MODELING AND ANALYSIS OF RADIAL FLOW TURBINE

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ABSTRACT

The radial-inflow turbine has been in use for many years. It first appeared as a practical power-producing unit in the hydraulic turbine field. Basically, a centrifugal compressor with reversed flow and opposite rotation, the radial-inflow turbine was the first used in jet engine flight in the late 1930s. It was considered the natural combination for centrifugal compressor used in the same engine. Designers thought it easier to match the thrust from the two rotors and that the turbine would have a higher efficiency than the compressor for the same rotor because of the accelerating nature of the flow.The performance of the radial -inflow turbine is now being investigated with more interest by the transportation and chemical industries: in transportation, this turbine is used in turbochargers for both spark ignition and diesel engines; in aviation, the radial -inflow turbine is used as n expander in environmental control systems, and in the petrochemical industry, it is used in expander designs, gas liquefaction expanders, and other cryogenic systems. The radial -inflow turbine are also used in various small gas turbines to power helicopters and as standby generating units.The radial -inflow turbine's greatest advantage is that the work produced by a single stage is equivalent to that of two or more states in an axial turbine. This phenomenon occurs because a radial-inflow turbine usually has a higher tip speed than an axial turbine. Since the power output is a function of the square of the tip speed (p α u^2) for a given flow rate, the work is greater than in a single-stage axial-flow turbine.In this paper the modeling of radial flow turbine is done with the SOLID WORKS and the CFD analysis is done with ANSYS software for calculating the efficiency of radial flow turbine by keeping different flow entry angle of liquid i.e. 20,25 and 30 degrees at a pressure of 10 m/s, 7 m/s and 4 m/s. and compared with theoretical values.

Introduction

The radial flow turbine has had a long history of development being first conceived for the purpose of producing hydraulic power over 170 years ago. A French engineer, Fourneyron, developed the first commercially successful hydraulic turbine (c. 1830) and this was of theradial-outflow type. A radial-inflow type of hydraulic turbine was built by Francis and Boyden in the U.S.A. (c.1847) which gave excellent results and was highly regarded. This type of machine is now known as the Francis turbine, a simplified arrangement of it being shown in Figure 1.1. It will be observed that the flow path followed is from the radial direction to what is substantially an axial direction. A flow path in the reverse direction (radial-outflow), for a single stage turbine anyway, creates several problems one of which is low specific work. However, as pointed out by Shepherd (1956) radial-outflow steam turbines comprising many stages have received considerable acceptance in Europe. Figurefrom Kearton (1951), shows diagrammatically the Ljungstro¨m steam turbine which, because of the tremendous increase in specific volume of steam, makes the radial-outflow flow path virtually imperative. directions so that they can both be regarded as rotors.

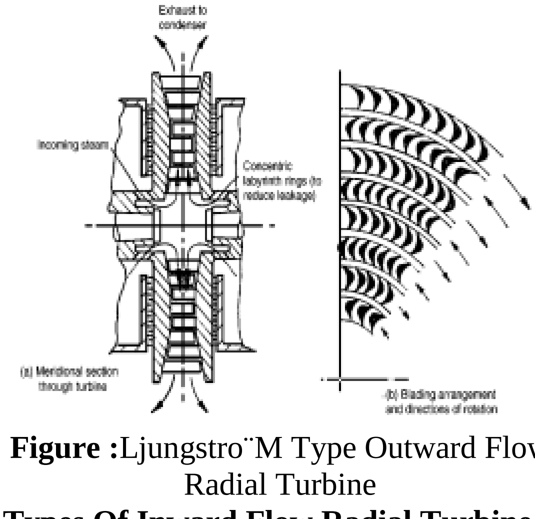


Figure : Ljungstro¨M Type Outward Flow Radial Turbine

Types Of Inward Flow Radial Turbine

In the centripetal turbine energy is transferred from the fluid to the rotor in passing from a large radius to a small radius. For the production of positive work the product of U_c at entry to the rotor must be greater than U_e at rotor exit (eqn.). This is usually arranged by imparting a large component of tangential velocity at rotor entry, using single or multiple nozzles, and allowing little or no swirl in the exit absolute flow.

Cantilever Turbine

Figure1.2 shows a cantil ever IFR turbine where the blades are limited to the region of the rotortip, extending from the rotor in the axial direction. In practice the cantil ever blades are usually of the impulse type (i.e.Lowreaction), by which it is implied that the reislittle change in relative evelocity at in let and outlet of the rotor. There is no fundamental reason why the blading should not be of the reaction type. However, the resulting expansion through the rotor would require an increase inflow area. This extra flow area is extremely difficult to accommodate in a small radial distance, especially as the radius decreases through the rotorrow.

Aerodynamically, the cantilever turbine is similar to an axial-impulse turbine and can even be designed by similar methods. Figure 1.3 shows the velocity triangles at rotor inlet and outlet. The fact that the flow is radial inwards hardly alters the design procedure because the blade radius ratio r2/r3 is close to unity anyway.

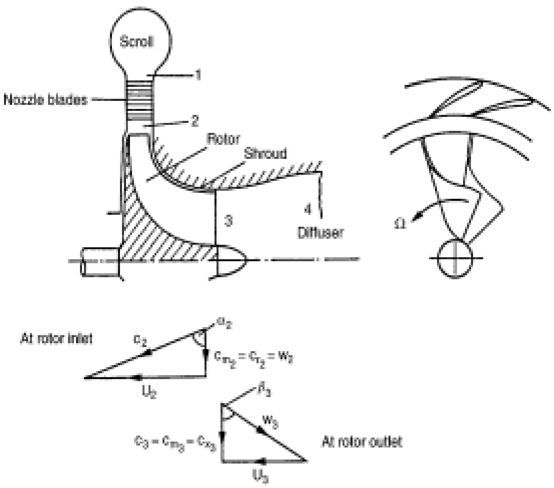


Figure: Layout And Velocity Diagrams For A 90 Deg Inward Flow Radial Turbine At The Nominal Design Point.

