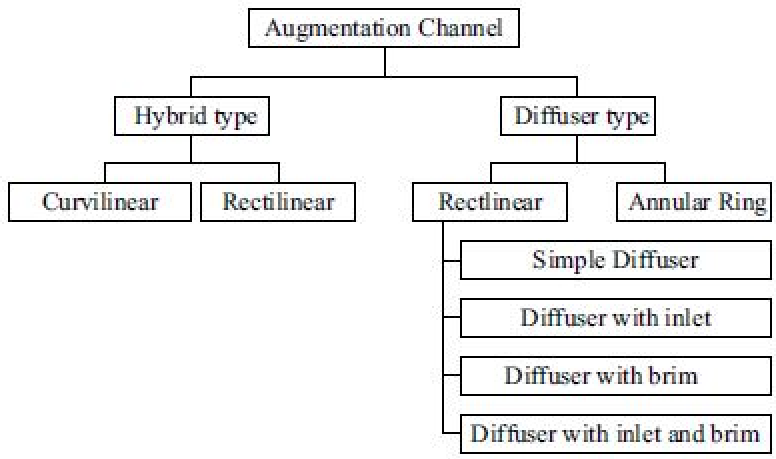


Fig.5 Classification of augmentation channel

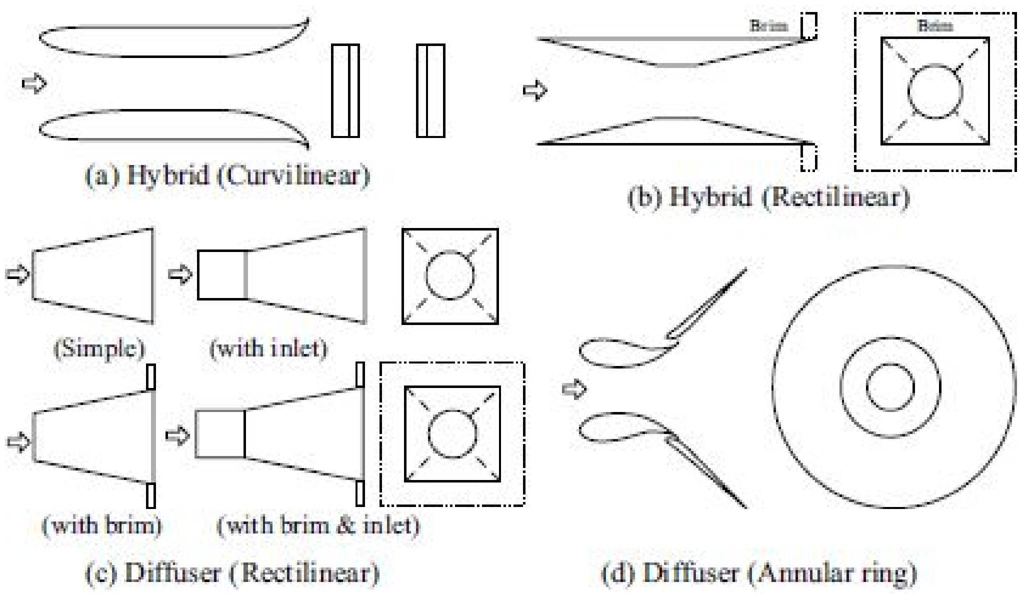


Fig.6 Shapes of augmentation channel

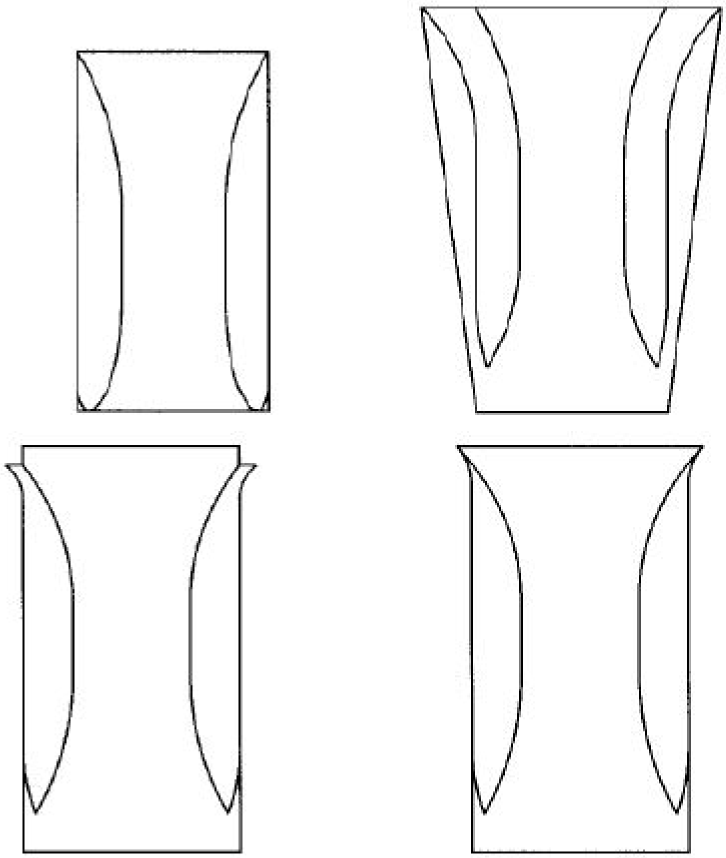


Fig.7 Types of channel

The design of the channelling used to improve performance of rotor, the same power output to be obtained from a smaller rotor. Reduction of the size of moving parts simplifies the design process and reduces the initial cost of the unit. Local speed flow increment in the rotor produces effect of a smaller turbine size implies a reduction of the rotor tip radius, that is, at equal flow speed, an increase in the rotor angular velocity for the same tip speed ratio and the velocity in the neighbourhood of the rotor is higher than the free stream velocity of the river, for a constant, the rotor revolves faster. Thus, there would be fewer saving as compared to the use of a channelling device, as proposed here. The channelling device also acts as a flow speed amplifier with a high gain at low river current velocities and with a progressive reduction of gain when the river speed increases. Thus, the channelling device acts as a regulator, stabilizing flow speed in the neighbourhood of the rotor. Moreover, as the output power depends on the cube of flow velocity in the neighborhood of the rotor, small variations of flow speed produce very significant fluctuations of generated power. Basically the channel section acts as a nozzle device which improves the speed of water at turbine section [34]. The hyperbola channelling device is proved to be a kind of optimizing channelling device which has much more advantages such as less resistance effects and more fluency line shape and so on.

2. LITERATURE REVIEW

1.Garrett C et. al proposed that power generation of hydro turbines in unconstrained channels is comparable to wind turbines. The extracted power by the turbine can be established using the following equation.

$P_{extracted} = 0.5\ \rho\ A\ u_0{}^3\ \eta_e$

where, $\eta_e = C_p$ is the extraction efficiency or power coefficient, ρ is the fluid density, A is the area, u_0 is the upstream, uninterrupted speed of the flow.[4].

2. **Lanchester (**1915) and a few years later by Betz (1920) , it was found that the highest power can be harnessed from an ideal turbine with no channel constraints based on conservation of mass and momentum on the streamtube, greatest power is achieved when induction factor is 1/3. In this case, $\eta_{e,max}$ is 16/27 or 0.59 which means only 59% of the kinetic power of the flow with no constraint can be extracted by the turbine. The derivation for this theory can be found in [5].

3. **B Jones** performed experiments on various fluids and gave the concept of relationship between wind turbines and hydrokinetic turbines, so later on all experiments were carried on accordingly using concepts of wind turbines. He gave the concept of tip speed ratio (TSR), defined as the ratio of speed of blade at tip to speed of the flowing water.

$\lambda = \frac{\omega R}{2 \times V}$, where ω is the rotational speed of the turbine, R is the radius and V is the speed of the water stream.

4**. G. Riegler** gave the concepts of coefficients of power (by performing experiments on wind flow). He found that there must be enough time lapses between blades travelling through the same location so that new water can enter and next blade can harness power from the new water, not the used turbulent water. As per the definition of coefficient of power, it is the ratio of power developed by the turbine to the power of available stream. The relation between power coefficient Cp and TSR can be understood naturally. If the blade of the turbine spin too slowly, then the majority of the water will get ahead of the rotor without being tapped by the blades. On the other hand, if the turbine spins too speedily, then the blades will always move through turbulent, already used water. Enough time lapses must be between the two blades travelling through the same locality so that fresh water can penetrate and the subsequent blade can exploit the power from that new water, not the turbulent, used water [6].

5. **Veronica B Miller et. al** in their review work stated that hydrokinetic energy extraction (HEE) has received great attention recently as a sustainable means to lessen the strain on current energy technologies. HEE extracts kinetic energy rather than potential energy (as in the conventional hydropower dams), and can be applied to some ocean, tidal, and river energy uses. Their main focus was on extraction of river hydrokinetic energy. Even though they anticipated many new types of hydrokinetic devices, comparison of wind turbine theory

to hydrokinetic governing principles and CFD were discussed. The effort presents a CFD analysis for underwater water wheel turbine for both 2D and 3D cases. Insights into the consequential flow, behaviour, environmental impact and the implications for power extraction were discussed [7].

6. Tahir Yavuz performed several experiments on hydrokinetic turbines and concluded tha hydrokinetic energy holds noteworthy promise as an innovative, carbon-free energy supply. Power from the hydrokinetic turbine is harnessed from moving water without the construction of a dam. Operational success of the wind and hydrokinetic turbines depends on how effective the airfoils we choose. By tradition, benchmark airfoils, like GOTINGEN and NACA, are used for wind and hydrokinetic turbines generating energy have the highest lift coefficient about 1.30 and at the stall angle of attack of around 12. At these values, the flow velocities to generate electric energy are 3.0 m/s and 7.0 m/s for hydrokinetic turbine and wind turbine respectively. By means of double blade airfoil, the fluid dynamics overriding the flow field minimizes the separation bubble by the insertion of the high momentum fluid all the way through the gab of a double-blade of airfoil by meaning of the flow control delays the stall with a maximum angle of attack of 20, with a highest lift coefficient of 2.06. Consequently, by the use of double blade airfoils in the wind and hydrokinetic turbines, least wind and hydrokinetic flow velocities to produce cost-effective energies will be 1–1.5 m/s or less for hydrokinetic turbines and 3.0–4.0 m/s for wind turbines. Thus, the wind power and hydrokinetic potentials of Turkey will be re-defined and increased accordingly [8].

7**. Zhen Hu et. al** focussed on reliability of hydrokinetic turbines. Reliability is an important aspect to be considered during the designing of hydrokinetic turbine. The reliability of turbine blade acts a critical role in the on the whole reliability of the hydrokinetic turbine system. In this work, they developed an analysis model of time-dependence reliability for the blades of hydrokinetic turbine. The blade element momentum theory was used to set up the limit-state function. The results show that the model can very effectively estimate the reliability of the hydrokinetic turbine blade over a period of fixed time duration. The analysis also estimated that a cut-out velocity affects the reliability of the hydrokinetic turbine in the following two manners: First, with a cut-out river velocity, the role of the river flow velocity to the possibility of failure minimizes. Second, a properly selected cut-out river velocity can decrease the probability of failure of the blade drastically [9].

8. **O. Paish et. al** in their research concluded that potential in hydro are of two which can be exploited to generate electricity i.e. hydrokinetic and hydrostatic. Hydrostatic is the potential energy of a water body due to its altitude with respect to a reference point. Traditional hydropower plants make use dams and construct reservoirs to store water with a large amount of hydrostatic energy with the intention of harnessing the energy in a handy manner to produce electricity. Hydrokinetic technology has the potential to produce a large amount of electricity with a least impact to the environment [10].

9. **F. Ponta et. al** discussed the importance of augmentation channel in his papers. Augmentation channels bring about a sub-atmospheric pressure inside a constrained area and

thus increase the flow velocity. If a turbine is placed in such a channel, the velocity in the region of the rotor would be greater than that of a free rotor. This increases the achievable overall power capture considerably. Besides, it may help control the speed of the rotor and trim down the problems of low-speed drive train design [11].

10. **B. Kirke** in his research work, after performing several reviews concluded that based on the alignment of the rotor axis with respect to water flow, two general types could be formed, that is, the cross flow and axial turbines. The axial turbines have their axes parallel to the flow of fluid and make use of propeller type rotors. Alternatively, the cross flow types of turbine encounter flowing water orthogonal to the axis of the rotor and predominantly come out as cylindrical rotating structures [12].

11. **Antonio Carlos Fernandes et. al** in his study propose an innovative turbine that consists of a vertical flat plate, which is allowed to rotate freely about a vertical axis of symmetry and exploits the autorotation phenomenon to produce the energy from the current; the system is known as Vertical Axis Autorotation Current Turbine (VAACT) here. The VAACT utilises the additional mass moment of inertia to improve the rotational quality which is state of the art in this turbine. This paper covers the principle behind the VAACT and outlines procedures and results for experiments proving the basic feasibility. Also, the operation of the turbine will be discussed by experimental results of the torque coefficient, performance and kinematic parameters. Tests were conducted in LOC (Laboratory of Wave and Current) at COPPE/Federal University of Rio de Janeiro (COPPE/UFRJ). Utilizing the extra mass moment of inertia on the VAACT causes to extract the energy as an efficient turbine in very low head current. The maximum VAACT efficiency has been obtained experimentally in optimum non-dimensional moment of inertia in range of 0.5 and 0.6 [13]

11. **S.J. Savonius** developed a rotor called which was basically designed for wind power extraction. This rotor was named after him later i.e. Savonius. Savonius rotor is a vertical axis turbine which is principally a drag type rotor and was proposed by Finnish Engineer Savonius. The fundamental shape of this rotor is an ‘S’ form, which has a small overlap amid two semi-circular blades. Savonius rotor is very simple construction with low noise and low cost. It has good starting characteristics and an ability to accept fluid from any direction [14].

12. **M. A. Kamoji et. al** came to the conclusion that Savonius turbine has small aerodynamic efficiency corresponding to Darrieus type turbine. The working theory of Savonius rotor is based on the variation of the drag force between the convex and the concave parts of the rotor blades when they rotate about a vertical shaft. The drag coefficient for convex surface is less than concave surface. So the returning blade with concave side facing the water flow would experience less drag force than the advancing blade. The traditional single stage Savonius rotor has a great static torque changes with the rotor angle. The rotor also develops a negative static torque at certain angular positions. The torque variation and the negative torque have an adverse impact on the use of Savonius rotor for different applications. They proposed that the savonius turbine can run at low speed. Since it has low starting torque as it is simple in construction and its cost is too low, it is recommended to use savonius turbine for

power generation and also it is very easy to couple it with generator and make it easy in maintenance [15].

13. **M. J. Khan et. al** in their research work writes tha river Current Turbines are electromechanical energy converters that harness kinetic energy of a stream of flowing river water to generate electricity. Research in this domain is limited and various concepts are emerging only recently. In this paper, an extensive technology survey and comparison of various system options are discussed in order to formulate a basis for further analysis. Simplified mathematical modelling of an augmentation device and Darrieus type rotor has been carried out. Simulations are done in MATLAB and results are given with graphical interpretations. In conclusion, directions for further investigation are given and the potential of this technology is re-stressed [16].

14. **A J Alexander et. al** defined the blockage ratio as the ratio of the frontal swept area of the rotor to the frontal area of the computational domain which is represented by the Eq. (viii). The blockage ratio has great physical significance on the performance of the turbine, the value of which must be confined to a certain limit. A large value of the ratio shall indicate increased velocity of water near the channel walls along with excessive turbulence [17].

15. **K Golecha Alexander et. al**, in their research work found that the blockage ratio above 30% essentially needs correction. Also the work elucidated that a blockage ratio of 30% to 35% necessitates the velocity correction factor with an order of 40% to 50%. In the present work, this ratio has a small value and thus the correction required is extremely less and thus can be neglected. The velocity of the water current is different at different water depth due to which the velocity of the water current has been measured at different layers from the bed level [18].

16. **M.A Kamoji et. al** emphasised in their paper about aspect ratio. Aspect ratio (α) is the ratio of height of the rotor (H) divided by its diameter (D). This is very important criterion for the evaluation of the hydro kinetic turbine. Aspect ratio at 0.7 has maximum efficiency of hydro kinetic turbine. Aspect ratio is shown in Fig. 5. Helical rotor without shaft at an overlap ratio of 0.0 and an aspect ratio of 0.88 is found to have almost the same coefficient of power when compared with the conventional Savonius rotor.
$\alpha = H/D$ [19].

17. **R Gupta et. al** from their investigation observed that with the increase of overlap, the power coefficients decreased. The maximum power coefficient of 51% was obtained at no overlap condition. Study revealed that there is a definite improvement in the power coefficient for the combined Savonius–Darrieus rotor [18]. From the present investigation, it is observed that with the increase of overlap, the power coefficients decreased. The maximum power coefficient of 51% was obtained at no overlap condition. Study revealed that there is a definite improvement in the power coefficient for the combined Savonius–Darrieus rotor [20].

18. **K. Golecha et. al** also worked on blade arc angle in his research. Blade arc angle (ψ) is the angle between blade profiles. In their results, the modified Savonius rotor without overlap ratio, with blade arc angle of 124° and with an aspect ratio of 0.7 had a maximum power coefficient of 0.21 at Reynolds number of 150,000 [21].

19. **Fernando Ponta et al** explained the importance of channelling device in their work. The channelling device also acts as a flow speed amplifier with a high gain at low river current velocities and with a progressive reduction of gain when the river speed increases. Thus, the channelling device acts as a regulator, stabilizing flow speed in the neighbourhood of the rotor. Moreover, as the output power depends on the cube of flow velocity in the neighbourhood of the rotor, small variations of flow speed produce very significant fluctuations of generated power. Basically the channel section acts as a nozzle device which improves the speed of water at turbine section [22].

20. **Shujie Wang et. al** came to the conclusion from various experimental works carried by them that the hyperbola channelling device proved to be a kind of optimizing channelling device which has much more advantages such as less resistance effects and more fluency line shape and so on [23]. Same designs of channel shown in below fig.7.

21. **Ibrahim Mabrouki et. al** stated in their paper that Savonius rotor is simple in design and easy to fabricate at a lower cost. The basic driving force of Savonius rotor is drag. Hence, the advancing blade with concave side facing the water flow would experience more drag force than the returning blade, thus forcing the rotor to rotate. In this work, we are interested on the study of the effect of the Savonius rotor height on the hydrodynamic characterisations. In particular, we present a detailed description of the test section and the various manipulations made to develop the hydrodynamics characteristics of the water Savonius rotor such as power, torque and its coefficients [24].

22. **M. Goodarzi et. al** after their research work, two large Savonius turbine had been proposed to use near the radiators of a natural draft dry cooling tower as a substitute of formerly proposed solid windbreakers. A numerical procedure has been used to calculate the flow field unsteadily, and calculate the cooling improvement and power generation in turbines. Numerical results showed that rotating turbines could improve cooling capacity as the same order of solid windbreakers. It was surprisingly concluded that presence of cooling tower near Savonius turbine increased its power generation. Ultimately, it was concluded that overall improvement of the proposed arrangement was considerable from thermal and clean energy production viewpoints [25].

23. **Rajen Pudur et. al** in their paper, a grid connected hydrokinetic power generation scheme with asynchronous generator coupled to Savonius rotor is presented. The Savonius rotor acts as prime mover to the asynchronous generator and thus, the proposed scheme presents a system with variable input power similar to that of a wind energy conversion system. In order to mitigate the effect of changes in input power to the generator and

resulting variations in utility voltage and frequency which affects the system performance, an AC–DC–AC converter with a dc link capacitor is designed and modeled. The operation of the power converter is realized by decoupled d–q control method. A MATLAB/SIMULINK model of Savonius rotor, asynchronous generator, power converter, LCL filters and control schemes is presented. The proposed scheme is tested under various load conditions under varying input power and the performances are observed to be satisfactory [26].