Radial Turbine Applications

The performance of the radial -inflow turbine is now being investigated with more interest by the transportation and chemical industries: in transportation, this turbine is used in turbochargers for both spark ignition and diesel engines; in aviation, the radial -inflow turbine is used as n expander in environmental control systems and in the petrochemical industry, it is used in expander designs, gas liquefaction expanders, and other cryogenic systems. The radial -inflow turbine are also used in various small gas turbines to power helicopters and as standby generating units. Radial turbines are also used in the aerospace area, where they are used for driving fuel pumps. The area where radial turbines are used in largest numbers is probably in the turbocharger application for Internal Combustion (IC) engines. In a turbocharger, the energy of the engine exhaust gas is extracted by expanding it through the turbine which drives the compressor by a shaft.

The Radial-Inflow Turbine Theoretical Calculation

The radial-inflow turbine has been in use for many years. It first appeared as a practical power-producing unit in the hydraulic turbine field. Basically a centrifugal compressor with reversed flow and opposite rotation, the radial-inflow turbine was the first used in jet engine flight in the late 1930s.. The nozzle blades in a vane turbine design are usually fitted around the rotor to direct the flow inward with the desired whirl component in the inlet velocity. The flow is accelerated through these blades. In low-reaction turbines the entire acceleration occurs in the nozzle vanes.

The rotor or impeller of the radial-inflow turbine consists of a hub, blades and in some cases , a shroud .the hub is the solid ax-symmetrical portion of the rotor. It defines the inner boundary of the flow passage and is sometimes called the disc. The blades are integral to the hub and exert a normal force on the flow stream. Flow behavior is observed inside the turbine at different guide vane angles and got the efficiency for all cases between theoretical efficiency. Characteristic curve is verified for all the different guide vane angles. For every case inlet velocity is changed as 10m/s, 7m/s, 4 m/sThe outlet diffuser is used to convert the high absolute velocity leaving the exducer into static pressure. If this conversion is not done, the efficiency of the unit will be low. The conversion of the flow to a static head must be done carefully, since the low -energy boundary layers cannot tolerate great adverse pressure gradients.

Derivation Of The Efficiency Of A Radial Inflow Turbine

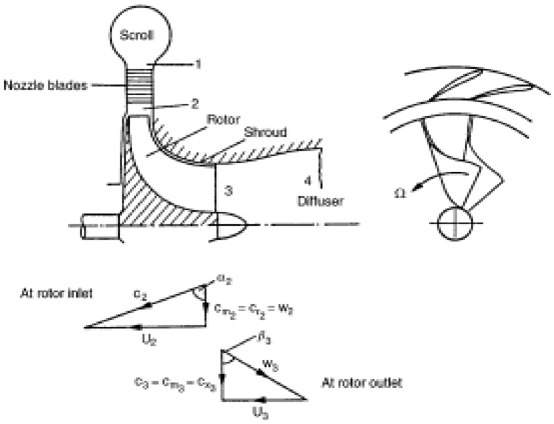


Figure: Layout And Velocity Diagrams For A 90 Deg Inward Flow Radial Turbine At Nominal Design Point.

The complete adiabatic expansion process for a turbine comprising a nozzle blade row, a radial rotor followed by a diffuser corresponding to the layout of Figure 2.1, is represented by the Mollier diagram shown in Figure 2.2. In the turbine, frictional processes cause the entropy to increase in all components and these irreversibility's are implied in Figure

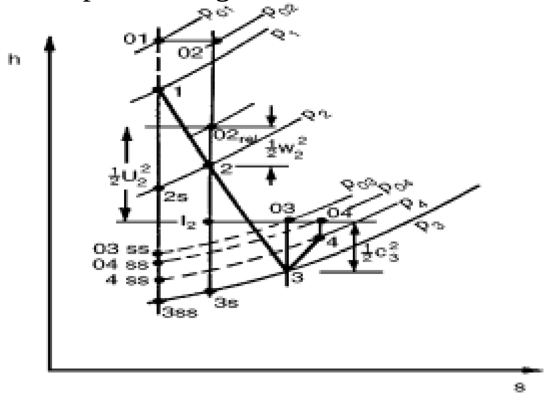


Figure: Mollier diagram

Across the nozzle blades the stagnation enthalpy is assumed constant, h01 = h0

And, therefore, the static enthalpy drop is,

h_1-h_2= 1/2 (c_2^2-c_1^2)

Corresponding to the static pressure change from P1 to the lower pressure P2. The ideal enthalpy change (h_1 -h_2 s) is between these same two pressures but at constant entropy.

I=h_{0rel}-1/2U^2 is constant for an adiabatic irreversible flow process, relative to a rotating component.

For the rotor of the 90 deg IFR turbine,

$h_{02rel}-1/2U^2_2 = h_{03rel}-1/2U^2_3$

Thus, as h_{0rel}= h+1/2 w^2,

$h_2-h_3= 1/2 \; (U^2_2-U^2_3)-(w^2_2-w^2_3)$

In this analysis the reference point 2 (Figure 2.3) is taken to be at the inlet radius r2 of the rotor (the blade tip speed U2 =Ωr_2

This implies that the nozzle irreversibility's are lumped together with any friction losses occurring in the annular space between nozzle exit and rotor entry (usually scroll losses are included as well). Across the diffuser the stagnation enthalpy does not change, h03 = h04, but the static enthalpy increases as a result of the velocity diffusion. Hence,

$h_4-h_3=1/2(c^2_3-c^2_4)$

The specific work done by the fluid on the rotor is

$\Delta W = h_{01}-h_{03} = U_2c_{\theta 2}-U_3c_{\theta 3}$

As $h_{01}=h_{02}$

$\Delta W = h_{02} - h_{03} = h_2-h_3+1/2(c^2_2-c^2_3)$

$= \; 1/2(U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$ ----------(2.0 a)

Each term in eqn. (2.0 a) makes a contribution to the specific work done on the rotor. A significant contribution comes from the first term, namely ½(U^2_2-U^2_1) and is the main reason why the inward flow turbine has such an advantage over the outward flow turbine where the contribution from this term would be negative. For the axial flow turbine, where U2 = U1, of course no contribution to the specific work is obtained from this term. For the second term in eqn. (2.0a) a positive contribution to the specific work is obtained when w3 > w2. In fact, accelerating the relative velocity through the rotor is a most useful aim of the designer as this is conducive to achieving a low loss flow. The third term in eqn. (2.0a) indicates that the absolute velocity at rotor inlet should be larger than at rotor outlet so as to increase the work input to the rotor. With these considerations in mind the general shape of the velocity diagram shown