24. **Yue Liu et. al** in their paper focuses on discussing the potential and feasibility of increasing hydropower production by installing hydrokinetic turbines behind existing conventional hydropower stations to establish "combined- cycle hydropower system (CCHS)". The CCHS will capture additional power from the energy remaining in water currents exiting dams. There are two modes of CCHS. The hydrokinetic turbine can be located directly behind the turbine of existing conventional hydropower plant or it can be placed at sites in the vicinity of powerhouse. The challenges and advantages associated with the CCHS are discussed in this paper. Although the technology of CCHS is still in its research and development phrase, not yet reaches mature and economically feasible; it is believed that it possesses significant potential to produce additional clean hydropower in the large-scale. It may become additional promising way of generating clean energy to mitigate climate change [27].

2.1 RESEARCH GAP

After going through the literature reviews it is found that as an emerging renewable energy solution, detailed study on hydrokinetic turbine systems is invaluable. Savonius turbines are mainly drag-driven devices made up of two or multiple blades. The difference in the drag forces experienced by the rotor blades makes the Savonius turbine to rotate. The efficiency of these turbines is very low as compared to lift based turbines. Hence a thorough study of all factors influencing their performance is imperative. The effect of different classes of augmentation channel on turbine performances is to be studied. Fluctuation in rotational speed is a common problem along with variations in torque. Most suitable orientation should be found to initiate turbine rotaion. Effect of turbulence and its variation along the domain of geometry is important to analyse. Computational analysis of flow characteristics around different types of Savonius rotor is required.

3. PROBLEM IDENTIFICATION, METHODOLOGY AND FORMULATION

3.1 PROBLEM STATEMENT

The objective of the study is to evaluate how the conventional Savonius hydrokinetic turbine performs when it rotates by the momentum of water current at low velocity from around 0.8-2m/s in an open water channel. An investigation with computational fluid dynamics (CFD) study using Ansys 15.0 has been carried out to bring about the objective of the work. To understand the implication of Savonius design in water application, the performance of the hydrokinetic turbine is found for two bladed turbine. The reason of using CFD is to facilitate a more thorough study on the velocity and torque distribution across the hydrokinetic turbine and hence to grow more insight of design information about its performance under low velocity condition. Finally the reason for enhanced performance of the hydrokinetic turbine is investigated from the computational study of flow characteristics of the hydrokinetic. Stable, smooth operation, and a better service life of the hydrokinetic turbine could be expected unlike the wind turbine.

3.2 Basic Terminologies

Tip speed ratio

As discussed above, tip speed ratio (λ), is the ratio of the speed of the blade, at its tip, to the speed of the flowing water and is considered one of the important parameter. It can be written as: $\lambda=\frac{\omega R}{2 \times V}$.

Blade arc angle

Blade arc angle (φ) is the angle between blade profiles.

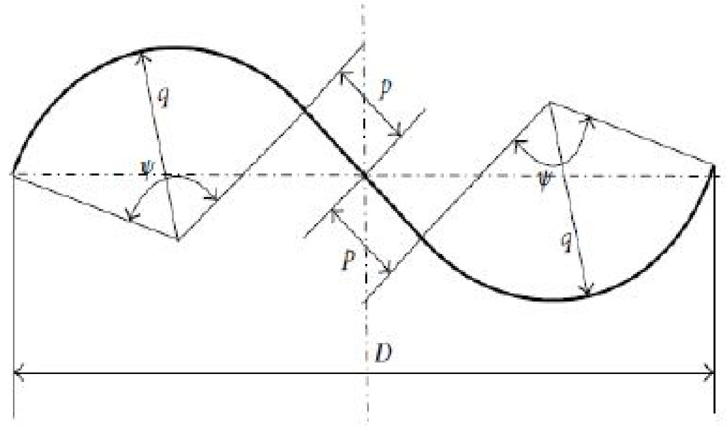


Fig.8 Blade Arc Angle

Blockage ratio

Blockage ratio (B) is defined as the ratio of the frontal swept area of the turbine to the frontal swept area of the channel. It can be written as: B = HD/Hw_w .

Overlap ratio

Overlap ratio (β), bucket overlap create effect on the efficiency of the turbine so it is also very important parameter. It is written as:$\beta = e/d$

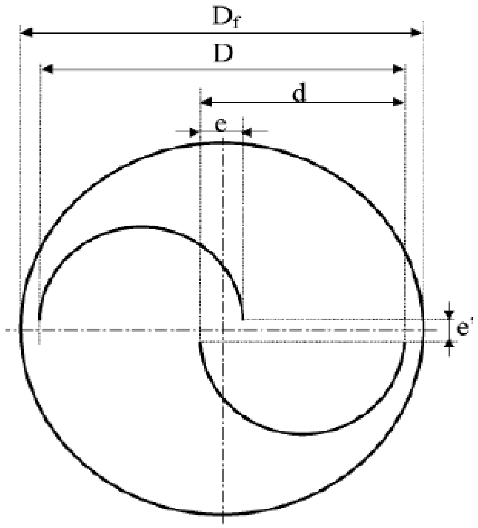


Fig.9 Overlap Ratio

Aspect ratio

Aspect ratio (α) is the ratio of height of the rotor (H) divided by its diameter (D). It is written as: $\alpha = H/D$.

3.3 NUMERICAL METHODOLOGY

3.3.1. CFD

Computational Fluid Dynamics (CFD) provides a qualitative (and sometimes even quantitative) prediction of fluid flows by means of mathematical modeling (partial differential equations), numerical methods (discretization and solution techniques) software tools (solvers, pre- and postprocessing utilities). CFD enables scientists and engineers to perform 'numerical experiments' (i.e. computer simulations) in a 'virtual flow laboratory'. Fluid (gas and liquid) flows are governed by partial differential equations which represent conservation laws for the mass, momentum, and energy. CFD gives an insight into flow patterns that are difficult, expensive or impossible to study using traditional (experimental) techniques. Since fluid (gas and liquid) flows are governed by partial differential equations which represent conservation laws for the mass, momentum, and energy, CFD uses the same equations.

The results of a CFD simulation are never 100% reliable because:

• The input data may involve too much guessing or imprecision

• The mathematical model of the problem at hand may be inadequate

• The accuracy of the results is limited by the available computing power

3.2.2. Experiments vs. Simulations:

As a rule, CFD does not replace the measurements completely but the amount of experimentation and the overall cost can be significantly reduced.

Table 3.

EXPERIMENTS	SIMULATIONS
expensive	cheap(er)
slow	fast(er)
sequential	parallel
single-purpose	multiple-purpose

ABOUT CFD IN FLUENT

FLUENT solvers are based on the finite volume method. Domain is discretized into a finite set of control volumes (or cells). General conservation (transport) equations for momentum, mass, species, energy, etc. are solved on this set of control volumes.

$$\underbrace{\frac{\partial}{\partial t}\int_V \rho\phi\, dV}_{\text{Unsteady}} + \underbrace{\oint_A \rho\phi\, \mathbf{V}\cdot d\mathbf{A}}_{\text{Convection}} = \underbrace{\oint_A \Gamma\nabla\phi\cdot d\mathbf{A}}_{\text{Diffusion}} + \underbrace{\int_V S_\phi\, dV}_{\text{Generation}}$$

Partial differential equations are discretized into a system of algebraic equations. z All algebraic equations are then solved numerically to render the solution field.

Equation	Variable
Continuity	1
X momentum	u
Y momentum	v
Z momentum	w
Energy	h

Table3 CFD Analysis – The Basic Steps

Problem Identification and Preprocessing	1.Define your modeling goals 2. Identify the domain you will model 3.Design and create the grid.
Solver Execution	4. Set up the numerical model. 5. Compute and monitor the solution.
Post-Processing	6. Examine the results. 7. Consider revisions to the model

3.3.2 EQUATIONS USED IN CFD

Incompressible Navier-Stokes Equations:

$$\frac{\partial \mathbf{u}}{\partial t} = -\mathbf{u}\cdot\nabla\mathbf{u} + \alpha\nabla^2\mathbf{u} - \frac{\nabla p}{\rho} \qquad \mathbf{u} = \begin{pmatrix} u \\ v \\ w \end{pmatrix}$$

$$\nabla\cdot\mathbf{u} = 0$$

[31]

The (hydrodynamic) pressure is decoupled from the rest of the solution variables. Physically, it is the pressure that drives the flow, but in practice pressure is solved such that the incompressibility condition is satisfied. The system of ordinary differential equations (ODE's) are transformed to a system of differential-algebraic equations (DAE's), where algebraic equations behaves like a constraint.

By adapting the law of Navier-Stokes model of rotation frame, the equations governing the behavior Savonious hydrokinetics turbines will be used in this study. The equations rule the behavior of fluid flow including conservation of mass and momentum equations (eq. 6 & 7). Two types of acceleration in the momentum equation representing two Savonious hydrokinetics turbine rotation are the Coriolis acceleration, ($2\tilde{\omega}$ x $\bar{\upsilon}r$) and the centripetal acceleration ($\tilde{\omega}$ x $\tilde{\omega}$ x $\hat{r}$).

$$\frac{\partial \rho}{\partial t} + \nabla . \rho \bar{\upsilon}_r = 0$$

$$\frac{\partial}{\partial t} (\rho \bar{\upsilon}_r) + \nabla . (\rho \bar{\upsilon}_r . \bar{\upsilon}_r) + \rho (2\tilde{\omega} \times \bar{\upsilon}_r + \tilde{\omega} \times \tilde{\omega} \times \hat{r}) = -\nabla p + \nabla \tau + F$$

3.3.3 TURBULENCE MODEL

In this study, the Standard k-ε turbulence model with enhanced wall conditions was used. All three momentum equations x, y and z components of velocity, turbulent

kinetic energy (k) and dissipation rate of turbulent kinetic energy (ε) have been solved with the help of the fluent solver. The standard k-ε equations can be expressed as two equations- [32]

For turbulent kinetic energy k

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + P_b - \rho\epsilon - Y_M + S_k \quad (1)$$

For dissipation ϵ

$$\frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_i}(\rho \epsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x_j}\right] + C_{1\epsilon}\frac{\epsilon}{k}(P_k + C_{3\epsilon}P_b) - C_{2\epsilon}\rho\frac{\epsilon^2}{k} + S_\epsilon \quad (2)$$

In this case, p is the static pressure, τ is the stress tensor, F represents the external body force, $\hat{r}$ is the radial position of the rotating domain forms, $\tilde{\omega}$ domain is the angular velocity of the rotor, and $\bar{\upsilon}r$ is the relative velocity. Implementing a combined formulation in Cartesian and cylindrical coordinates, so that simulate two separate regions of the domain, i.e. inflows rotate the rotor-stator and external. Standard K-ε model, Eq. (8) and (9), is used to simulate the turbulence in the flow field [23]. It is a widely used and provides satisfactory correctness and praiseworthy to represent the various types of flow. They are coupled to the Navier-Stokes equations although included in the convection zone. K-ε model is a two-equation models involving turbulent kinetic energy, k, and the dissipation rate, ε, as follows, In this model, Gk represents the generation of turbulence kinetic energy due to the velocity gradient, whereas Gb describe the generation of turbulence kinetic energy due to buoyancy, and YM is the contribution of the fluctuating dilatation to the overall dissipation rate. Variable σk and σε are Prandtl numbers for the turbulent with value of k =1.0 and ε = 1.3. Constants C1ε = 1.44 and C2ε = 1.92. This equation will be used to predict fluid flow through a turbine that will improve its performance.

SST k-ω turbulence model

For turbulent kinetic energy k

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + P_b - \rho\epsilon - Y_M + S_k$$

For specific dissipation rate ω

$$\frac{\partial \omega}{\partial t} + U_j\frac{\partial \omega}{\partial x_j} = \alpha S^2 - \beta\omega^2 + \frac{\partial}{\partial x_j}\left[(\nu + \sigma_\omega \nu_T)\frac{\partial \omega}{\partial x_j}\right] + 2(1 - F_1)\sigma_{\omega 2}\frac{1}{\omega}\frac{\partial k}{\partial x_i}\frac{\partial \omega}{\partial x_i}$$

The SST k-ω turbulence model [33] is a two-equation eddy-viscosity model which has become very popular. The shear stress transport (SST) formulation combines the best of two worlds. The use of a k-ω formulation in the inner parts of the boundary layer makes the model directly usable all the way down to the wall through the viscous sub-layer, hence the SST k-ω model can be used as a Low-Re turbulence model without any extra damping functions. The SST formulation also switches to a k-ε behaviour in the free-stream and thereby avoids the common k-ω problem that the model is too sensitive to the inlet free-stream turbulence properties. Authors who use the SST k-ω model often merit it for its good behaviour in adverse pressure gradients and separating flow. The SST k-ω model does produce a bit too large turbulence levels in regions with large normal strain, like stagnation regions and regions with strong acceleration. This tendency is much less pronounced than with a normal k-ε model though.

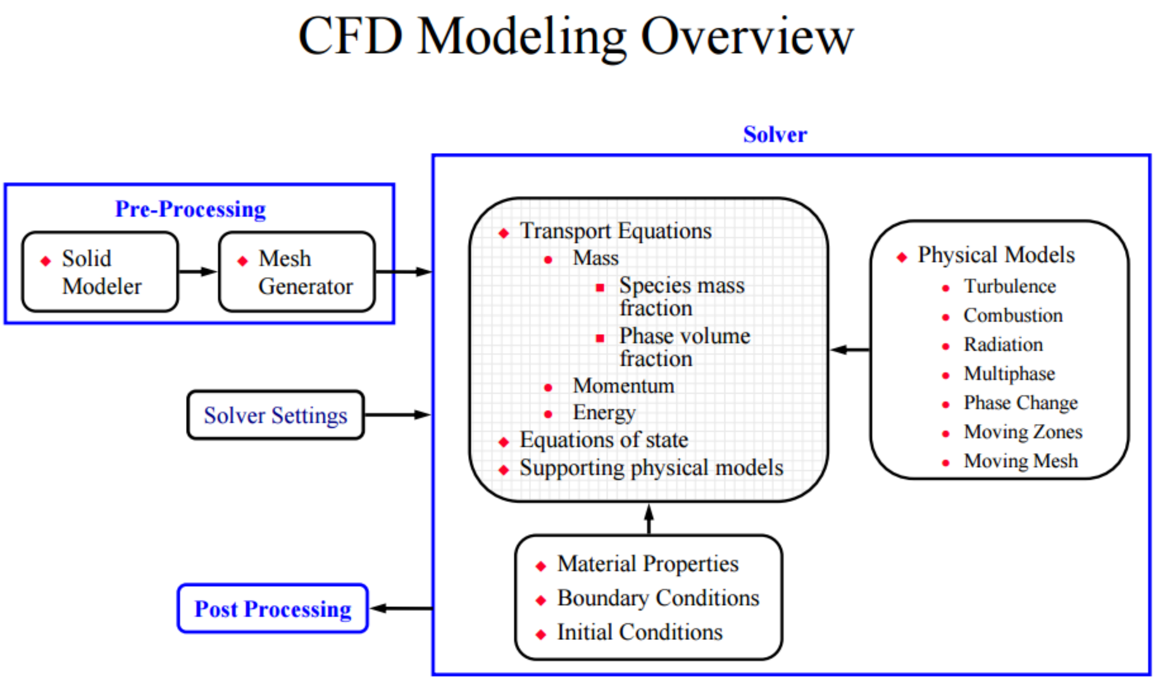


Fig.10.Modelling-Overview

3.3.4.COMPUTATIONAL MODELLING OF SAVONIUS WATER TURBINE

The two bladed simple Savonius water turbine with semi-circular cup shaped blades which has a blade diameter of 0.266 mm, height of 0.17 mm, which is designed in SOLID WORKS. Another blade of dia 0.45m , and height 0.315 m is taken for 2D analysis.

The blade material for modelling purposes was selected to be aluminium with thickness 1.5mm. The ratio H/D is taken as 0.7, which is known as aspect ratio.

3.3.5 PRE-PROCESSING OF THE MODEL

The Savonius water turbine, in practice, can be operated in a flow condition where the water velocity is significantly less along with a very low head. Such kind of situations exists in shallow river or other small running water bodies. The geometry of the Savonius water turbine has been designed computationally in SOLID WORKS(14), in which the rotor lies within a computational domain forming an open channel with side walls and open top. The computational domain has been fabricated in such a manner that the entire domain is segmented into two parts separating outer and the inner fluid zone. The inner fluid zone surrounding the rotor has been designed so that it allows the rotor to rotate during the simulation process and also the zone facilitates a detail study of the fluid behaviour around the subject of interest. The computational domain has been designed with a length of 7 times the diameter i.e. 3.15m and width of 3 times i.e. 1.35m. The inlet and exit faces are kept away on both sides by a distance which is more than three times rotor diameter, respectively. The entire domain has been designed with a height of 0.945 m with the inner fluid zone having a diameter and height 0.54 *m* and 0.378 *m* respectively such that it completely encloses the rotor. The created geometry has been meshed with a suitable grid in ICEM CFD. The modelled geometry is quite complex for which the tetrahedron mesh is chosen for 3D while trianglular mesh for 2D because of its capability to mesh complex geometries. The cylindrical bodies are meshed with more quality and accuracy by adopting finer mesh. However the purpose of both tetrahedron and trianglular mesh are done on ICEM CFD. A two-dimensional view of the rotor model was considered. It is because the rotor blade rotate in the same plane as the approaching water flow stream. The computational domain was discretisized using two dimensional unstructured mesh (triangular mesh). The left boundary had Velocity Inlet condition while the right boundary had Outflow condition. The top and bottom boundaries for the open channel sidewalls had symmetry conditions .Prism layer is formed around the blade for more accuracy. Also, the inner fluid zone is named as deforming fluid, given more fine meshing, while the other parts are kept little low quality mesh as compared to inner fluid zone and prism layers. Figure 6 shows the mesh of the created geometry and the inner mesh as well with the help of section plane so as to study the inner mesh. Various named selections have been created to define Savonius water turbine.

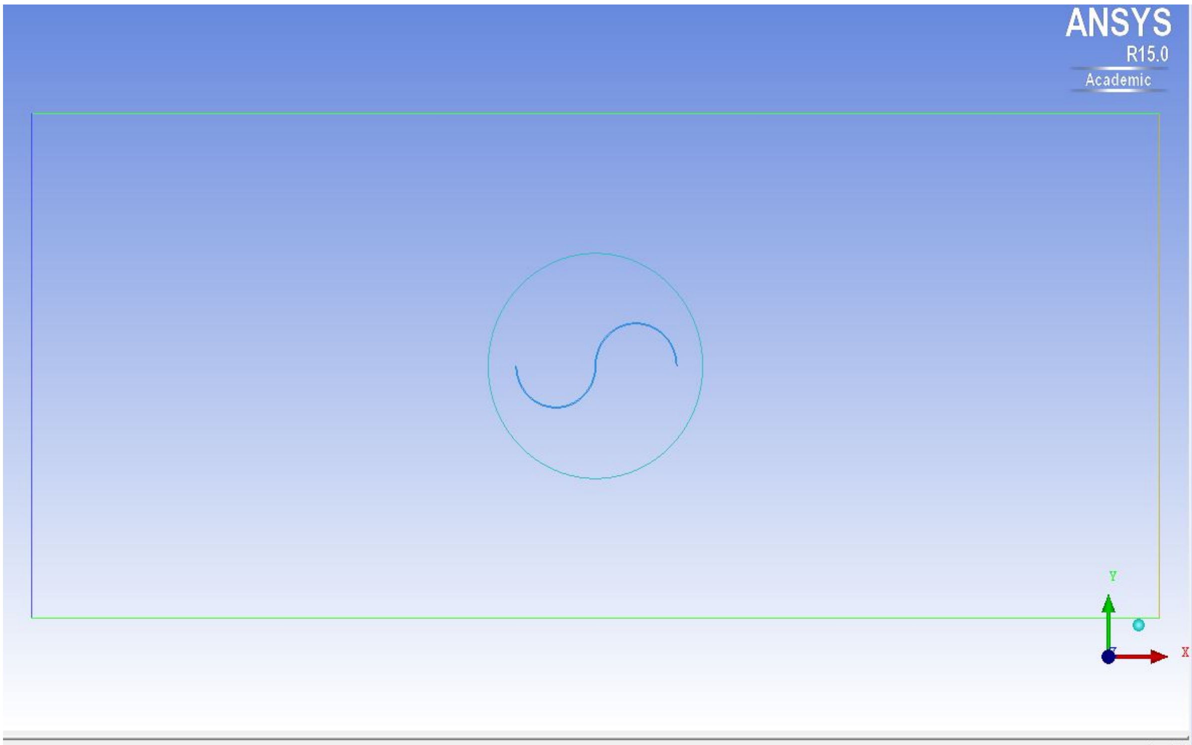


Fig. 11 2D geometry

3.3.6 CFD Formulation of the Model

For a CFD formulation the contacts must be defined. The slicing operation which is done with the surface of the inner fluid zone makes a surface to surface connection between the inner fluid and the outer fluid zone. Two interface-zones have been formed by creating two more named selections for the outer surface of the inner fluid zone (interface-inner) and for the in-contact surface of the outer fluid zones (interface-outer).

The present study involves fluid flow through vicinity of the complex geometries. Therefore standard k-ε model has been adopted that solves two separate transport equations and allows the independent determination of the turbulent kinetic energy (k) and its dissipation rate ($\in$) [28-30]. Second order upwind interpolation schemes is chosen for both turbulence momentum and to obtain results with more accuracy. The left side of the computational domain from where the water flow is projected onto the model, has been assigned with the boundary condition of 'velocity inlet' with pressure lying in the atmospheric level. The side walls and the bed layer of the domain are set to 'wall' by default and the open top is set as 'symmetry'. The Savonius water turbine placed inside the domain is assigned with 'moving wall' with rotational motion. As the turbine rotates inside the computational zone, a single frame of rotation has been used for the computational analysis. For the computational study, the same water velocities of 0.8 *m/s*, 1.0 *m/s* have been considered.

A tabulated form of the steps involved during simulation is as follows:

1) Solver:

• Solver: Pressure Based

• Time: Transient in 1st and Steady in 2nd cases

• Space: 2D in 1st case and 3D in 2nd cases

2) Viscous Model:

• Model: Standard k-epsilon (k-ε)

• Near-Wall Treatment: Standard Wall Functions

3) Material:

• Water (ρ= 1000 kg/m3)

4) Operating conditions:

• Atmospheric Pressure (1.0132 bar)

5) Boundary Conditions:

• Inlet: Velocity Inlet

• Sides: No slip wall

• Blades: Stationary Wall in 1st case and

• Outlet: Pressure Outlet

6) Solution controls:

• Pressure Velocity Coupling: SIMPLE

• Discretization: fluids

7) Pressure (Standard)