The Efficiency of a Radial Inflow Turbine

Radial turbine efficiency $= \frac{\Delta W}{\Delta h}$

The efficiency of a radial inflow turbine = work done / enthalpy

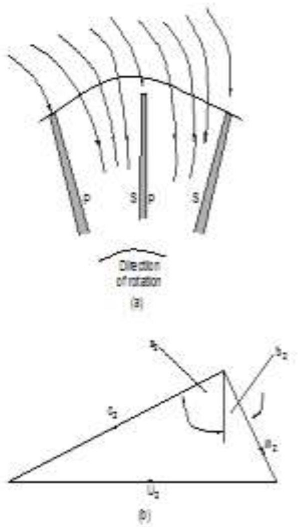


Figure: Optimum Flow Condition At Inlet To The Rotor. (A) Streamline Flow At Rotor Inlet;
P Is For Pressure Surface, S Is For Suction Surface. (B) Velocity Diagram For The Pitch wise Averaged Flow.

Theoretical Radial Inflow Turbine Efficiency

Calculation of radial turbine efficiency at 10m/s and flow entry 30 degree

Let us considering flow velocity 10 m/s

At the design point the absolute flow angle at rotor entry is 30 deg.

Rotor diameter is 247.65mm

According to angular velocity U2 =ω R2

$U_2 = \Pi ND \cot\alpha_2 /60$

Rotation per minute is N = $\frac{10x60x1000}{3.14x247.65}$

N = 319.55

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is U_2 = 10 m/s

Referring to Figure $W_2 = U_2 \cot\alpha_2$

W_2=17.32m/s

And c_2 = U2 sin α_2

c_2=5 m/s

$c^2_3=W^2_3-U^2_3=(2x17.32)^2-(1/2x10)^2=1199-25=1174m^2/s^2$

Hence,$U^2_2-U^2_3= U^2_2(1-1/4)=10^2(3/4)=75\ m^2/s^2$

$w^2_3 -w^2_2=3x\ w^2_2=899.94\ m^2/s^2$

$c^2_2-c^2_3 =862.12\ m^2/s^2$

All values substuting in work done equation

$\Delta W = h_{02} - h_{03} = h_2 - h_3 + 1/2(c^2_2-c^2_3)$

$= 1/2 \quad (U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W = \frac{75+899.94+862.12}{2}$

ΔW = 918.06 J/kg

Radial turbine efficiency = $\frac{\Delta W}{\Delta h}$ enthalpy

Δh=1006.43 J/kg

Ƞ=918.06x100/1006.43= 91.22%

Calculation of radial turbine efficiency at 10m/s and flow entry 25 degree

Let us considering flow velocity 10 m/s

At the design point the absolute flow angle at rotor entry is 25 deg.

Rotor diameter is 247.65mm

According to angular velocity U2 =ω R2

$U_2 = \Pi ND \cot\alpha_2 /60$

Rotation per minute is N=$\frac{10x60x1000}{3.14x247.65}$

N= 638.44

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is U2=10 m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

W_2=21.44m/s

and c_2 =U2 sin α_2

c_2=4.22 m/s

$c^2_3=W^2_3-U^2_3=(2x21.44)^2-(1/2x10)^2=1838.70-25=1813.70m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-1/4)=10^2(3/4)=75\ m^2/s^2$

$w^2_3 -w^2_2=3x\ w^2_2=1379\ m^2/s^2$

$c^2_2-c^2_3=481.56\ m^2/s^2$

all values substuting in work done equation

$\Delta W = h_{02} - h_{03} = h_2 - h_3 + 1/2\ (c^2_2-c^2_3) = ½\ (U^2_2-U^2_3) -(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W = \frac{75+1379+481.58}{2}$

ΔW = 967.78 J/kg

Radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

enthalpy Δh=1006.43 J/kg

Ƞ=967.78x100/1006.43= 96.16%

Calculation of radial turbine efficiency at 10m/s and flow entry 20 degree

Let us considering flow velocity 10 m/s

At the design point the absolute flow angle at rotor entry is 20 deg.

rotor diameter is 247.65mm

according to angular velocity U2 =ω R2

$U_2=\Pi ND \cot\alpha_2 /60$

Rotation per minute is $N=\frac{10x60x1000}{3.14x247.65}$

N= 1105.22

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is $U_2=10$ m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

$W_2=27.47$m/s

and $c_2 =U2 \sin \alpha_2$

$c_2=3.42$ m/s

$c^2_3=W^2_3-U^2_3=(2x27.47)^2-(1/2x10)^2=1509.20-25=1484.20m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-1/4) =10^2(3/4) =75 m^2/s^2$

$w^2_3 -w^2_2=3x w^2_2=1806 m^2/s^2$

$c^2_2-c^2_3=215.53 m^2/s^2$

all values substuting in work done equation

$\Delta W =h_{02} - h_{03} =h_2-h_3+1/2(c^2_2-c^2_3)= ½ (U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W =\frac{75+1806+215.53}{2}$

ΔW = 989.01 J/kg

Radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

enthalpy Δh=1006.43 J/kg

Ƞ=989.01x100/1006.43= 98.27%

Calculation of radial turbine efficiency at 7m/s and flow entry 30 degree

Let us considering flow velocity 7 m/s

At the design point the absolute flow angle at rotor entry is 30 deg.

rotor diameter is 247.65mm

according to angular velocity U2 =ω R2

$U_2=\Pi ND \cot\alpha_2 /60$

Rotation per minute is $N=\frac{10x60x1000}{3.14x247.65}$

N= 223.08

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is $U_2=7$ m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

$W_2=12.12$m/s

and $c_2 =U2 \sin \alpha_2$

$c_2=3.5$m/s

$c^2_3=W^2_3-U^2_3=(2x12.12)^2-(1/2x7)^2=587.57-12.25=575.32m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-1/4)=7^2(3/4)=36.75 m^2/s^2$

$w^2_3 -w^2_2=3x w^2_2=36.36m^2/s^2$

$c^2_2-c^2_3=1762.61m^2/s^2$

all values substuting in work done equation

$\Delta W =h_{02} - h_{03} =h_2-h_3+1/2(c^2_2-c^2_3) = ½ (U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W =\frac{36.75+36.36+1762.61}{2}$