• Inlet Velocity: (a) 0.8m/s, (b) 1m/s for transient cases

(a) 0.35m/s (b) 0.65m/s (c) 0.9m/s for steady cases

3.3.7 MESH GENERATION

The mesh is generated in ICEM CFD

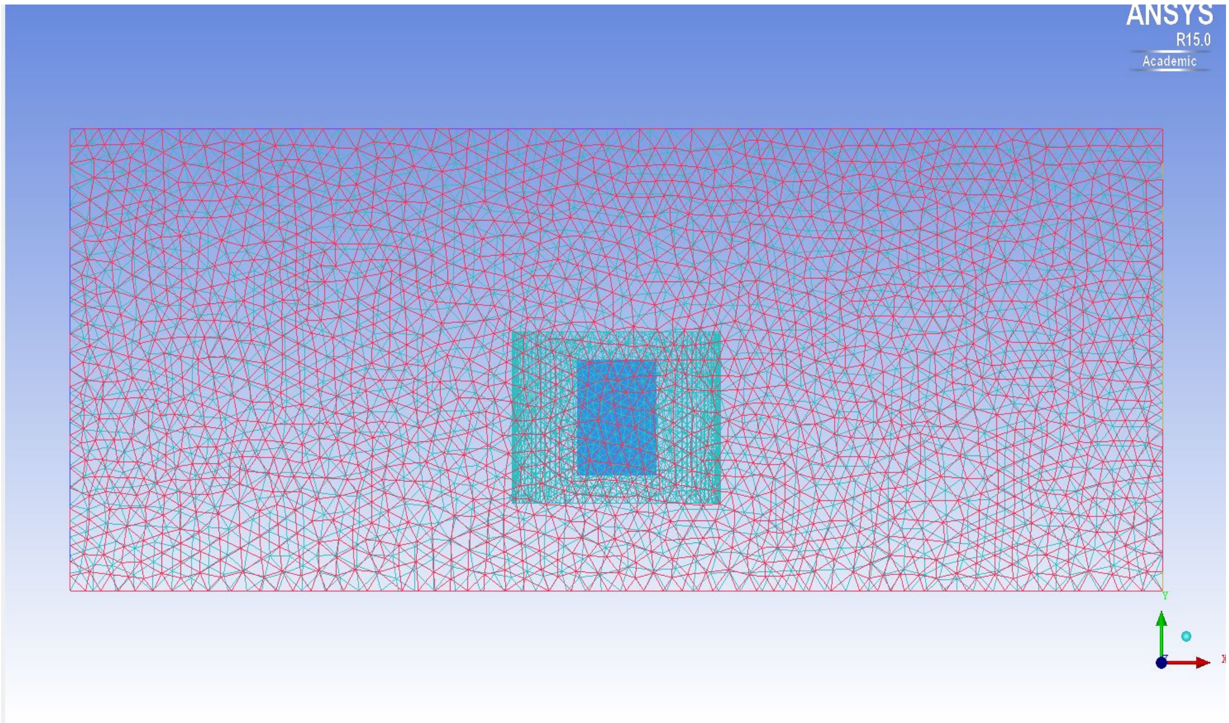


Fig.12 Meshing of overall geometry

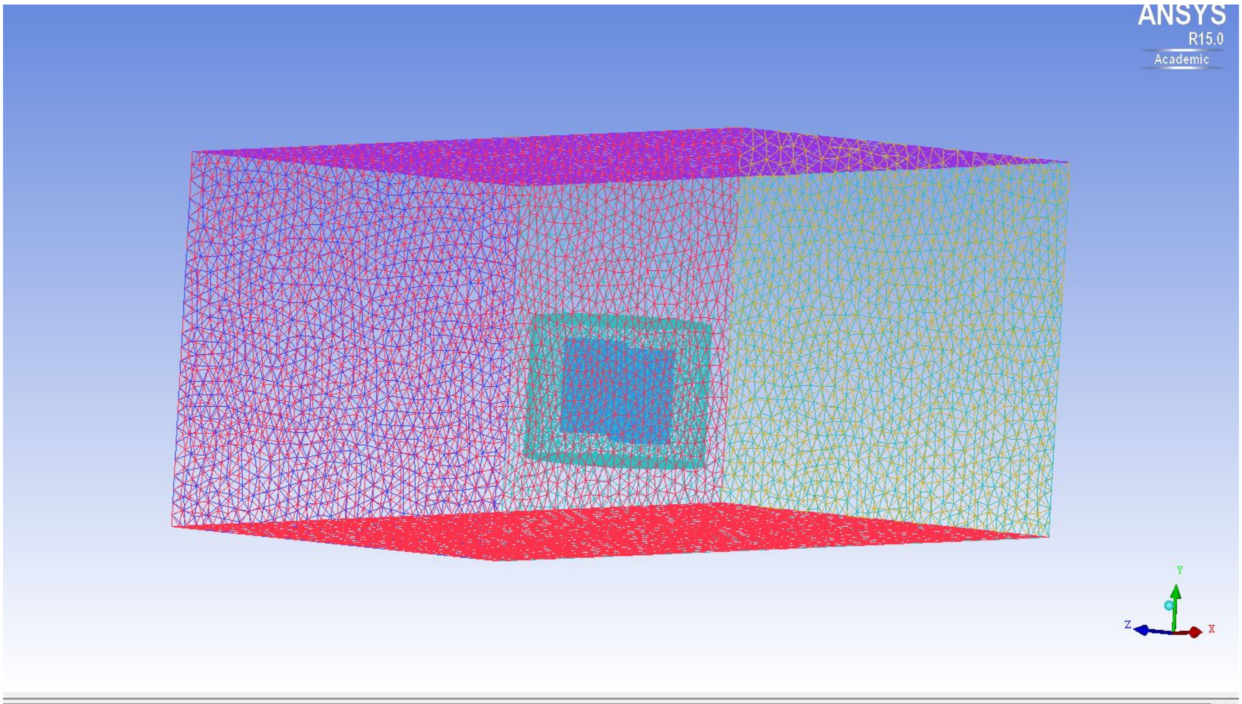


Fig. 13 View of mesh1

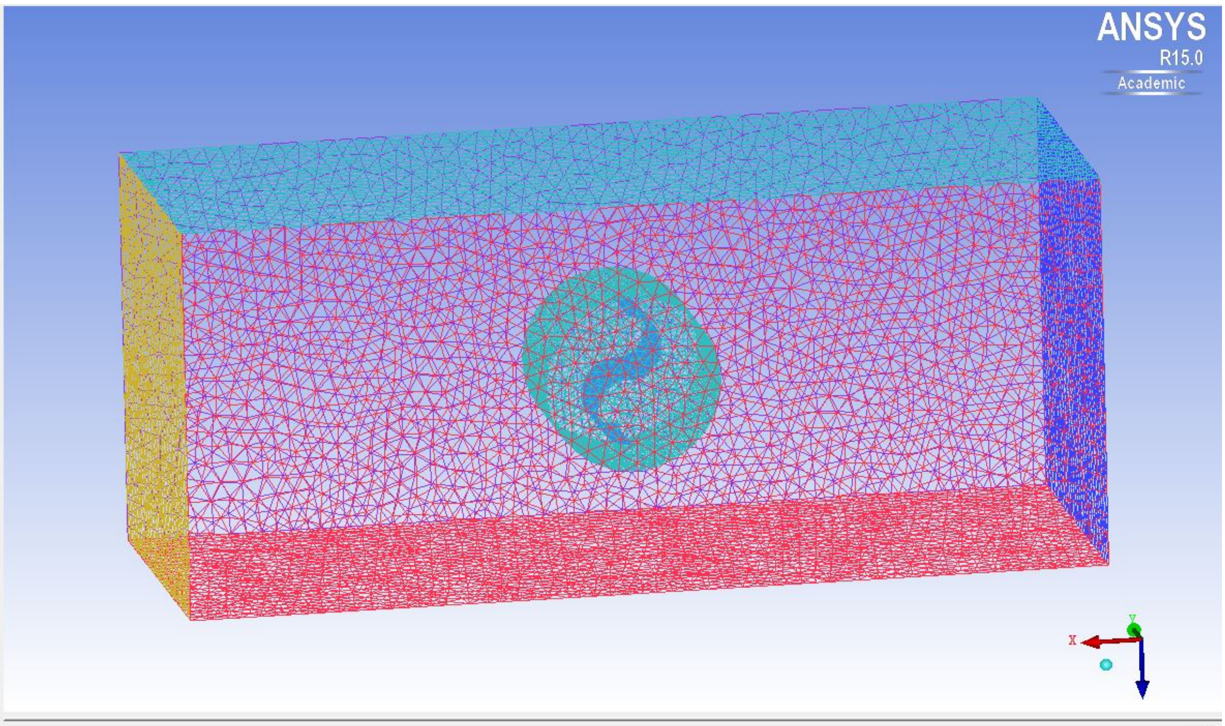


Fig.14 View of mesh2

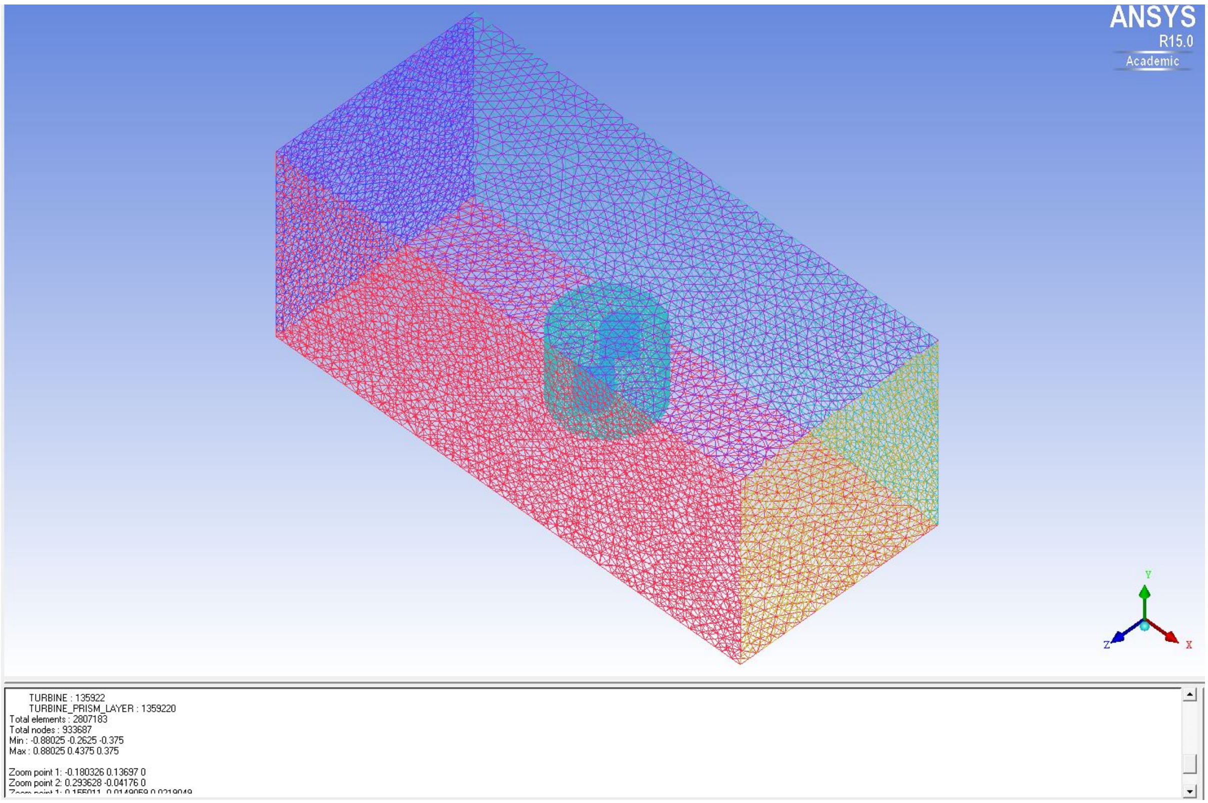


Fig.15 Mesh of turbine for 3D

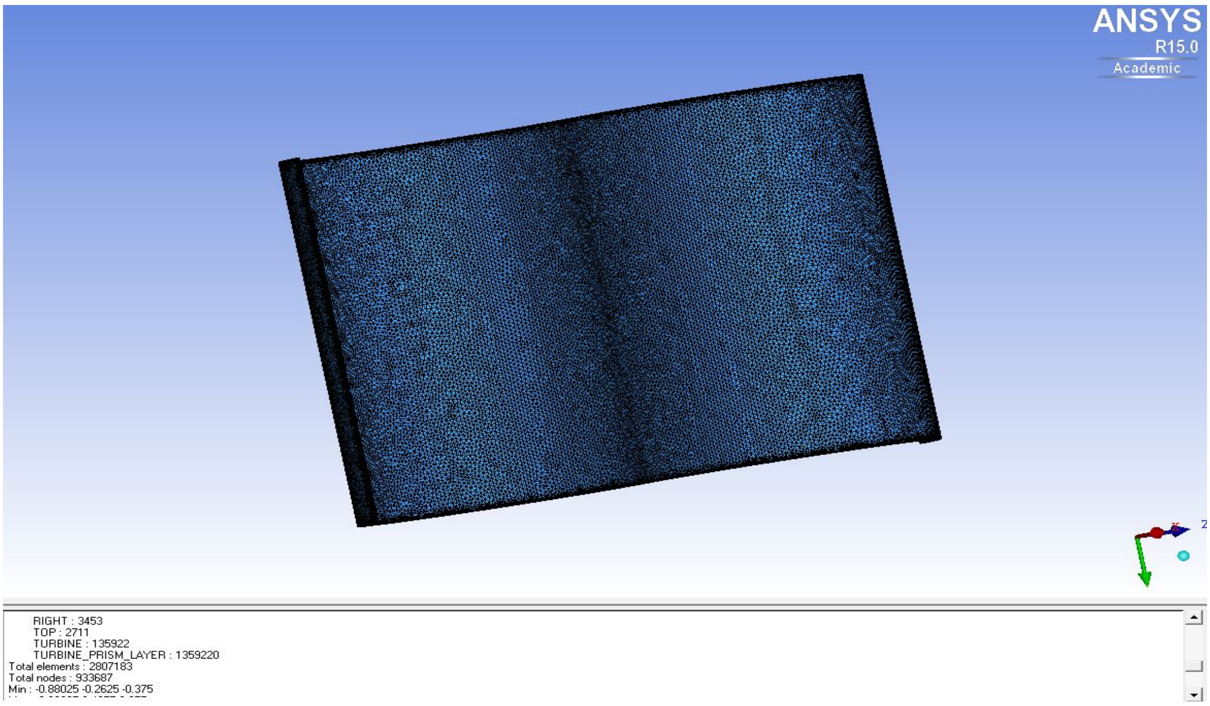


Fig. 16 Meshing of turbine blade

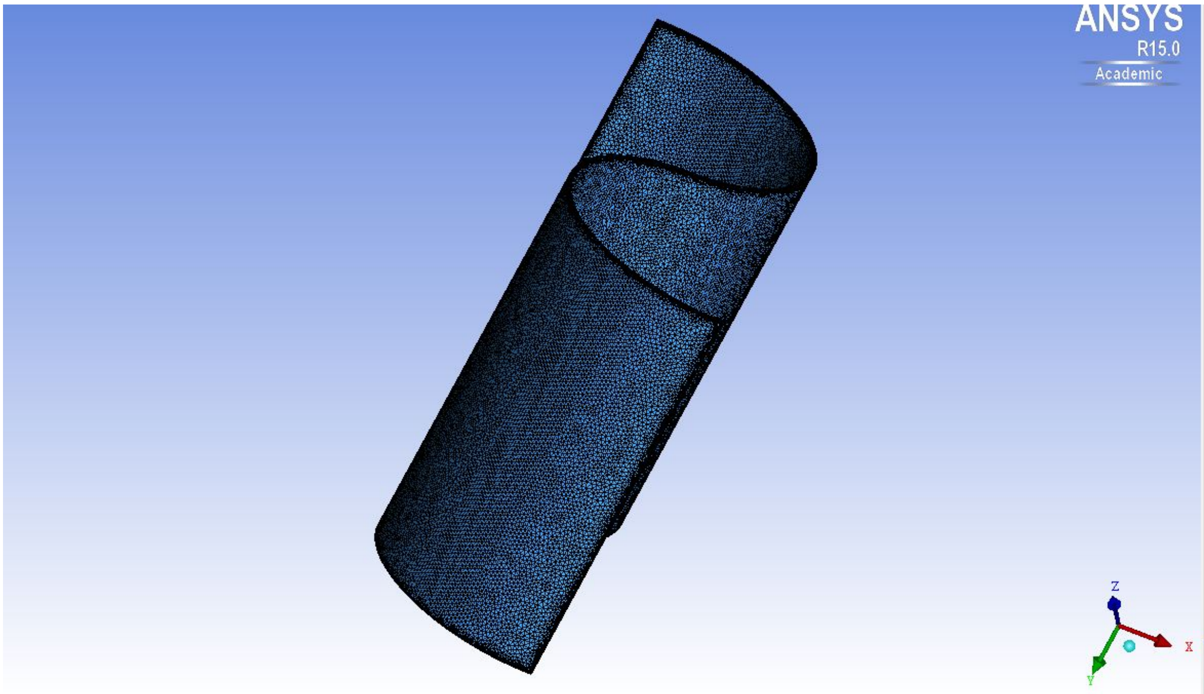


Fig.17 View of turbine mesh1

Then mesh for 2D is also generated

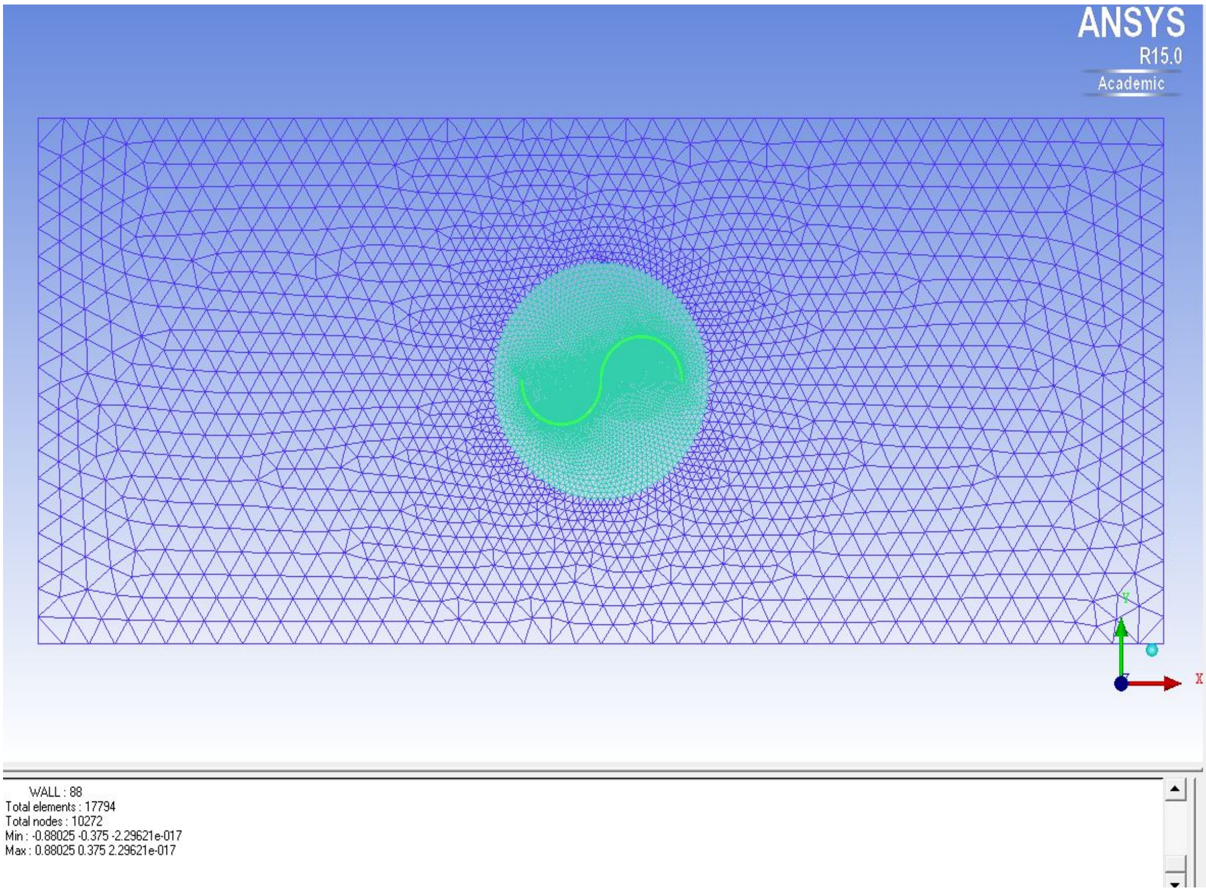


Fig.18 Mesh of 2D

The total number of elements created in the mesh for 2D is 17794, while total number of nodes is equal to 10272.

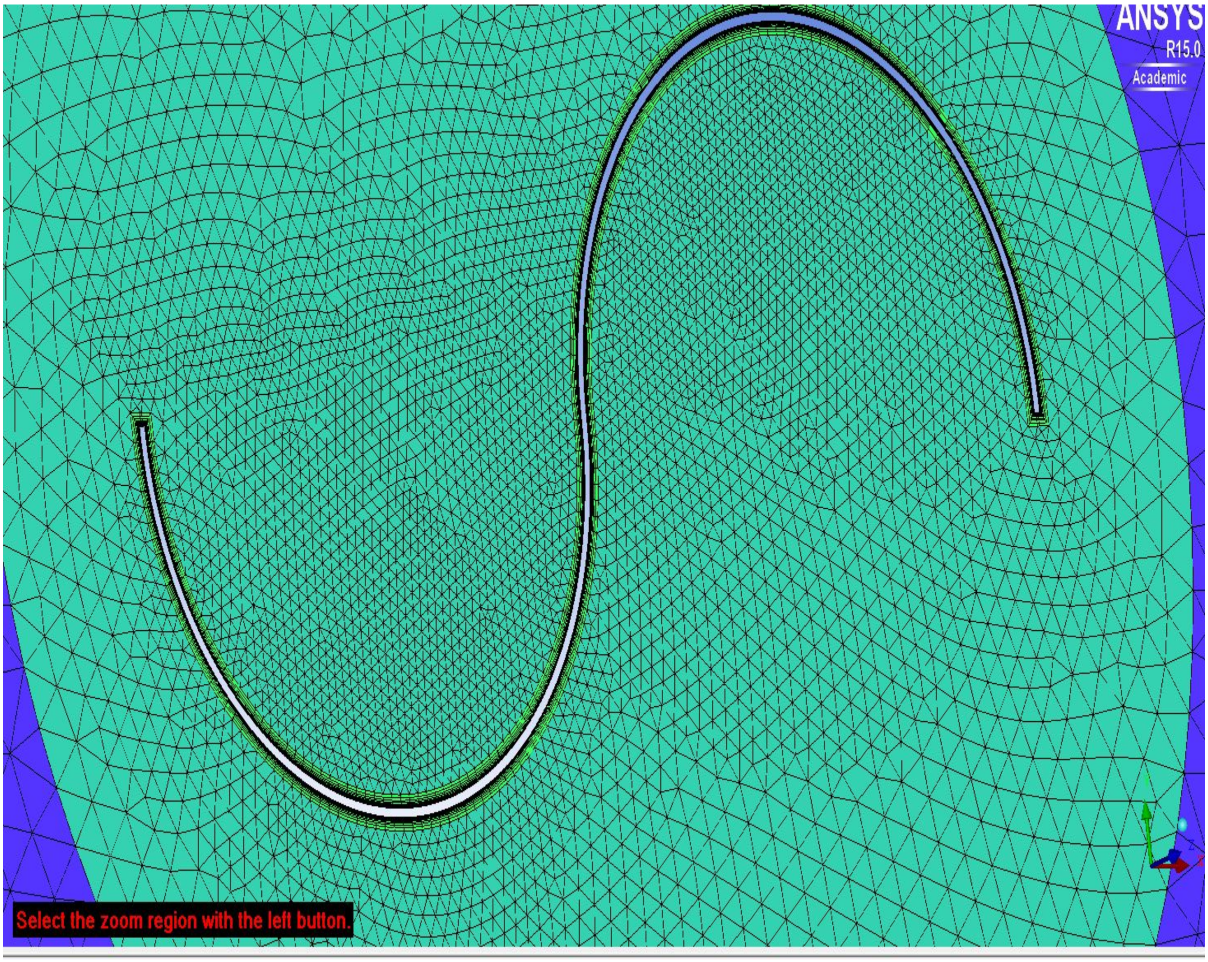


Fig.19 Close view of mesh near turbine

3.3.8.SolutionSetup

For transient analysis :

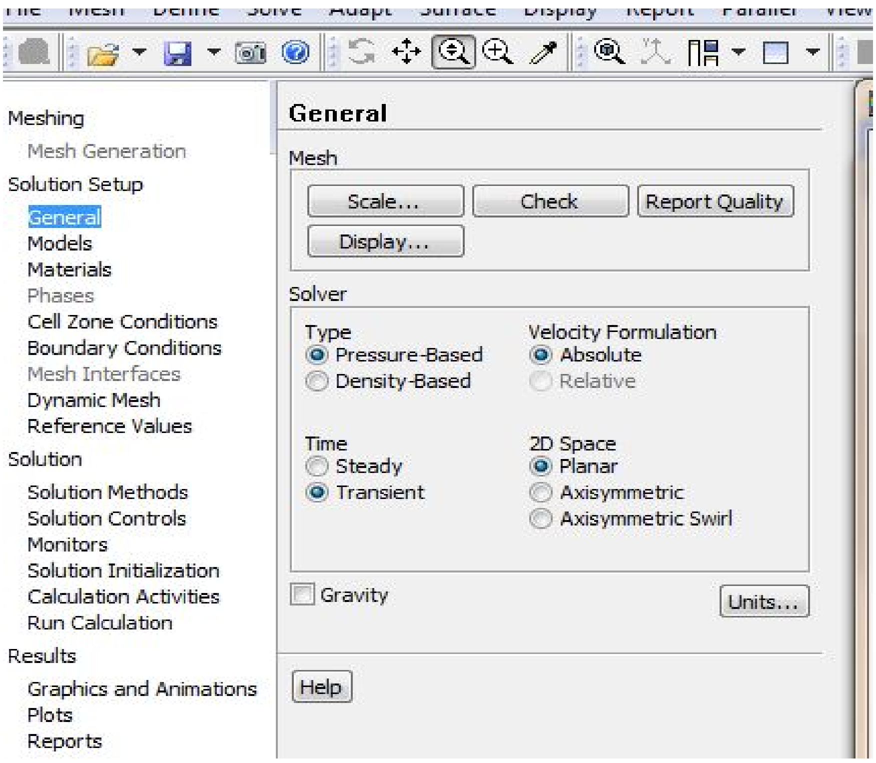


Fig.20 General Setup

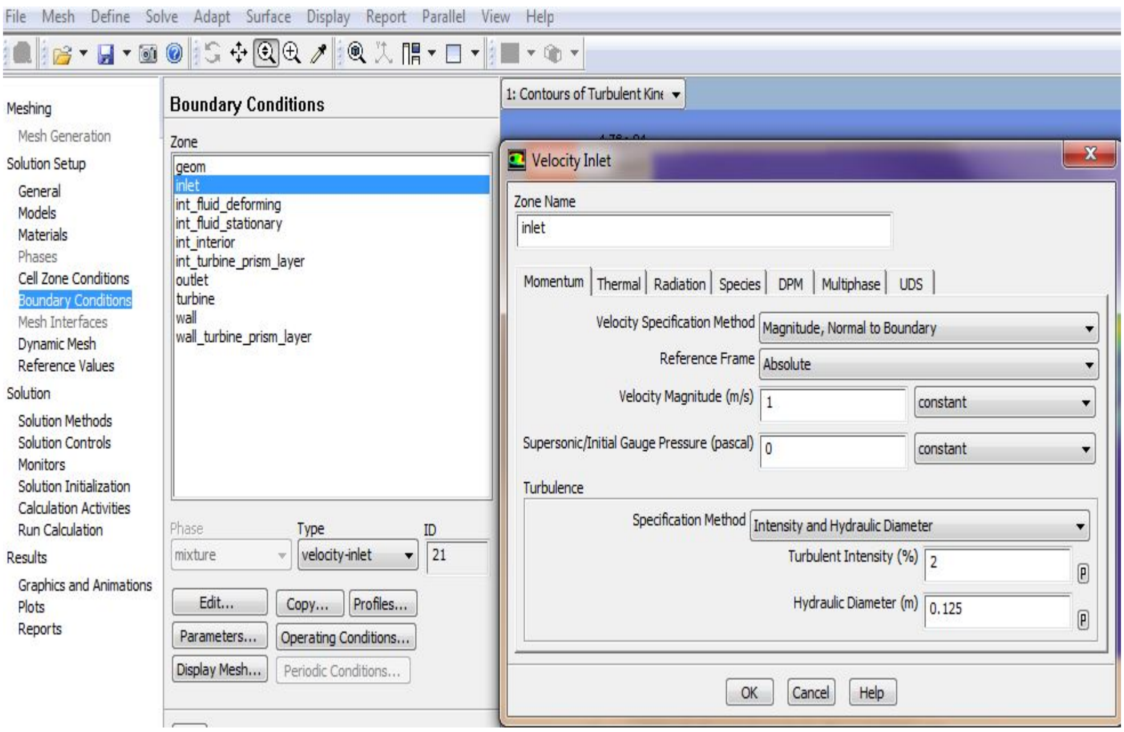


Fig.21 Boundary Condition

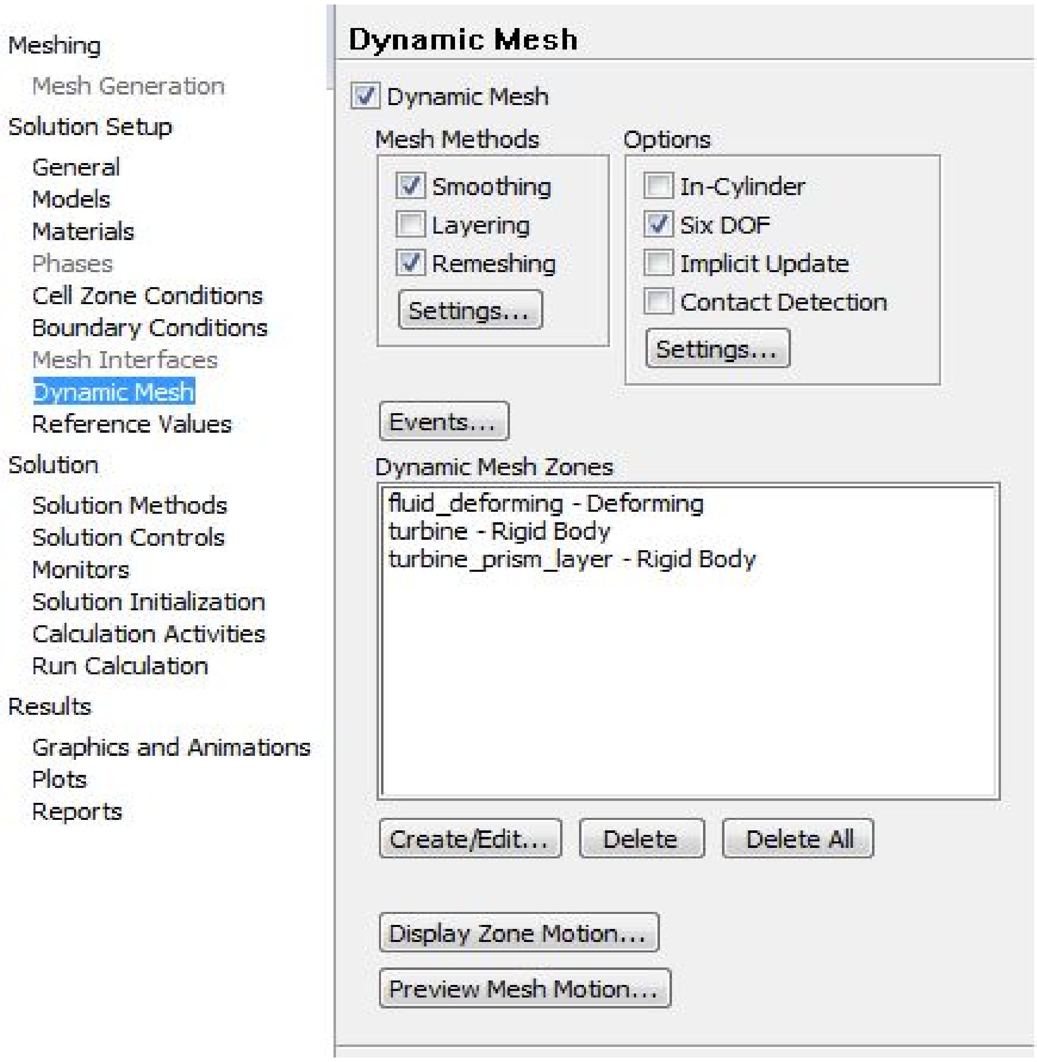


Fig.22 Dynamic Mesh

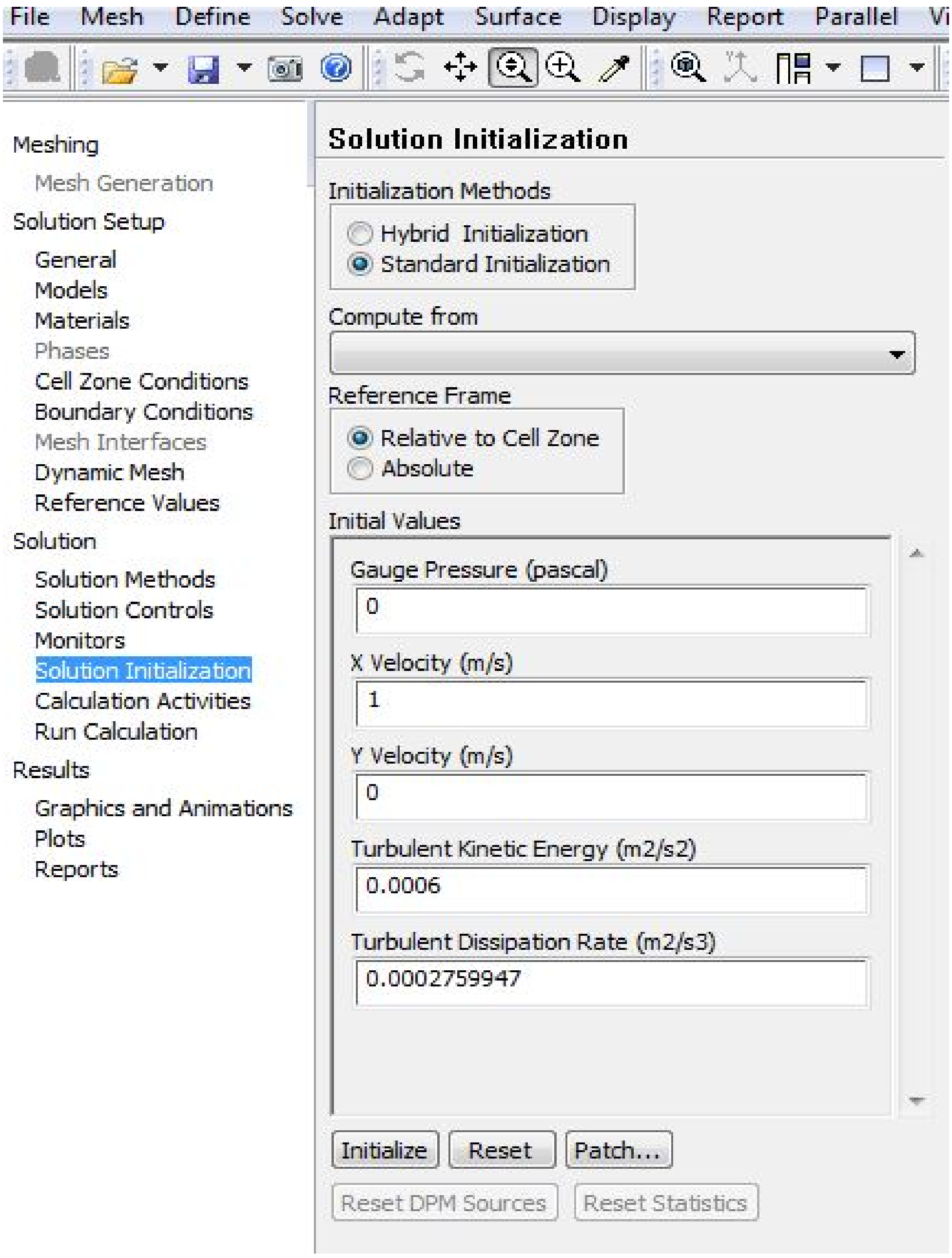


Fig.23 Solution Initialization

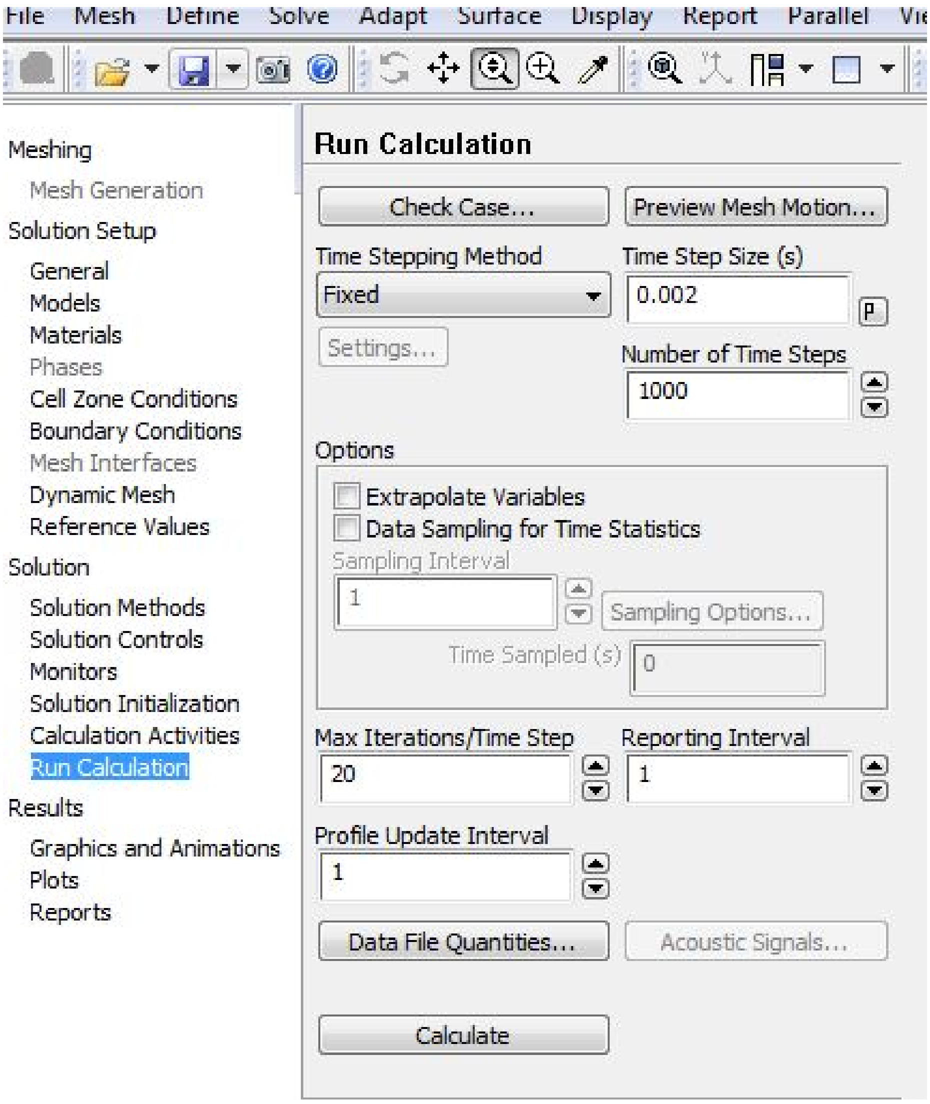


Fig.24 Run Calculation

For Steady Analysis

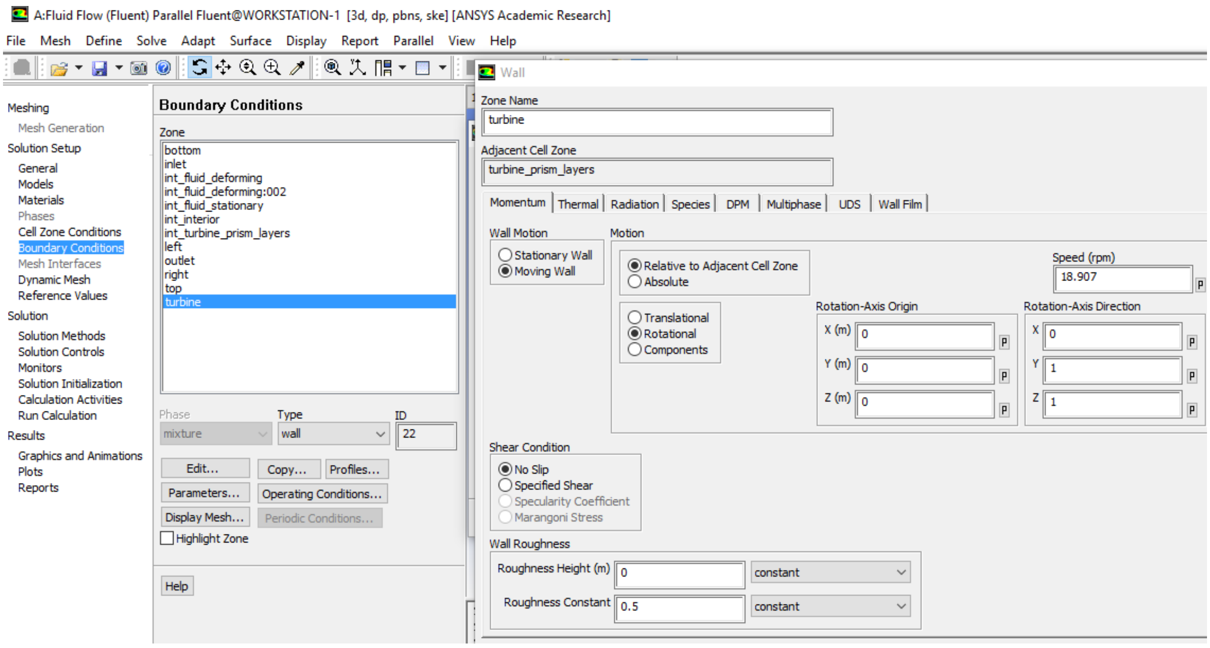


Fig.25 Boundary Condition for turbine

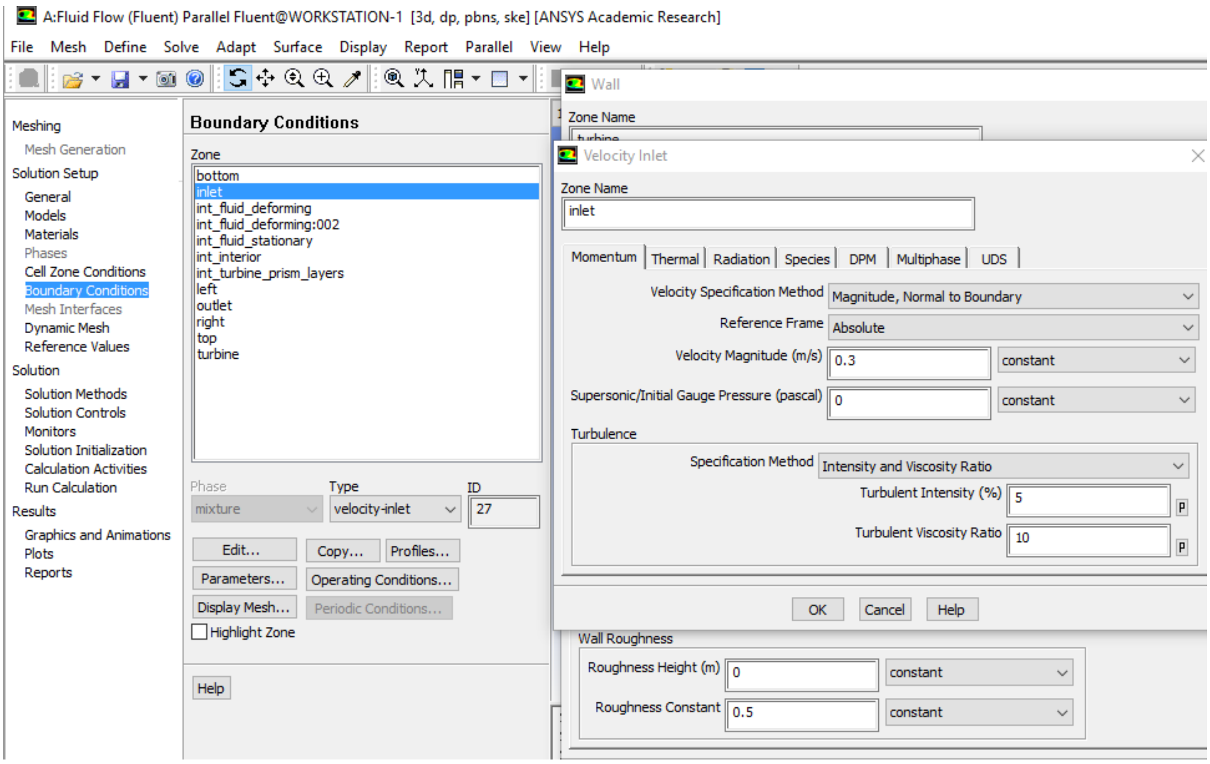


Fig.26 Boundary Condition-inlet

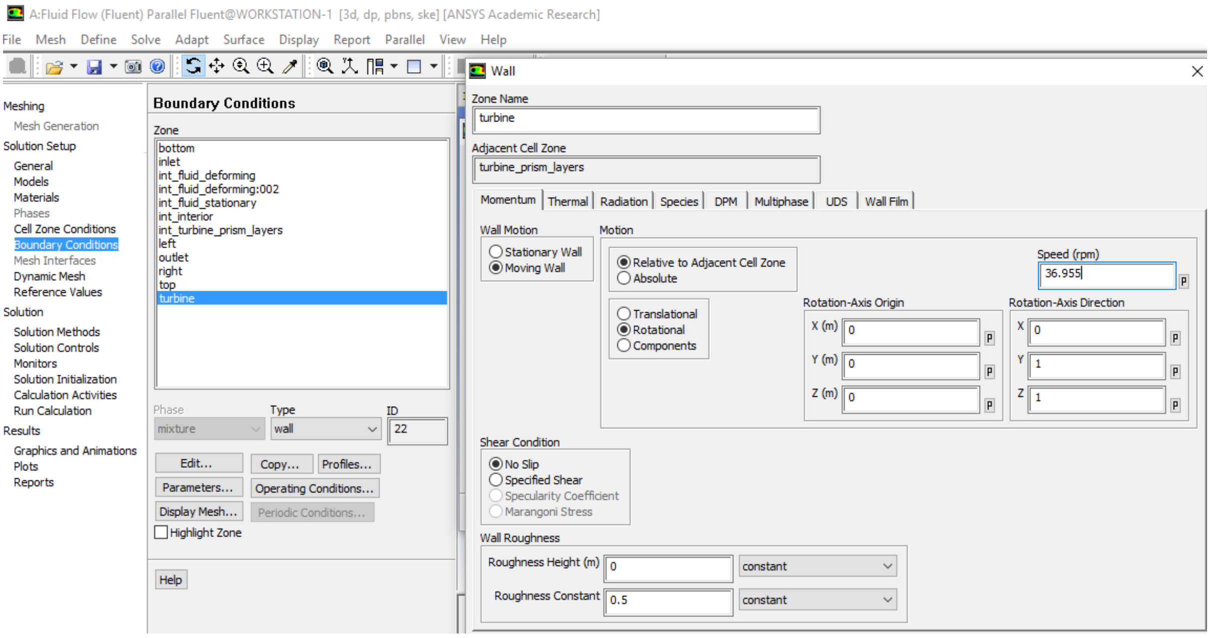


Fig.27 Boundary Condition for turbine

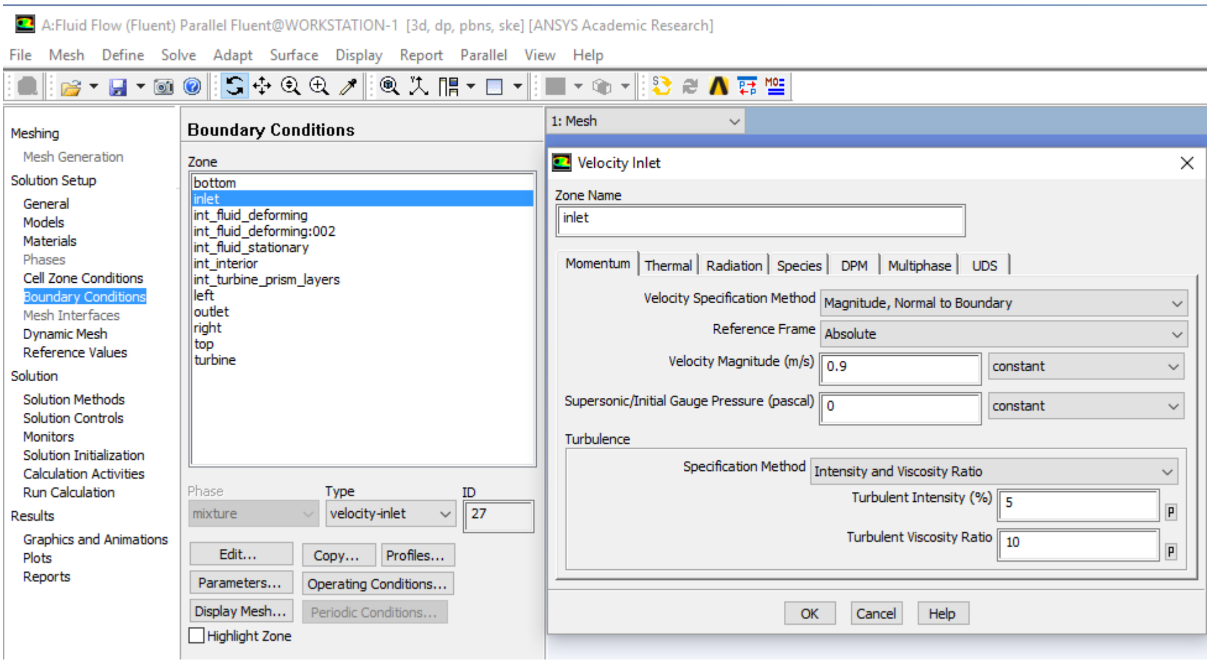


Fig.28 Boundary Condition for inlet

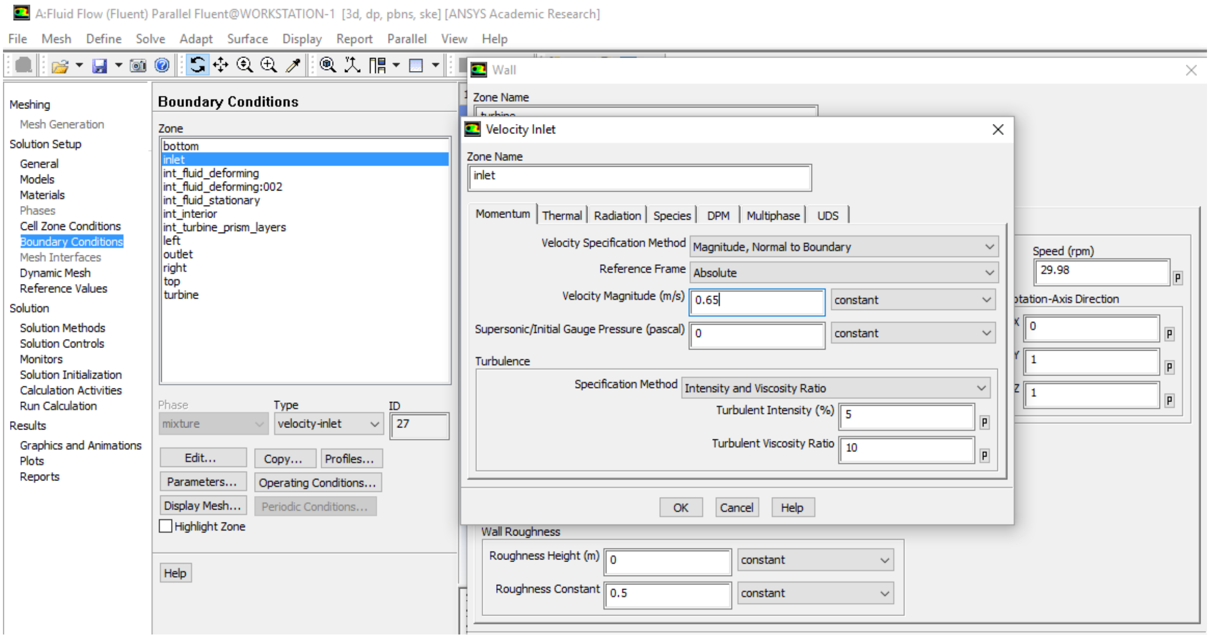


Fig.29 Boundary Condition for inlet

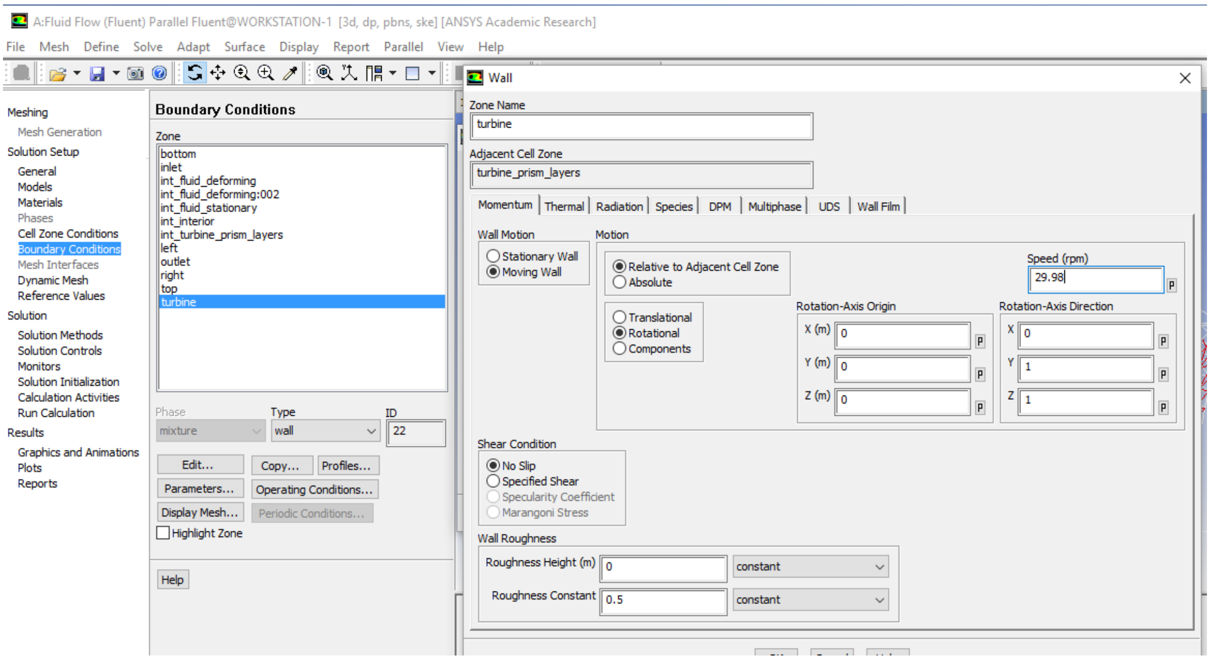


Fig.30 Boundary Condition for turbine

Moreover, a UDF (User Defined Function) is fed in the fluent file of different geometries at different conditions for transient analysis.

The UDF is of the format:

```
#include "udf.h"
#include "math.h"

DEFINE_SDOF_PROPERTIES(turbine_6dof, prop, dt, time, dtime)
{
prop[SDOF_MASS] = ;


prop[SDOF_IZZ] = ;

prop[SDOF_ZERO_TRANS_X] = TRUE;
prop[SDOF_ZERO_TRANS_Y] = TRUE;
prop[SDOF_ZERO_TRANS_Z] = TRUE;
prop[SDOF_ZERO_ROT_X] = TRUE;
prop[SDOF_ZERO_ROT_Y] = TRUE;
prop[SDOF_ZERO_ROT_Z] = FALSE;
}
```

Under Dynamic Mesh, six DOF is applied to get the motion of the turbine at different time intrervals.