ΔW = 917.86 J/kg

Radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

enthalpy Δh=1006.43 J/kg

Ƞ=917.86x100/1006.43= 91.20%

Calculation of radial turbine efficiency at 7m/s and flow entry 25 degree

Let us considering flow velocity 7 m/s

At the design point the absolute flow angle at rotor entry is 25 deg.

rotor diameter is 247.65mm

according to angular velocity U2 =ω R2

$U_2=\Pi ND \cot\alpha_2 /60$

Rotation per minute is $N=\frac{10x60x1000}{3.14x247.65}$

N= 445.90

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is $U_2=7$ m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

$W_2=15.01$m/s

and $c_2 =U2 \sin \alpha_2$

$c_2=2.35$m/s

$c^2_3=W^2_3-U^2_3=(2x15.01)^2-(1/2x7)^2=901.20-12.25=888.95m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-1/4)=7^2(3/4)=36.75 m^2/s^2$

$w^2_3 -w^2_2=3x w^2_2=675.90m^2/s^2$

$c^2_2-c^2_3=1224.71m^2/s^2$

all values substuting in work done equation

$\Delta W =h_{02} - h_{03} =h_2-h_3+1/2(c^2_2-c^2_3)$

$= ½ (U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W =\frac{36.75+675.90+1224.71}{2}$

ΔW = 968.68 J/kg

Radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

enthalpy Δh=1006.43 J/kg

Ƞ=968.68x100/1006.43= 96.25%

Calculation of radial turbine efficiency at

7m/s and flow entry 20 degree

Let us considering flow velocity 7 m/s

At the design point the absolute flow angle at rotor entry is 20 deg.

rotor diameter is 247.65mm

according to angular velocity U2 =ω R2

$U_2=\Pi ND \cot\alpha_2 /60$

Rotation per minute is $N=\frac{10x60x1000}{3.14x247.65}$

N= 815.25

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is $U_2=7$ m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

$W_2=19.23$m/s

and $c_2 =U2 \sin \alpha_2$

$c_2=2.39$m/s

$c^2_3=W^2_3-U^2_3=(2x19.23)^2-(1/2x7)^2=1479.17-12.25=1466.92m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-1/4)=7^2(3/4)=36.75\ m^2/s^2$

$w^2_3 -w^2_2=3x\ w^2_2=1109.37m^2/s^2$

$c^2_2-c^2_3=833.72m^2/s^2$

all values substuting in work done equation

$\Delta W =h_{02} - h_{03} =h_2-h_3+1/2(c^2_2-c^2_3)=$ ½ $(U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W =\frac{36.75+1109.37+833.72}{2}$

$\Delta W = 989.92$J/kg

Radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

enthalpy $\Delta h=1006.43$ J/kg

$\eta=989.92x100/1006.43= 98.36\%$

Calculation of radial turbine efficiency at 4m/s and flow entry 30 degree

Let us considering flow velocity 4 m/s

At the design point the absolute flow angle at rotor entry is 30deg.

rotor diameter is 247.65mm

according to angular velocity U2 =ω R2

$U_2=\Pi ND \cot\alpha_2 /60$

Rotation per minute is $N=\frac{10x60x1000}{3.14x247.65}$

N= 127.80

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is $U_2=4$ m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

$W_2=6.92$m/s

and $c_2 =U2 \sin \alpha_2$

$c_2=2$m/s

$c^2_3=W^2_3-U^2_3=(2x6.92)^2-(1/2x4)^2=191.54-4=187.54m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-1/4)=4^2(3/4)=12m^2/s^2$

$w^2_3 -w^2_2=3x\ w^2_2=143.65m^2/s^2$

$c^2_2-c^2_3=1679.07m^2/s^2$

all values substuting in work done equation

$\Delta W =h_{02} - h_{03} =h_2-h_3+1/2(c^2_2-c^2_3)=$ ½ $(U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W =\frac{36.75+1109.37+833.72}{2}$

$\Delta W = 917.36$J/kg

Radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

enthalpy $\Delta h=1006.43$ J/kg

$\eta=917.36x100/1006.43= 91.15\%$

Calculation of radial turbine efficiency at 4m/s and flow entry 25 degree

Let us considering flow velocity 4 m/s

At the design point the absolute flow angle at rotor entry is 25 deg.

rotor diameter is 247.65mm

according to angular velocity U2 = ω R2

$U_2=\Pi ND \cot\alpha_2 /60$

Rotation per minute is $N=\frac{10x60x1000}{3.14x247.65}$

N= 254.80

Now find the radial turbine efficiency = $\frac{\Delta W}{\Delta h}$

The blade tip speed is $U_2=4$ m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

$W_2=8.57$m/s

and $c_2 =U2 \sin \alpha_2$

$c_2=1.69$m/s

$c^2_3=W^2_3-U^2_3=(2x8.57)^2-(1/2x4)^2=293.77-4=287.77m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-$

Axis of turbine	Vertical
Types of draft tube	Elbow type
Inlet runner diameter	247.65 mm
Outlet runner diameter	163.94 mm
No. of blades	10
Inlet blade angle	37.17
Outlet blade angle	17.73
guide vane angle	25
No of guide vanes	18
Blade width at inlet	26.56 mm
Blade width at outlet	108 mm
Inlet guide vane diameter	431.8 mm
Outlet guide vane diameter	247.65 mm

$1/4)=4^2(3/4)=12 m^2/s^2$

$w^2_3 - w^2_2= 3x\ w^2_2= 77.13m^2/s^2$

$c^2_2-c^2_3= 1824.1m^2/s^2$

all values substuting in work done equation

$\Delta W = h_{02} - h_{03} = h_2 - h_3 + 1/2(c^2_2 - c^2_3)$

$= \frac{1}{2} (U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W = \frac{36.75+77.13+1824.1}{2}$

$\Delta W = 968.99 J/kg$

Radial turbine efficiency $= \frac{\Delta W}{\Delta h}$

enthalpy Δh=1006.43 J/kg

η=968.99x100/1006.43= 96.28%

Calculation of radial turbine efficiency at 4m/s and flow entry 20 degree

Let us considering flow velocity 4 m/s

At the design point the absolute flow angle at rotor entry is 20 deg.

rotor diameter is 247.65mm

according to angular velocity $U2 = \omega R2$

$U_2 = \Pi N D \cot\alpha_2 /60$

Rotation per minute is $N=\frac{10x60x1000}{3.14x247.65}$

N= 442.08

Now find the radial turbine efficiency $= \frac{\Delta W}{\Delta h}$

The blade tip speed is U_2=4 m/s

Referring to Figure $W_2=U_2 \cot\alpha_2$

W_2=10.98 m/s

and $c_2 = U2 \sin \alpha_2$

c_2=1.36m/s

$c^2_3=W^2_3-U^2_3=(2x10.98)^2-(1/2x4)^2=482.24-4=478.24m^2/s^2$

Hence, $U^2_2-U^2_3= U^2_2(1-1/4)=4^2(3/4)=12m^2/s^2$

$w^2_3 -w^2_2=3x\ w^2_2=241.12m^2/s^2$

$c^2_2-c^2_3=1700.35m^2/s^2$

all values substuting in work done equation

$\Delta W = h_{02} - h_{03} = h_2 - h_3 + 1/2(c^2_2 - c^2_3)$

$= \frac{1}{2} (U^2_2-U^2_3)-(w^2_2-w^2_3)+ (c^2_2-c^2_3)$

$\Delta W = \frac{36.75+241.12+1700.35}{2}$

$\Delta W = 989.11 J/kg$

Radial turbine efficiency $= \frac{\Delta W}{\Delta h}$

enthalpy Δh=1006.43 J/kg

η=989.11x100/1006.43= 98.28%

Modeling of radial turbine

Radial Turbine Geometry Is Designed With the Help of CATIA V5R20We are taking values for Hydraulic turbines are the machines to Radial Turbine Geometry. Reaction turbines are those turbines which operate under hydraulic pressure energy and part of kinetic energy. In this case, the water reacts with the vanes as it moves through the vanes and transfers its pressure energy to the vanes so that the vanes move in turn rotating the runner on which they are mounted. Hydraulic turbines are the machines that convert the hydraulic energy into electricity, which are produced since many years ago. However, reaching such efficiencies is a difficult task and it requires a high engineering effort because hydraulic turbines are usually unique products which must be designed for determined local conditions.

Geometric Modeling:

Radial Turbine Geometry is designed with the help of CATIA V5R20 software. These are the basic component of Radial Turbine.

1. Runner
2. Guide vanes
3. Spiral casing
4. Draft tube

Runner is designed in CATIA V5R20 software and guide vanes, spiral casing and draft tube is designed in CATIA V5R20. Finally these components are assembled in a single radial turbine with the help of CATIA Assembly Design.

Table: Specification of Turbine

Runner is designed in CATIA V5R20 software and guide vanes, spiral casing and draft tube is designed in CATIA V5R20

Fig: Geometry Of Spiral Casing

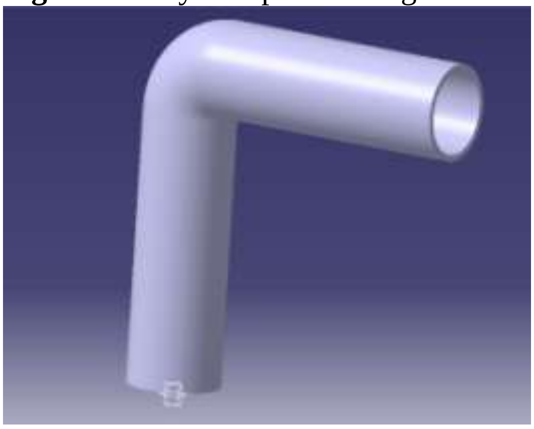


Fig: Draft Tube

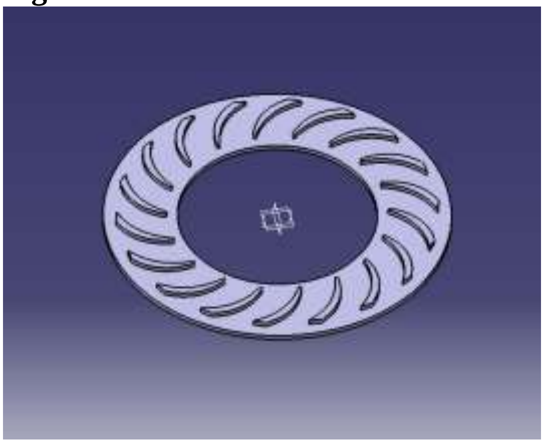


Fig: Geometry of Guide Vanes

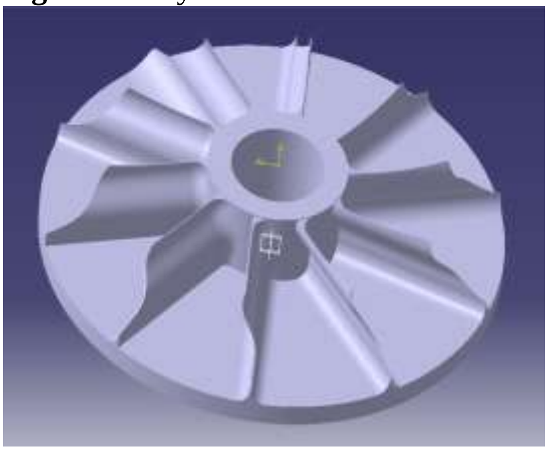


Fig: Runner

The components are assembled in a single radial turbine with the help of CATIA Assembly Design

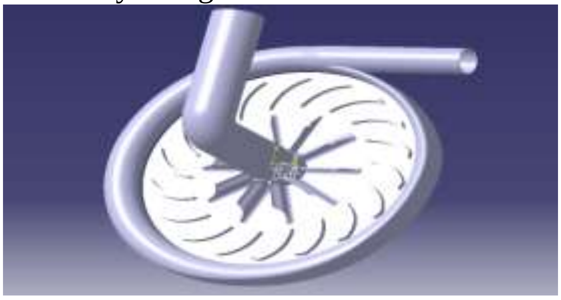


Figure: Complete Assembled Geometry of Radial Turbine

CFD Analysis of Radial Turbine

The analysis is carried in fluent by importing the meshed file saved in ANSYS ICEM CFD. The steps that are followed are given below which include all the conditions and the boundaries values for the problem statement. In this project three different cases are taken through changing the guide vane angle and for each angle three different inlet velocities are taken. Results can get through streamlines, vector plot and velocity and pressure contour.