4.1 RESULTS

At the start of turbine, static analysis of the blade under the above mentioned conditions is done. The static pressure contour of the turbine blade,as well as entire geometry is shown as:

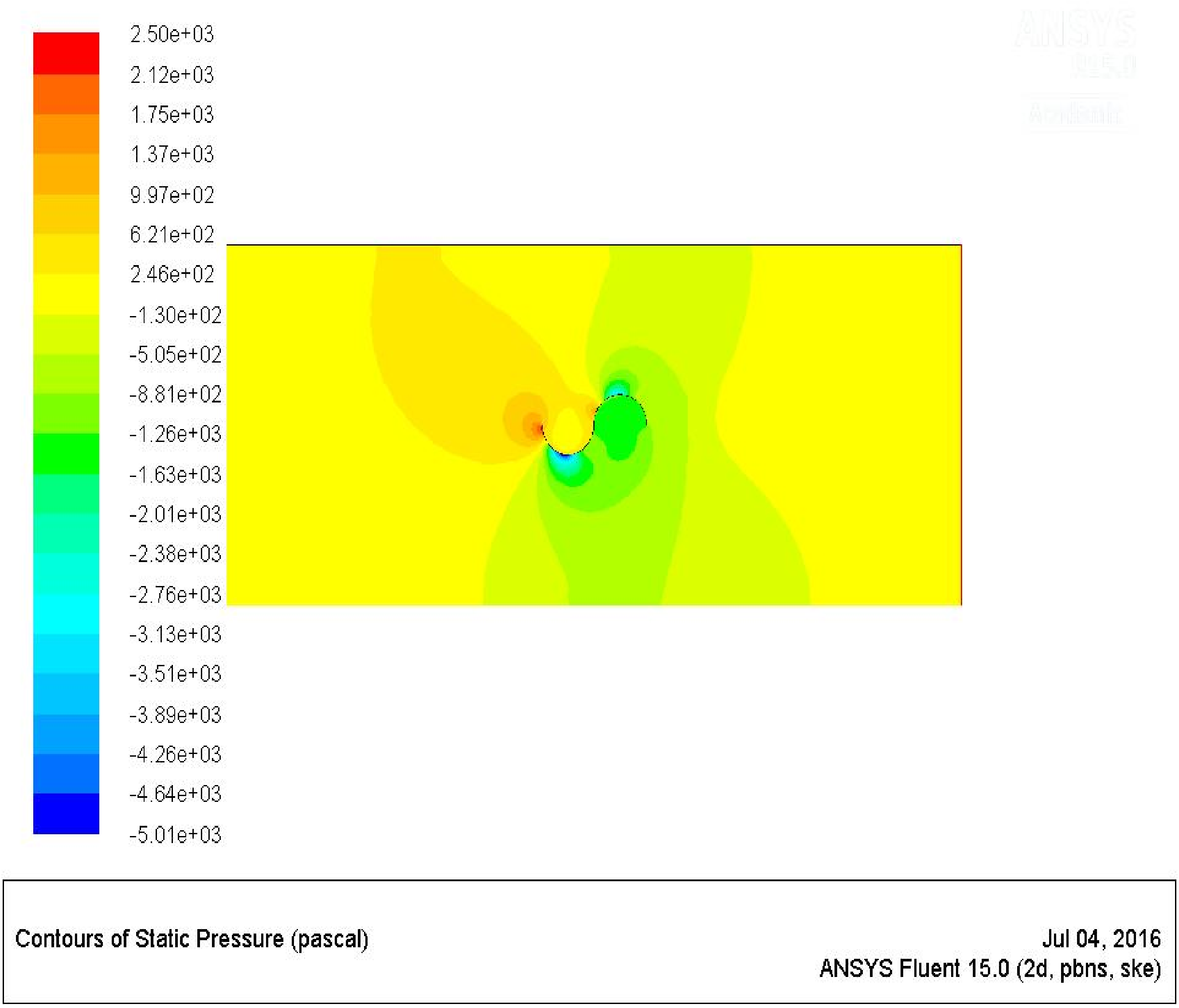
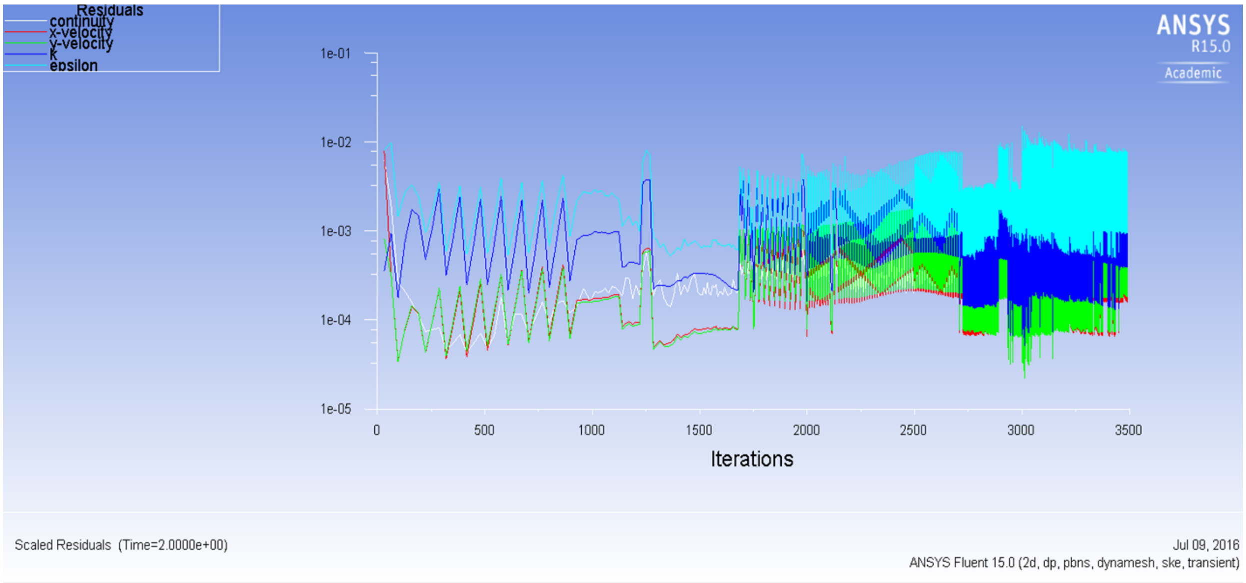


Fig.31 Contour of static pressure at the beginning



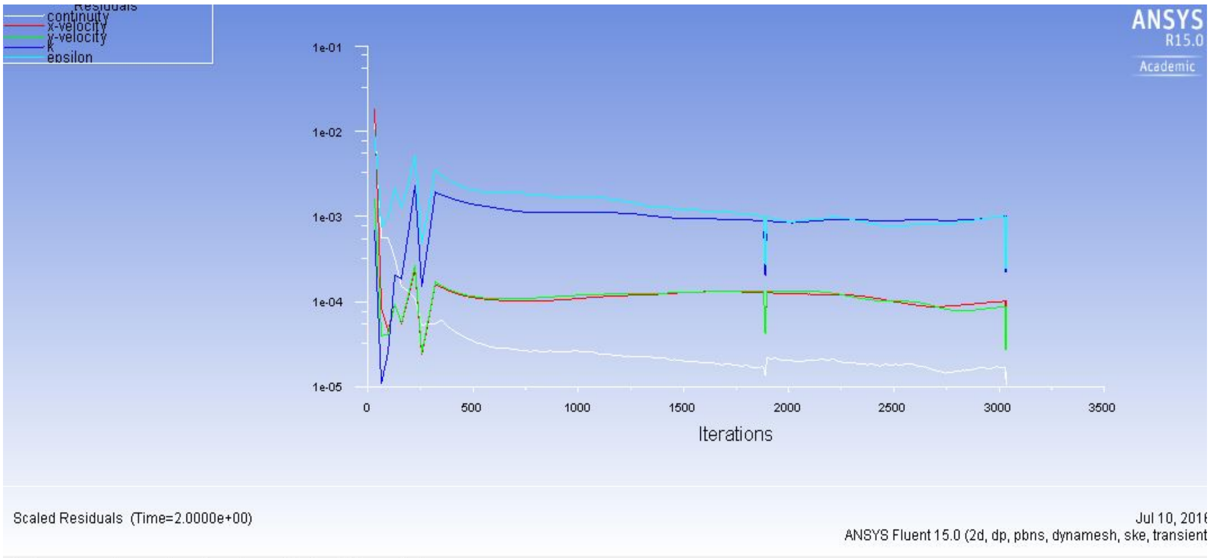

```
turbine                 (0 0 4.488631)                           (0 0 -0.24210363)                        (0 0 4.2465273)                          (0 0 0.36642611)
wall                    (0 0 -54.015794)                         (0 0 -0.31339761)                        (0 0 -54.329191)                         (0 0 -4.4095398)
-------------------------  -----------------------------------------  -----------------------------------------  -----------------------------------------  ----------------
Net                     (0 0 -49.527163)                         (0 0 -0.55550124)                        (0 0 -50.082664)                         (0 0 -4.0431137)

Moments - Moment Center (0 0 0) Moment Axis (0 0 1)
                          Moments (n-m)                                          Coefficients
Zone                      Pressure        Viscous         Total            Pressure        Viscous         Total
turbine                   4.488631        -0.24210363     4.2465273        0.36642611      -0.019763953    0.34666216
wall                      -54.015794      -0.31339761     -54.329191       -4.4095398      -0.025583985    -4.4351238
------------------------- --------------- --------------- ---------------  --------------- --------------- ---------------
Net                       -49.527163      -0.55550124     -50.082664       -4.0431137      -0.045347937    -4.0884617

Moments - Moment Center (0 0 0)
                          Moments (n-m)                                                                                                       Coefficients
Zone                      Pressure                                 Viscous                                  Total                                    Pressure
turbine                   (0 0 4.488631)                           (0 0 -0.24210363)                        (0 0 4.2465273)                          (0 0 0.36642611)
-------------------------  -----------------------------------------  -----------------------------------------  -----------------------------------------  ----------------
Net                       (0 0 4.488631)                           (0 0 -0.24210363)                        (0 0 4.2465273)                          (0 0 0.36642611)

Moments - Moment Center (0 0 0) Moment Axis (0 0 1)
                          Moments (n-m)                                          Coefficients
Zone                      Pressure        Viscous         Total            Pressure        Viscous         Total
turbine                   4.488631        -0.24210363     4.2465273        0.36642611      -0.019763953    0.34666216
------------------------- --------------- --------------- ---------------  --------------- --------------- ---------------
Net                       4.488631        -0.24210363     4.2465273        0.36642611      -0.019763953    0.34666216
```

Fig.32 Net Torque1




```
mkdir: cannot create directory 'D:': File exists
Writing "D:\Six DOF-1-2.0000.dat"...
Done.

Flow time = 1.999999999999891s, time step = 2000

Moments - Moment Center (0 0 0)
                          Moments (n-m)
Zone                      Pressure                                     Viscous                                      Total
turbine                   (0 0 3.6934338)                              (0 0 0.055610489)                            (0 0 3.7490443)
------------------------- -------------------------------------------- -------------------------------------------- ---------------
Net                       (0 0 3.6934338)                              (0 0 0.055610489)                            (0 0 3.7490443)

Moments - Moment Center (0 0 0) Moment Axis (0 0 1)
                          Moments (n-m)                                 Coefficients
Zone                      Pressure       Viscous        Total           Pressure       Viscous        Total
turbine                   3.6934338      0.055610489    3.7490443       0.30151077     0.0045397216   0.30605049
------------------------- -------------- -------------- --------------- -------------- -------------- ---------------
Net                       3.6934338      0.055610489    3.7490443       0.30151077     0.0045397216   0.30605049
```