Assumptions

The following **assumptions** were taken for simulation:

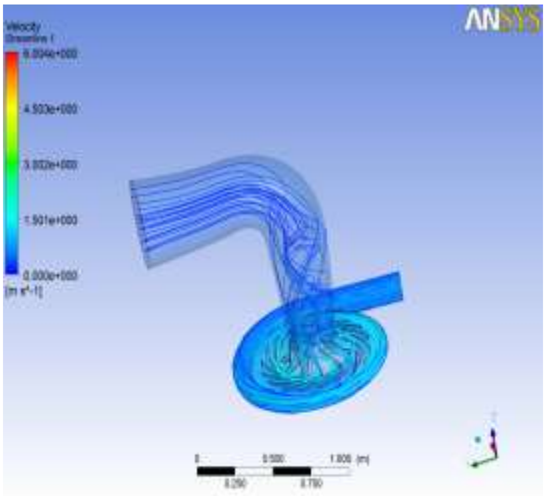
- The walls of the casing were assumed to be smooth hence any disturbances in flow due to roughness of the surface were neglected.
- The friction co-efficient for all surfaces were set to 0, hence friction between the walls and fluid was neglected.
- Steady state conditions and compressible fluid flow.

Solution parameters

- 3-D double precision solver used to solve for simulation.
- Multiple reference frame technique used to simulate the pump performance.
- Natural air is taken as working fluid.
- Standard K-omega simulation model is used for turbulence modelling.

- Convergence criteria for continuity, velocity and turbulence parameters was set to10^{-3}
- Second order scheme is used for pressure correction as well as for solving momentum, turbulent kinetic energy and turbulence dissipation rate.
- Simple scheme is used for pressure velocity coupling

Streamline and Vector Plot of Radial Turbine



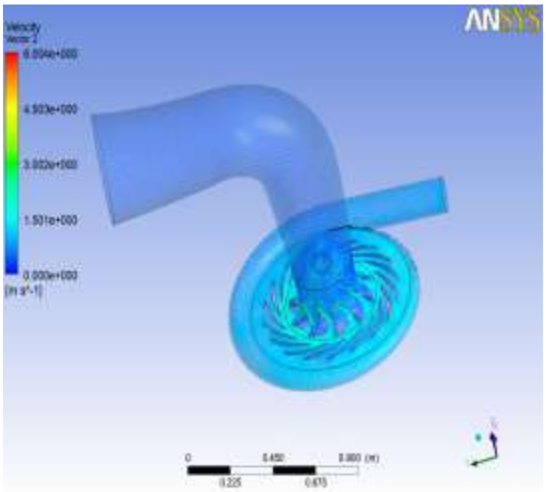
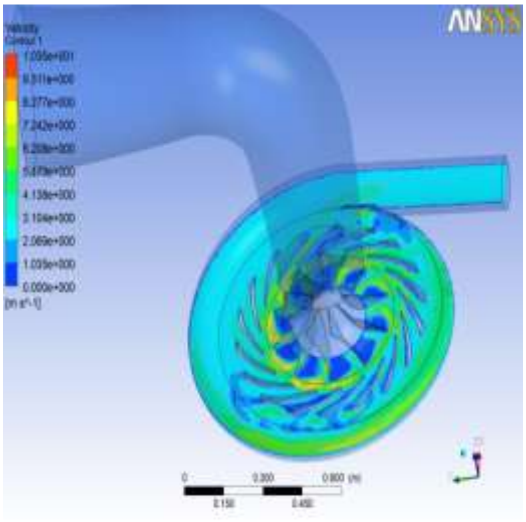


Figure: Streamline and Vector Plot Of Fluid Flow in Radial Turbine

Velocity And Pressure Contour For Different Cases

Case 1.1

Velocity and pressure distribution at 10 m/s velocity of fluid flow for 30 degree guide vane angle and 319.55 rpm at inlet



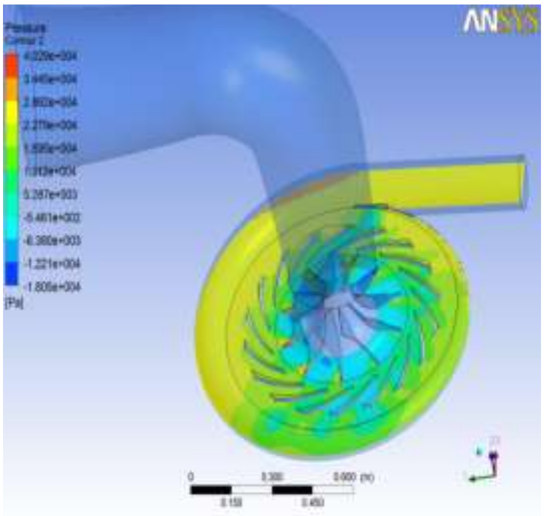


Figure: Velocity and Pressure Contour Of Fluid Flow for 10m/s At Inlet

Case 1.2

Velocity and pressure distribution at 7 m/s velocity of fluid flow for 30 degree guide vane angle and 223.08 rpm at inlet

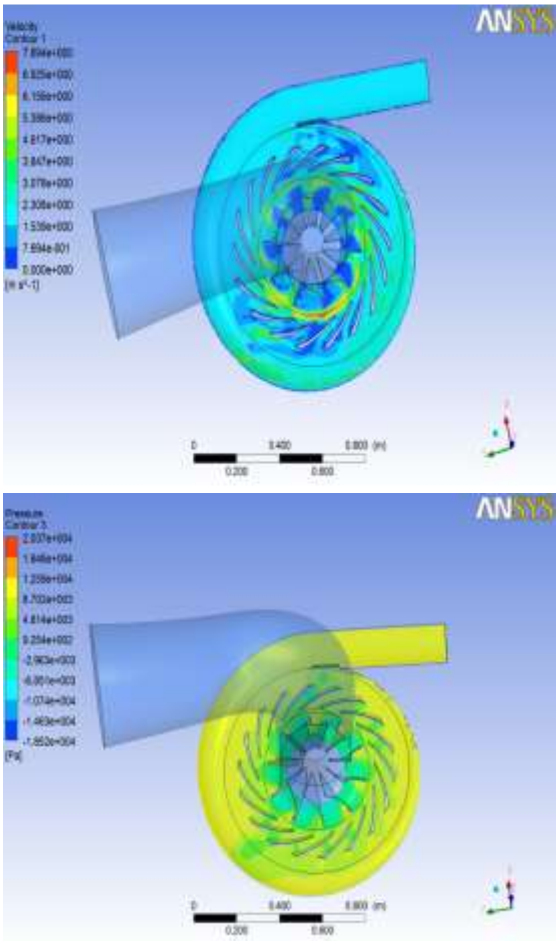


Figure: Velocity Contour of Fluid Flow For 7m/s At Inlet

Case 1.3

Velocity and pressure distribution at 6 m/s velocity of fluid flow for 30 degree guide vane angle and 127.80 rpm at inlet

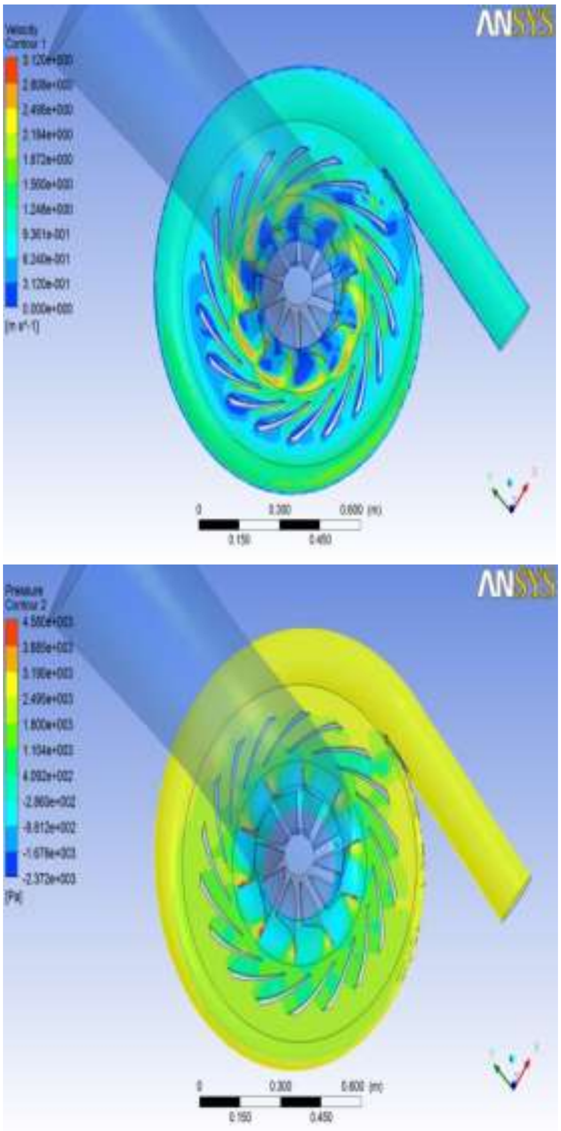
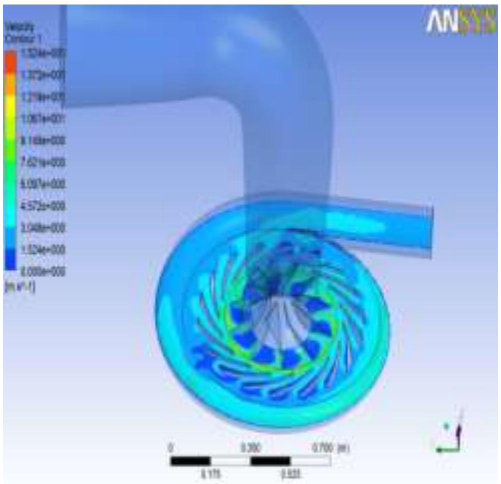


Figure: Velocity And Pressure Contour Of Fluid Flow For 4 m/s At Inlet

Case 2.1

Velocity and pressure distribution at 10 m/s velocity of fluid flow for 25 degree guide vane angle and 638.66 rpm at inlet



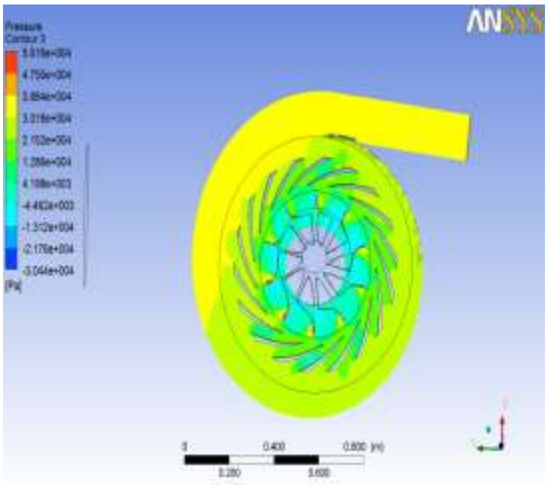
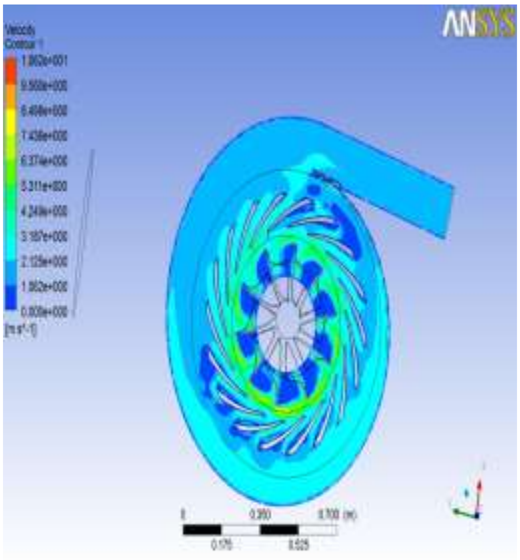


Figure: Velocity And Pressure Contour Of Fluid Flow For 10 m/s At Inlet

Case 2.2

Velocity and pressure distribution at 7 m/s velocity of fluid flow for 25 degree guide vane angle and 665.90 rpm at inlet