Fig. 33 Net Torque2

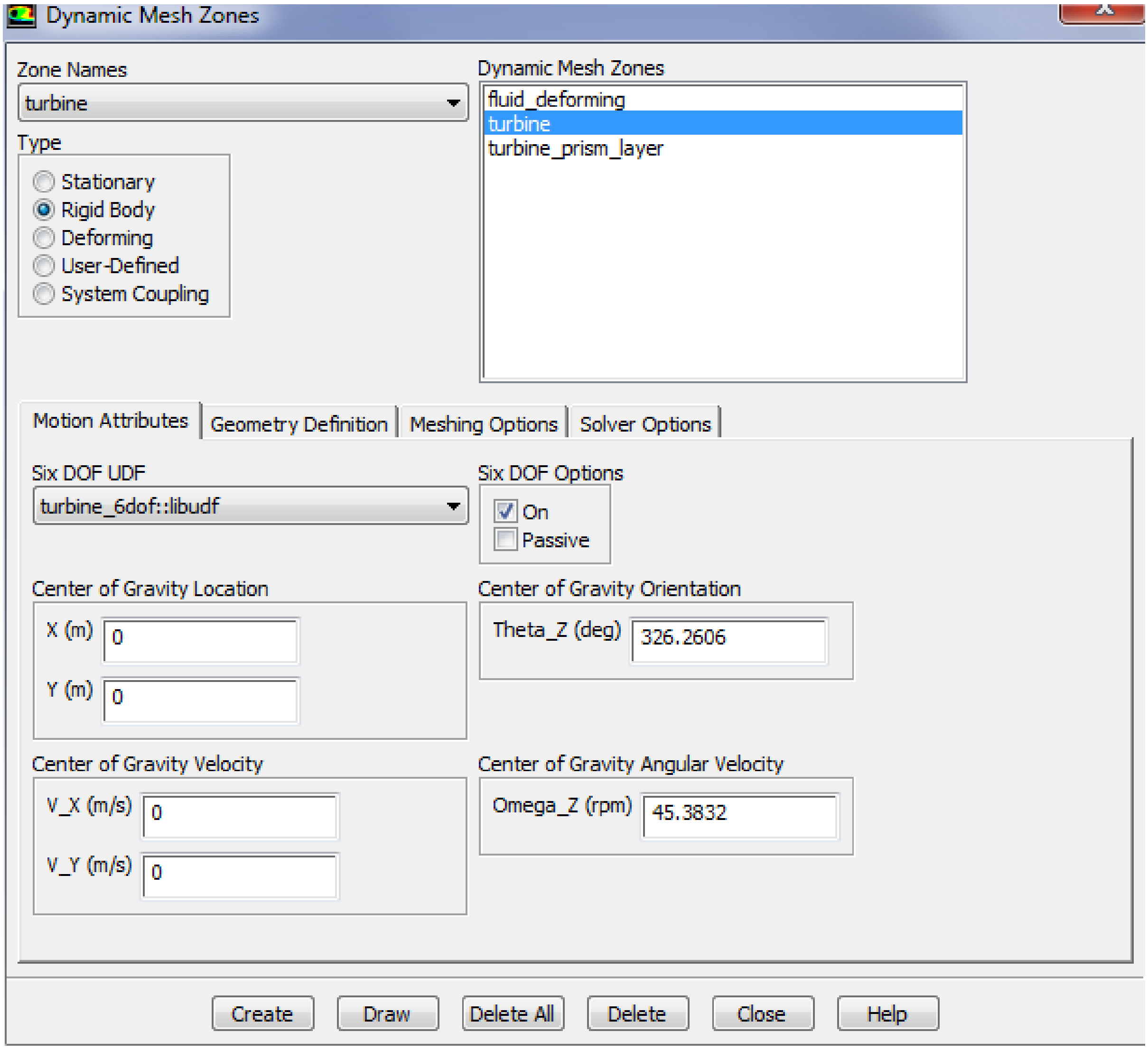


Fig.34 Six DOF

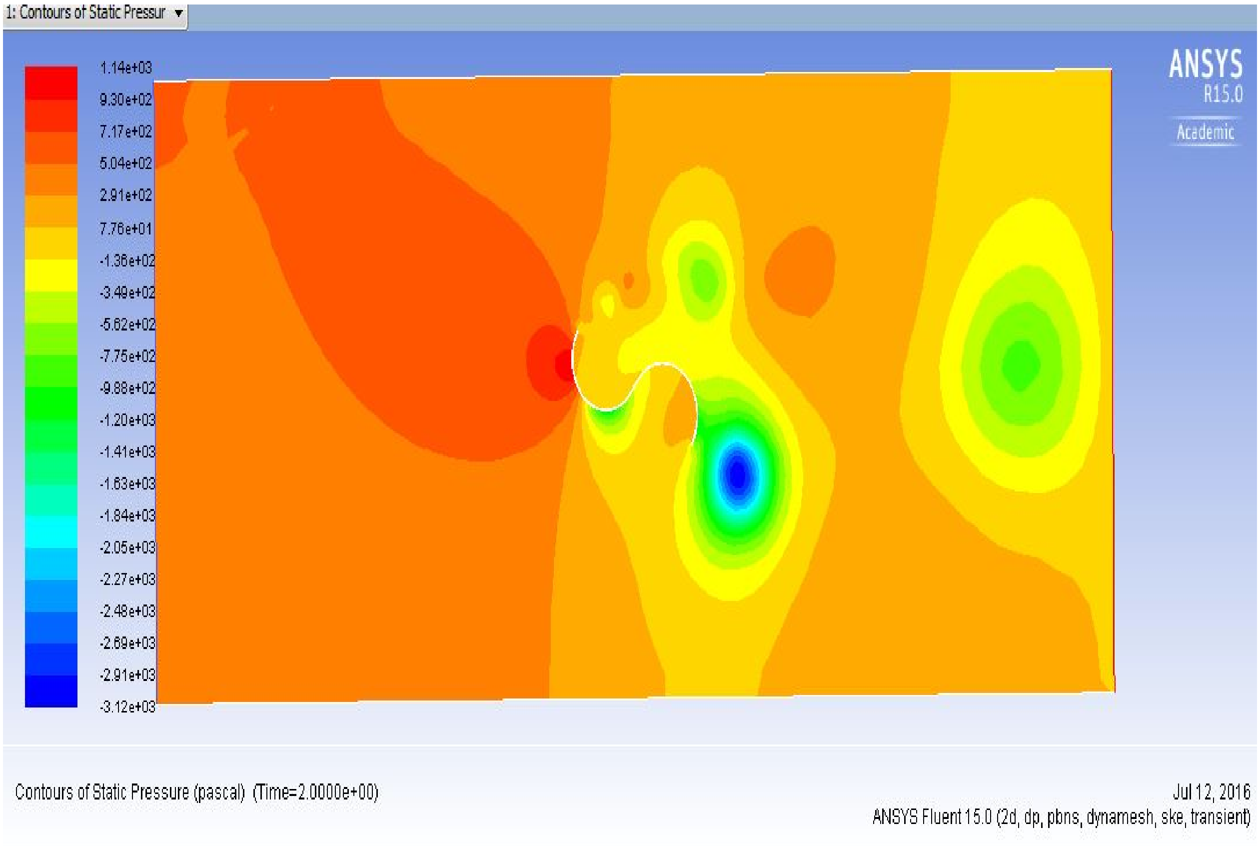


Fig.35 Pressure contour

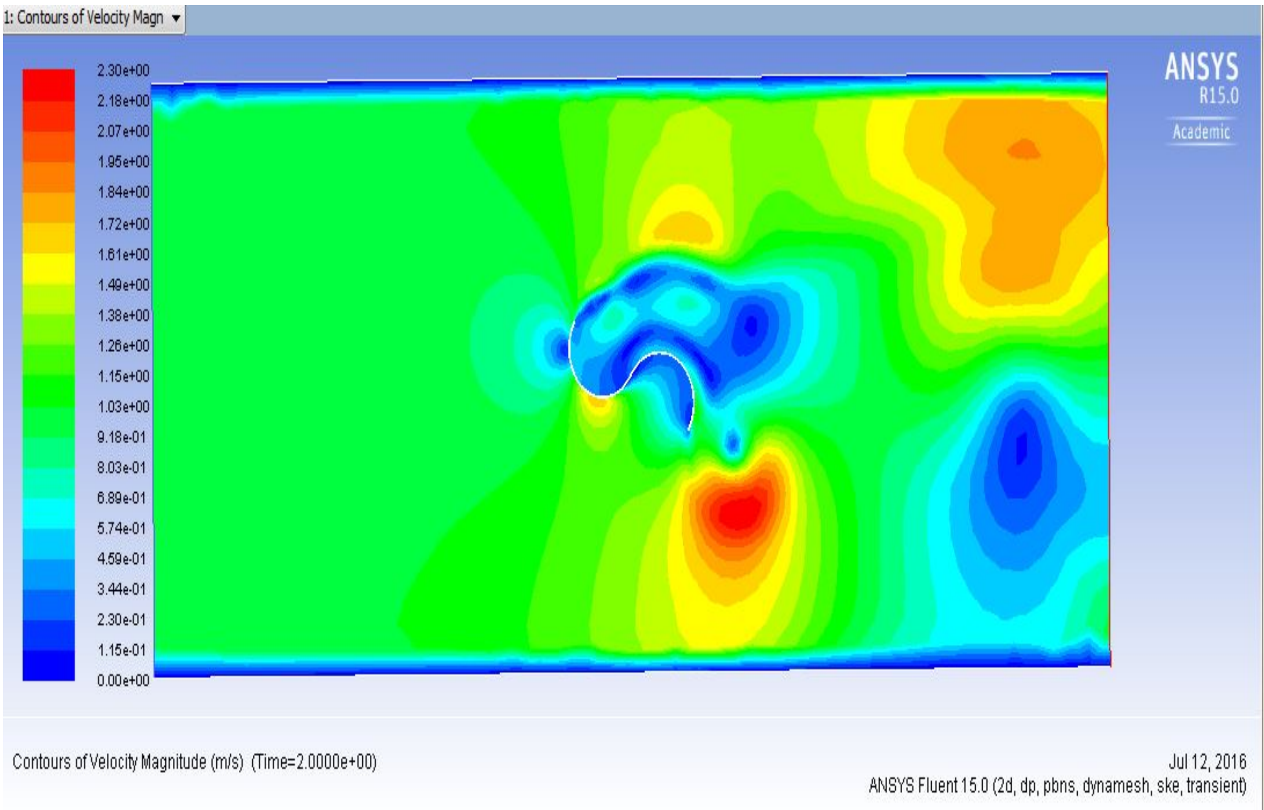


Fig.36 Velocity Contour

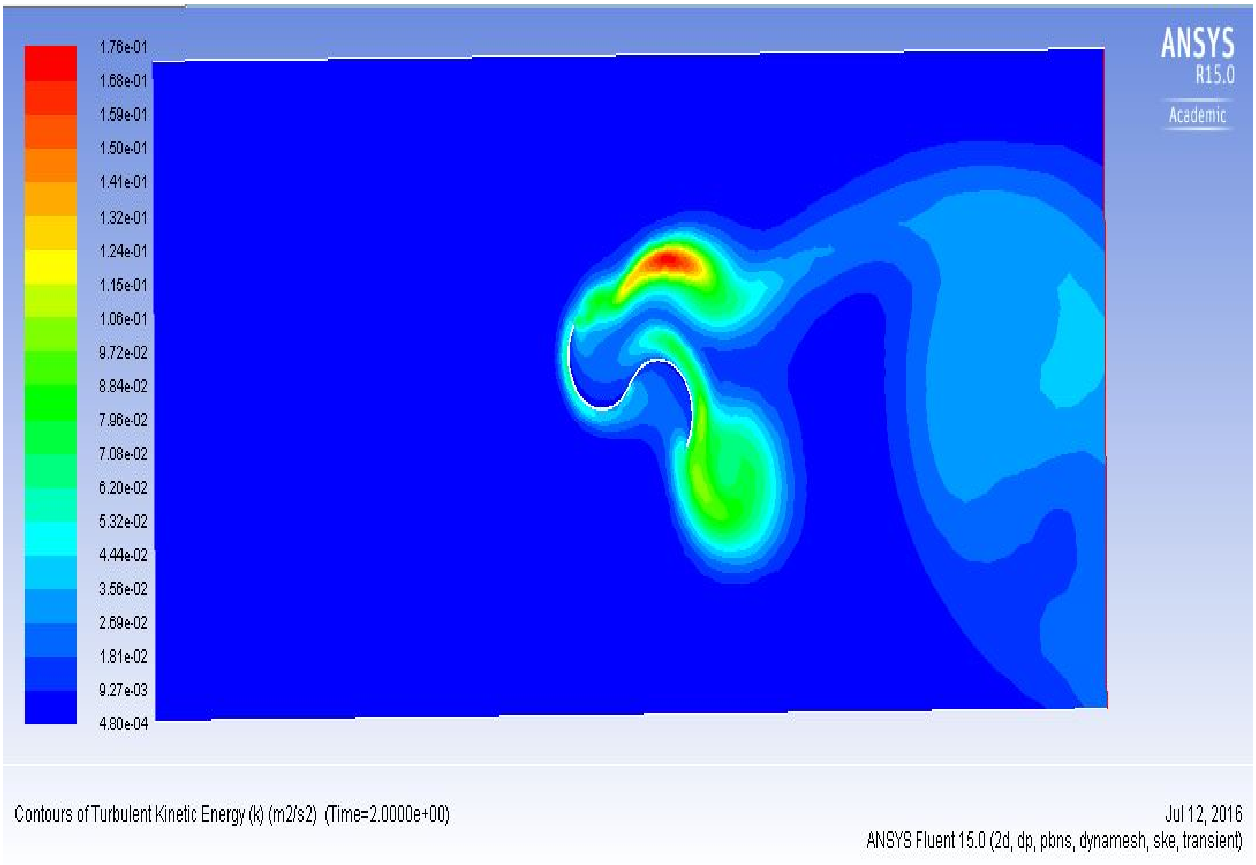


Fig.37 Turbulence Contour

The time versus rpm graph has been plotted at 1m/s for the turbine :

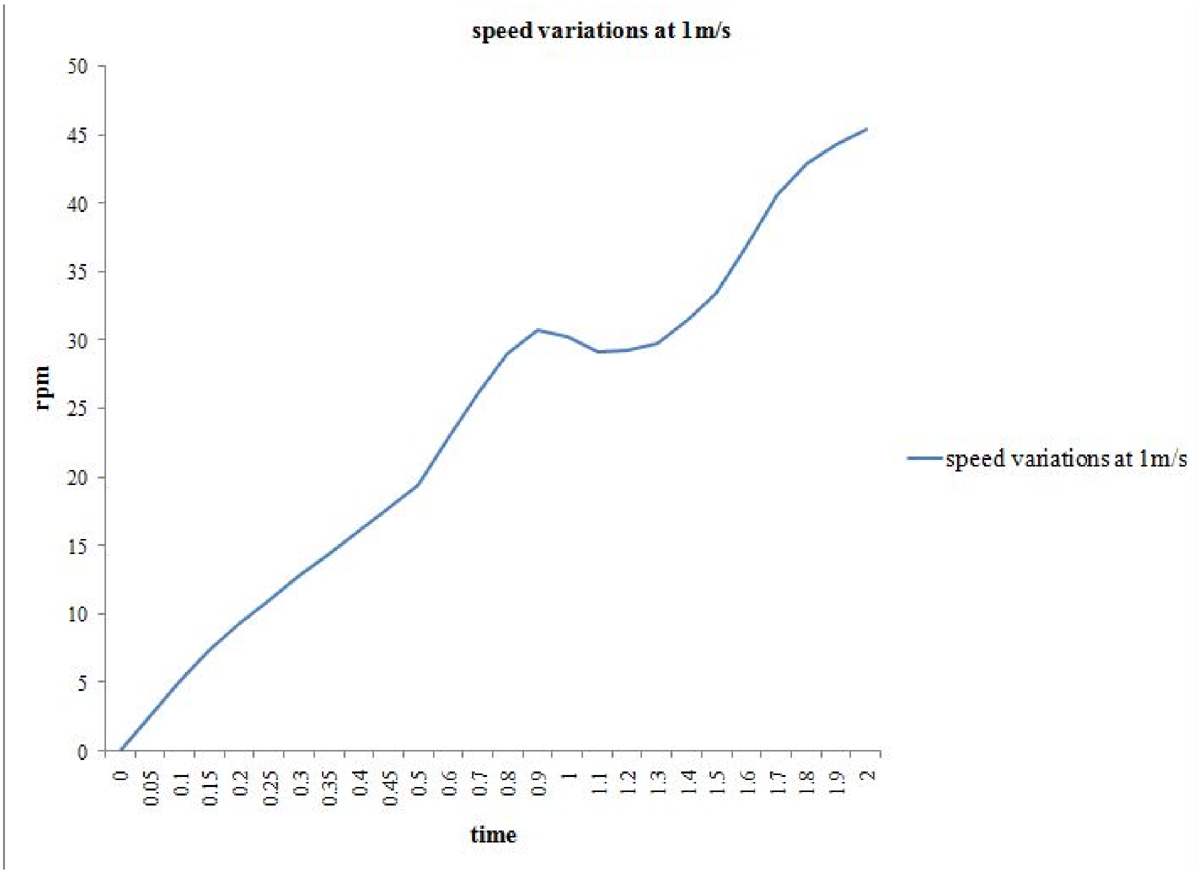
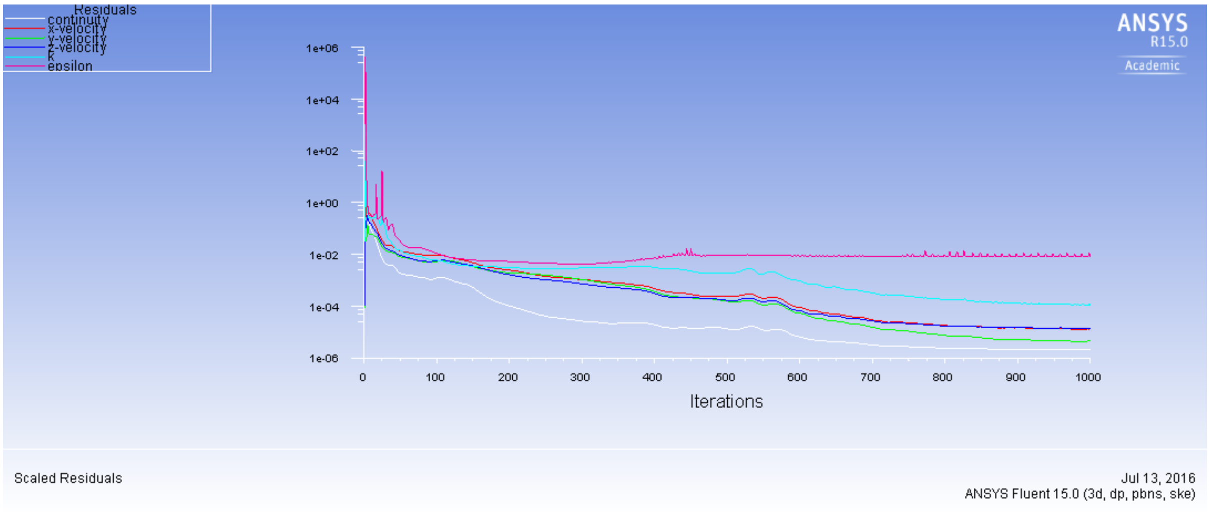


Fig.38 speed (rpm) versus time (second)

For determining the change in rotational speed, dynamic mesh is used under which a user defined function is made to be read by the solver. At every time interval, rpm is read and finally a time versus rpm graph is plotted along x-y axes respectively.

The torque result for 0.3 m/s is:




```
  996  2.1720e-06  1.3711e-05  4.6175e-06  1.4520e-05  1.1445e-04  8.2677e-03  0:00:08    4
  997  2.1741e-06  1.3585e-05  4.6473e-06  1.4487e-05  1.1423e-04  8.3988e-03  0:00:06    3
  998  2.1759e-06  1.3565e-05  4.6832e-06  1.4599e-05  1.1767e-04  1.0851e-02  0:00:04    2
  999  2.1785e-06  1.3835e-05  4.6800e-06  1.4669e-05  1.2667e-04  1.0326e-02  0:00:02    1
 1000  2.1756e-06  1.3638e-05  4.6694e-06  1.4674e-05  1.1794e-04  8.5340e-03  0:00:00    0

Moments - Moment Center (0 0 0)
                          Moments (n-m)
Zone                      Pressure                                       Viscous                                         Total
turbine                   (0.0024991262 0.27138317 0.013872742)          (0.0037280464 -0.022769872 -0.0051056917)       (0.0062271727 0
------------------------- ---------------------------------------------- ----------------------------------------------- ---------------
Net                       (0.0024991262 0.27138317 0.013872742)          (0.0037280464 -0.022769872 -0.0051056917)       (0.0062271727 0

Moments - Moment Center (0 0 0) Moment Axis (0 1 0)
                          Moments (n-m)                                  Coefficients
Zone                      Pressure        Viscous         Total          Pressure        Viscous         Total
turbine                   0.27138317      -0.022769872    0.24861329     0.00067129024   -5.6323291e-05  0.00061496695
------------------------- --------------- --------------- -------------- --------------- --------------- ---------------
Net                       0.27138317      -0.022769872    0.24861329     0.00067129024   -5.6323291e-05  0.00061496695
```

Fig.39 Torque at 0.3m/s

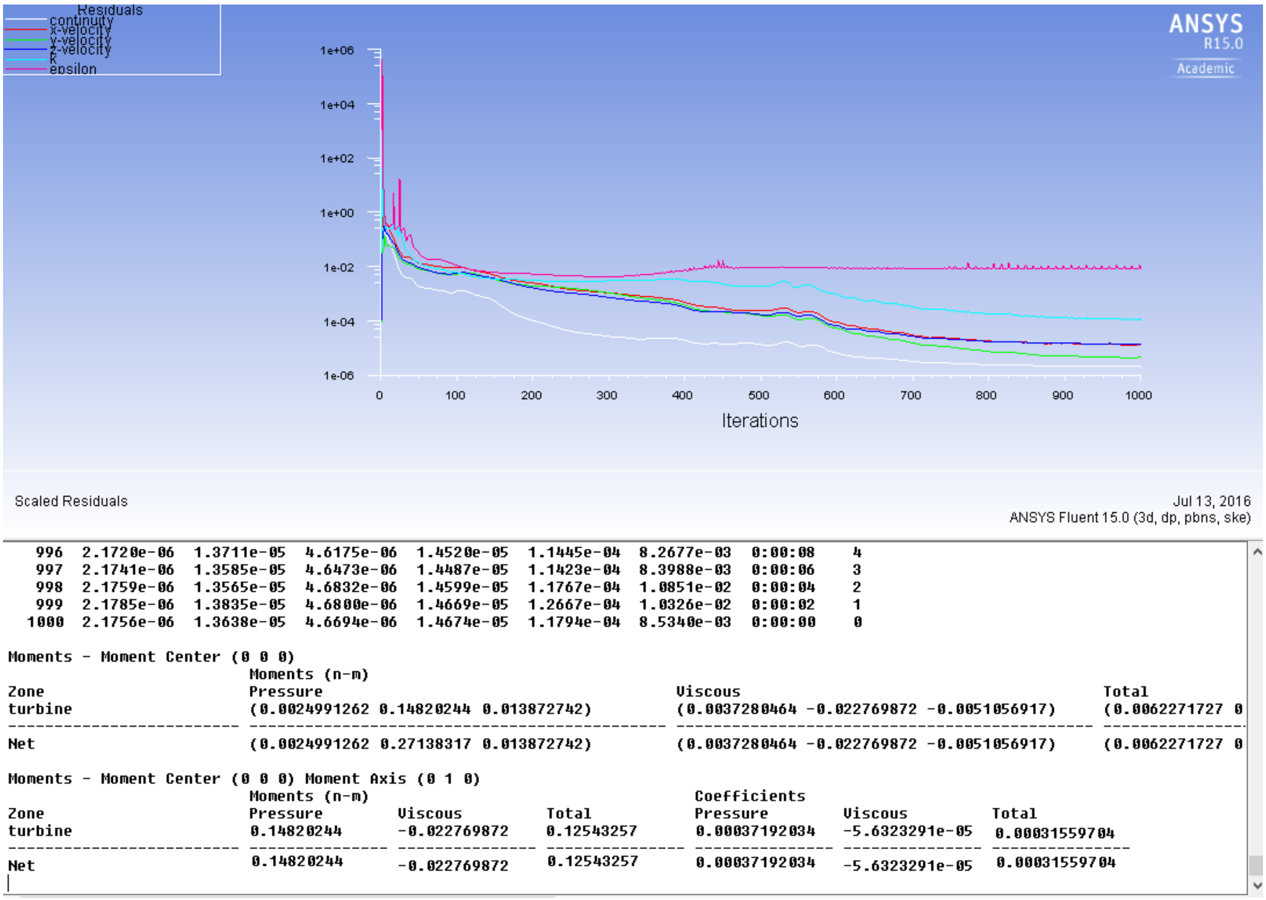
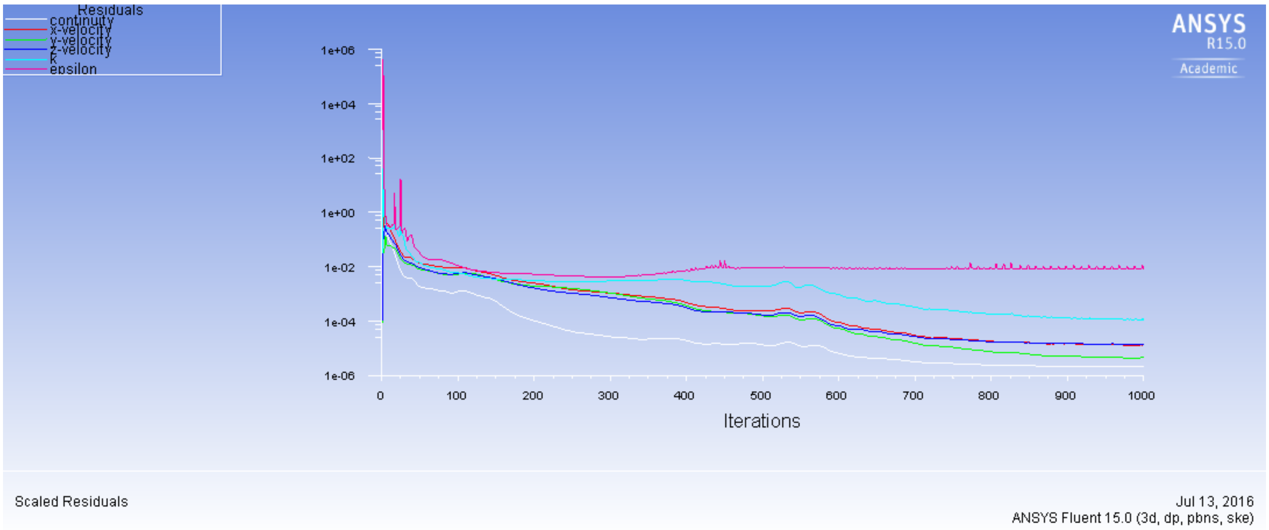


Fig.40 Torque at 0 velocity corresponding to 18.907 rpm

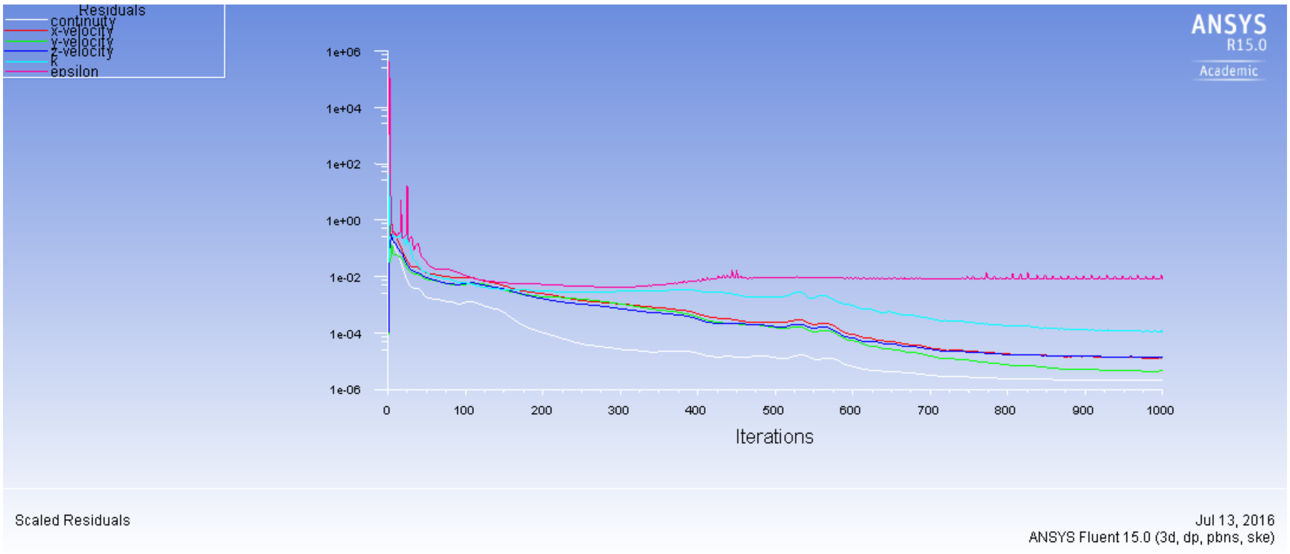



```
   996  2.1720e-06  1.3711e-05  4.6175e-06  1.4520e-05  1.1445e-04  8.2677e-03  0:00:08    4
   997  2.1741e-06  1.3585e-05  4.6473e-06  1.4487e-05  1.1423e-04  8.3988e-03  0:00:06    3
   998  2.1759e-06  1.3565e-05  4.6832e-06  1.4599e-05  1.1767e-04  1.0851e-02  0:00:04    2
   999  2.1785e-06  1.3835e-05  4.6800e-06  1.4669e-05  1.2667e-04  1.0326e-02  0:00:02    1
  1000  2.1756e-06  1.3638e-05  4.6694e-06  1.4674e-05  1.1794e-04  8.5340e-03  0:00:00    0

Moments - Moment Center (0 0 0)
                                Moments (n-m)
Zone                            Pressure                                         Viscous                                           Total
turbine                         (0.0024991262 1.08553268 0.013872742)            (0.0037280464 -0.022769872 -0.0051056917)         (0.0062271727 0
-------------------------       -----------------------------------------------  -----------------------------------------------  ----------------
Net                             (0.0024991262 0.27138317 0.013872742)            (0.0037280464 -0.022769872 -0.0051056917)         (0.0062271727 0

Moments - Moment Center (0 0 0) Moment Axis (0 1 0)
                                Moments (n-m)                                      Coefficients
Zone                            Pressure        Viscous         Total              Pressure          Viscous           Total
turbine                          1.08553268     -0.022769872    0.87823215         0.00023543264     -4.4538626e-05    0.00019089401
-------------------------       --------------- --------------- ---------------  ---------------   ---------------   ---------------
Net                              1.08553268     -0.022769872    0.87823215         0.00023543264     -4.4538626e-05    0.00019089401
```

Fig.40 Torque at 0.65m/s

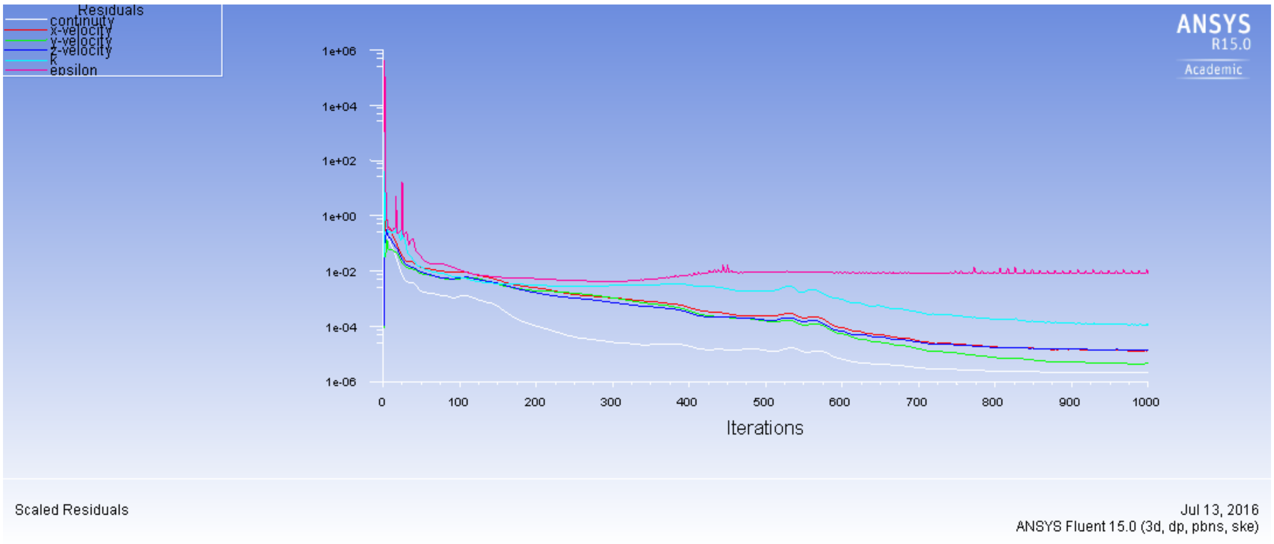



```
   996  2.1720e-06  1.3711e-05  4.6175e-06  1.4520e-05  1.1445e-04  8.2677e-03  0:00:08     4
   997  2.1741e-06  1.3585e-05  4.6473e-06  1.4487e-05  1.1423e-04  8.3988e-03  0:00:06     3
   998  2.1759e-06  1.3565e-05  4.6832e-06  1.4599e-05  1.1767e-04  1.0851e-02  0:00:04     2
   999  2.1785e-06  1.3835e-05  4.6800e-06  1.4669e-05  1.2667e-04  1.0326e-02  0:00:02     1
  1000  2.1756e-06  1.3638e-05  4.6694e-06  1.4674e-05  1.1794e-04  8.5340e-03  0:00:00     0

Moments - Moment Center (0 0 0)
                           Moments (n-m)
Zone                       Pressure                                       Viscous                                        Total
turbine                    (0.0024991262 0.39893421 0.013872742)          (0.0037280464 -0.022769872 -0.0051056917)      (0.0062271727 0
-------------------------  ---------------------------------------------  ---------------------------------------------  ----------------
Net                        (0.0024991262 0.27138317 0.013872742)          (0.0037280464 -0.022769872 -0.0051056917)      (0.0062271727 0

Moments - Moment Center (0 0 0) Moment Axis (0 1 0)
                           Moments (n-m)                                      Coefficients
Zone                       Pressure        Viscous         Total              Pressure        Viscous         Total
turbine                    0.39893421      -0.013080195    0.38585402         0.00032435179   -4.7964538e-05  0.00027638725
-------------------------  --------------- --------------- ---------------    --------------- --------------- ---------------
Net                        0.39893421      -0.013080195    0.38585402         0.00067129024   -4.7964538e-05  0.00027638725
```

Fig.41 Torque at 0 velocity corresponding to 29.98 rpm

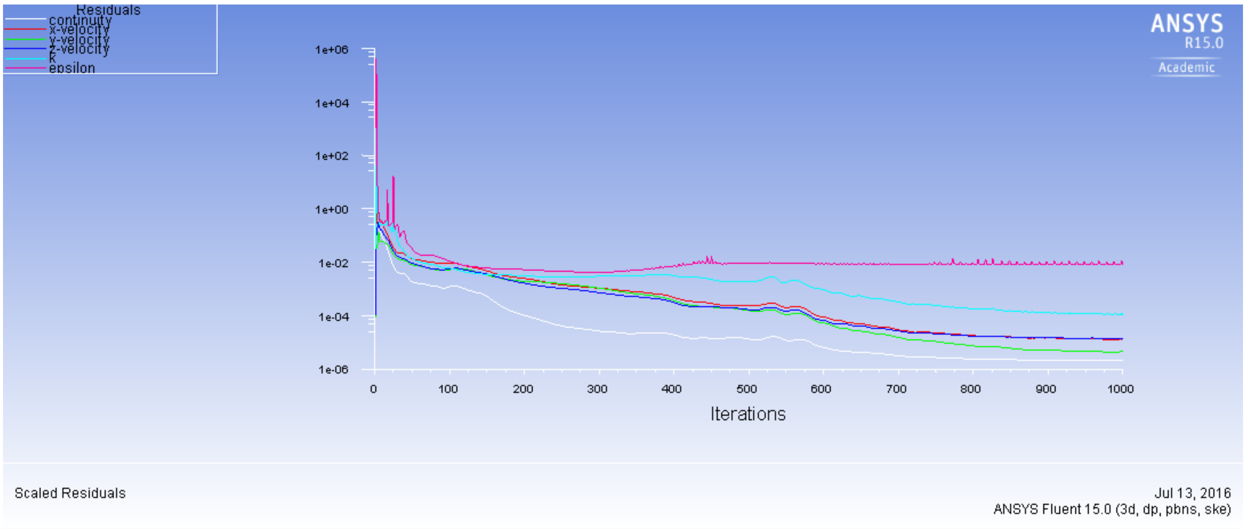



```
  996  2.1720e-06  1.3711e-05  4.6175e-06  1.4520e-05  1.1445e-04  8.2677e-03  0:00:08    4
  997  2.1741e-06  1.3585e-05  4.6473e-06  1.4487e-05  1.1423e-04  8.3988e-03  0:00:06    3
  998  2.1759e-06  1.3565e-05  4.6832e-06  1.4599e-05  1.1767e-04  1.0851e-02  0:00:04    2
  999  2.1785e-06  1.3835e-05  4.6800e-06  1.4669e-05  1.2667e-04  1.0326e-02  0:00:02    1
 1000  2.1756e-06  1.3638e-05  4.6694e-06  1.4674e-05  1.1794e-04  8.5340e-03  0:00:00    0

Moments - Moment Center (0 0 0)
                             Moments (n-m)
Zone                         Pressure                                        Viscous                                          Total
turbine                      (0.0024991262 2.01342861 0.013872742)           (0.0037280464 -0.022769872 -0.0051056917)        (0.0062271727 0
---------------------------- ----------------------------------------------- ------------------------------------------------ ---------------
Net                          (0.0024991262 0.27138317 0.013872742)           (0.0037280464 -0.022769872 -0.0051056917)        (0.0062271727 0

Moments - Moment Center (0 0 0) Moment Axis (0 1 0)
                             Moments (n-m)                                     Coefficients
Zone                         Pressure        Viscous         Total             Pressure        Viscous         Total
turbine                      2.01342861      -0.037905212    1.97552340        0.00096454377   -4.9746432e-05  0.00091479733
---------------------------- --------------- --------------- ---------------   --------------- --------------- ---------------
Net                          2.01342861      -0.037905212    1.97552340        0.00096454377   -4.9746432e-05  0.00091479733
```

Fig.42 Torque at 0.9 m/s




```
  996  2.1720e-06  1.3711e-05  4.6175e-06  1.4520e-05  1.1445e-04  8.2677e-03  0:00:08    4
  997  2.1741e-06  1.3585e-05  4.6473e-06  1.4487e-05  1.1423e-04  8.3988e-03  0:00:06    3
  998  2.1759e-06  1.3565e-05  4.6832e-06  1.4599e-05  1.1767e-04  1.0851e-02  0:00:04    2
  999  2.1785e-06  1.3835e-05  4.6800e-06  1.4669e-05  1.2667e-04  1.0326e-02  0:00:02    1
 1000  2.1756e-06  1.3638e-05  4.6694e-06  1.4674e-05  1.1794e-04  8.5340e-03  0:00:00    0

Moments - Moment Center (0 0 0)
                          Moments (n-m)
Zone                      Pressure                                    Viscous                                     Total
turbine                   (0.0024991262 0.88575307 0.013872742)       (0.0037280464 -0.022769872 -0.0051056917)   (0.0062271727 0
------------------------- ------------------------------------------- ------------------------------------------- ----------------
Net                       (0.0024991262 0.27138317 0.013872742)       (0.0037280464 -0.022769872 -0.0051056917)   (0.0062271727 0

Moments - Moment Center (0 0 0) Moment Axis (0 1 0)
                          Moments (n-m)                                  Coefficients
Zone                      Pressure        Viscous         Total           Pressure        Viscous         Total
turbine                   0.88575307      -0.017656547    0.86809653      0.00086753210   -4.2370741e-05  0.00082516135
------------------------- --------------- --------------- --------------- --------------- --------------- ---------------
Net                       0.88575307      -0.017656547    0.86809653      0.00086753210   -4.2370741e-05  0.00082516135
```

Fig.43 Torque at 0 velocity corresponding to 36.955 rpm

4.2 CALCULATIONS

The available power at the inlet to the turbine is given as:

$P_{in} = 0.5\rho \text{ A } v^3$

Case 1: if v = 0.8m/s,

P_{in} =36.288 W

Torque obtained =3.749 Nm, ω= 17.524 rpm

Hence power obtained = 6.88 W

Hence, $C_p = \frac{P_{obtained}}{P_{in}} = \frac{6.88}{36.288}$ =0.189

Case 2: if v = 1m/s

$P_{in} = 0.5\rho \text{ A } v^3$

P_{in} = 70.875

Torque obtained =4.24 Nm, ω = 46.654 rpm

Hence, power obtained = 20.7149 W

Hence, $C_p = \frac{P_{obtained}}{P_{in}} = \frac{20.7149}{70.875}$ =0.2922

CALCULATION FOR 3D- STEADY SETUP

(1)
At v = 0.3 m/s, ω= 1.98 rad/s
Torque due to rotation and flow velocity = 0.2486 Nm
Torque at 0 velocity =0.12 Nm

Hence, net toque developed by the blade = 0.2486 – 0.12
= 0.1286 Nm
Hence, power developed = $T_{net} \times \omega$ =0.2286 watt.

Hence, $C_p = \frac{P_{developed}}{0.5 \times \rho \times A \times V^3}$ = 0.397

Similarly,
(2)
At v = 0.65 m/s, ω= 3.14 rad/s
Torque due to rotation and flow velocity = 0.878 Nm
Torque at 0 velocity =0.3858 Nm

Hence, net toque developed by the blade = 0.878 – 0.3858
= 0.4922 Nm
Hence, power developed = $T_{net} \times \omega$ =1.5455 watt.

Hence, $C_p = \frac{P_{developed}}{0.5 \times \rho \times A \times V^3}$ = 0.2489

(3)
At v = 0.9 m/s, ω= 3.87 rad/s
Torque due to rotation and flow velocity = 1.9755 Nm
Torque at 0 velocity = 0.8680 Nm

Hence, net toque developed by the blade = 1.9755 – 0.8680
= 1.1075 Nm
Hence, power developed = $T_{net} \times \omega$ =4.286 watt.

Hence, $C_p = \frac{P_{developed}}{0.5 \times \rho \times A \times V^3}$ = 0.2600 [35]

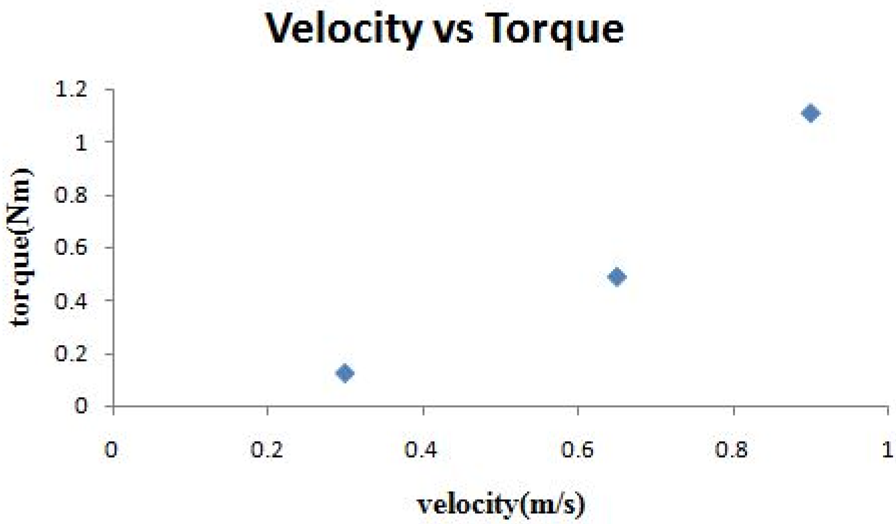


Fig.44 Velocity vs Torque

According to earlier research work:

Velocity(m/s)	rpm	C_p
0.3	18.907	0.30
0.65	29.98	0.31
0.9	36.955	0.33

4.3. DISCUSSION

In the contour plots, the blade on the left hand side is the advancing blade and that on the right hand side is the returning blade. The contour plots predict the variations in velocity and pressure in various regions near the blades within the flow domain. It can be observed from the pressure contour plots that pressure drop occur across the rotor from upstream to downstream side. This pressure drop indicates power that can be extracted by the rotor causing its rotation. CFD analysis on the Savonius turbine facilitates a more thorough study on the pressure, velocity and torque distribution across the hydrokinetic turbine and therefore to cultivate more insight of design information about its performance under low velocity states. Taken as a whole, study of the two bladed Savonius water turbine hints that the turbine has large potential to be used for significant power generation. It is clear that the water turbine is one of the most emerging technologies in the prospective area of renewable energy. The technology must evolve even more and rise in respect to the need of the whole mankind. The study carries immense scope to be researched in future.

5. **CONCLUSION**

The two bladed Savonius water turbine is computationally analyzed (CFD) under very low velocity conditions. The performance of the Savonius water turbine with an aspect ratio of 0.7 has been evaluated by finding out generation of torque and power with corresponding coefficient of power (*Cp*) values. The flow characteristics in the vicinity of the turbine have been studied as it is very significant on the performance. The following conclusions have been inferred from the work:

As the velocity of flowing stream increases, toque increases but from the results it is also observed that if the velocity of stream increases further, turbulency increases and the turbulent water strikes the blade, which may lead to mechanical damage. Speed of turbine changes with time for some time after which it attains almost a constant range value. This can be observed from the graph plotted between time and speed on x and y axes respectively.
The power extracted by the turbine also depends on angular velocity of the turbine which increases as the energy content in the free stream increases.

6. **FUTURE RESEARCH**

Since hydrokinetic technology is very recent, and almost every company is still in the research and development phase, there will be considerable research opportunities for some time, particularly on long-term performance. Environmental research still needs to be done on hydrokinetic turbines. Despite initial tests already explained in this thesis, more extensive research is needed on environmental effects using such turbines'. The selection of turbines, rotor blades and installation sites create many environmental questions. Environmental research is in particular important to relieve not only the permitting process, but also the shift into commercialization. Unidentified environmental factors could set hurdles or stop potential permits or sales and hurt the prospect of a promising technology. The permitting process needs more research to help put right what is usually considered a broken system. Research to demonstrate easier permitting methods that remain effective could lay down a major step for hydrokinetic energy to develop and reach its potential. Momentous reforms may be on the way for hydrokinetic permits, but regardless research on permitting hindrances and the impacts it has on future companies needs to be methodically addressed so that permitting doesn't continue to be a roadblock for technological and commercial progress in the hydrokinetic market. A vital but complicated research topic is the long-term performance of hydrokinetic turbines in various situations. This research cannot easily be carried out because hydrokinetics is such a recent technology. When turbines are purchased and installed for long periods of time, however, it will be important to track the consistency and longevity of the turbine, especially in harsher natural environments where damage, wear and tear are more likely. Data on this subject could help greatly with future turbine designs and projected power outputs, leading to more accurate turbine pricing, which would help create a more reliable and sturdy market. Turbine spacing represents an area of study that could drastically change the hydrokinetic market. More research on how turbines impact water velocity and how far they need to be apart will allow companies to sell and install the maximum number per site, increasing power output and profit.

Lastly, much research needs to be done on the impact different rotor blades have on hydrokinetic turbines. These blades will likely have a direct effect on many issues, making it arguably the most important research topic for hydrokinetic energy. Choosing a rotor blade will greatly affect a turbine's power output, while also possibly affecting the turbine's longevity and any environmental impacts.

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