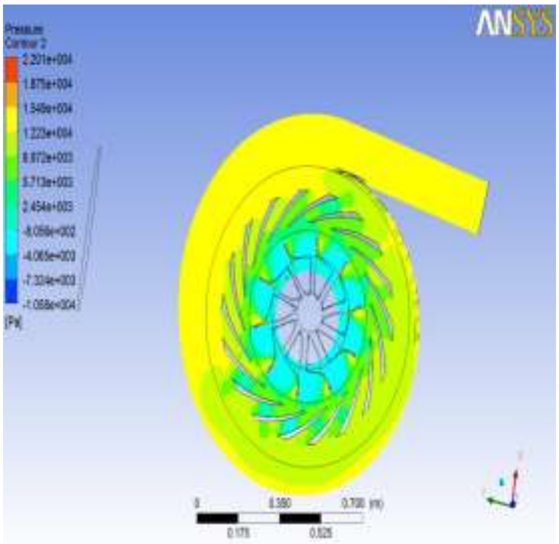
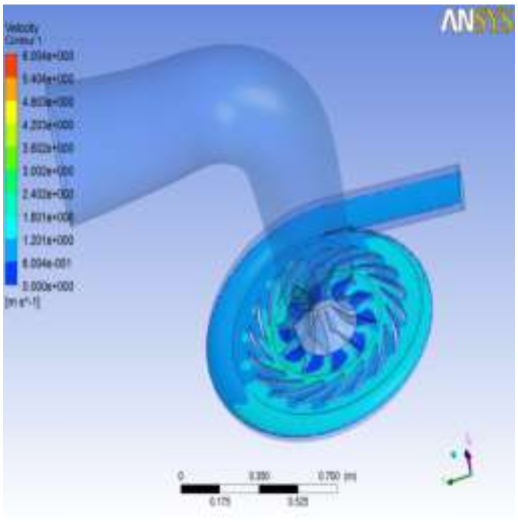


Figure: Velocity Contour Of Fluid Flow For 7 m/s At Inlet

Case 2.3

Velocity and pressure distribution at 6 m/s velocity of fluid flow for 25 degree guide vane angle and 256.80 rpm at inlet



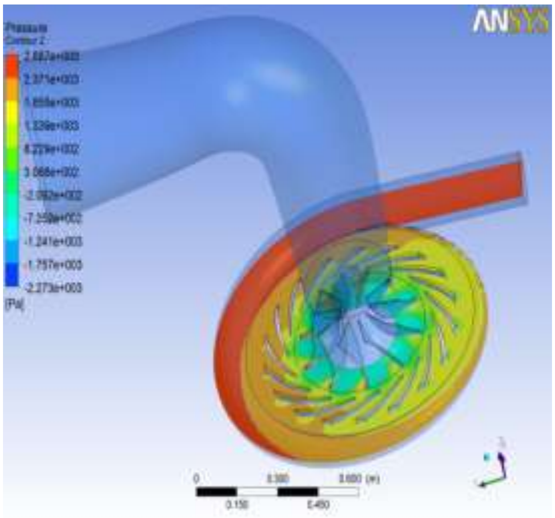
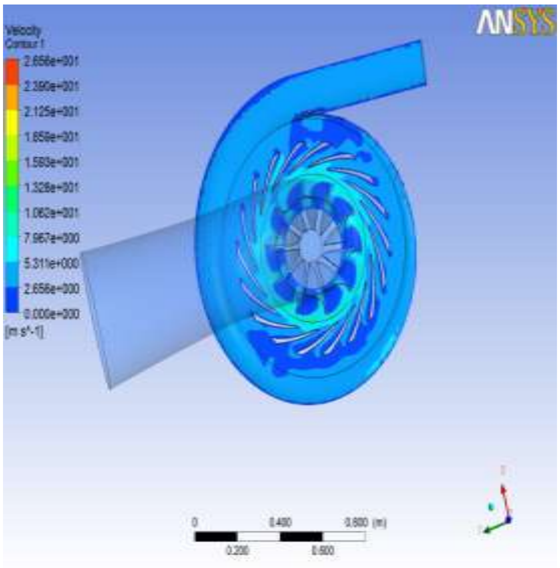


Figure : Velocity Contour Of Fluid Flow For 4 m/s At Inlet

Case 3.1 Velocity and pressure distribution at 10 m/s velocity of fluid flow for 20 degree guide vane angle and 1105.22 rpm at inlet



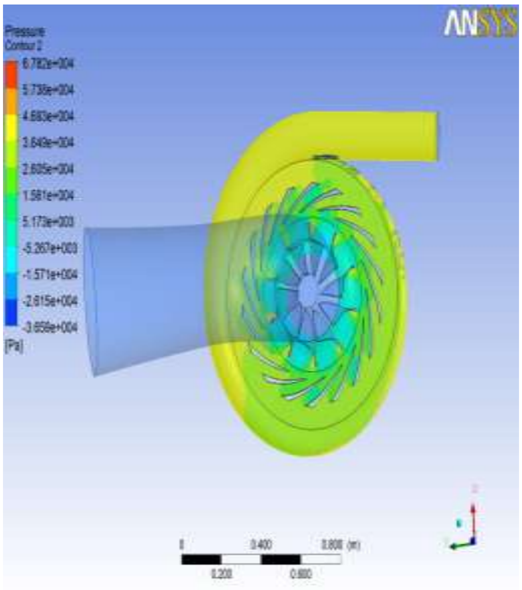
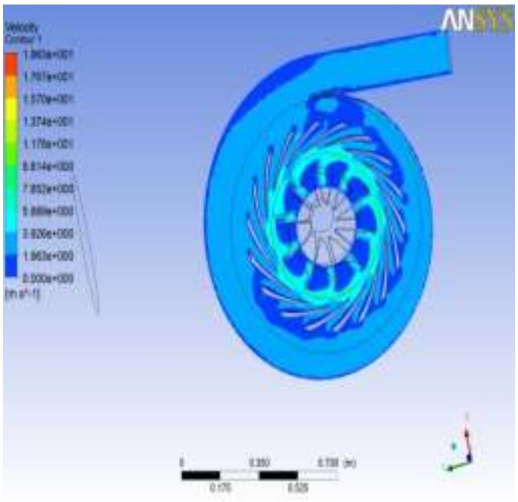


Figure:velocity contour of fluid flow for 10 m/s at inlet

Case 3.2 Velocity and pressure distribution at 7 m/s velocity of fluid flow for 20 degree guide vane angle and 815.25 rpm at inlet



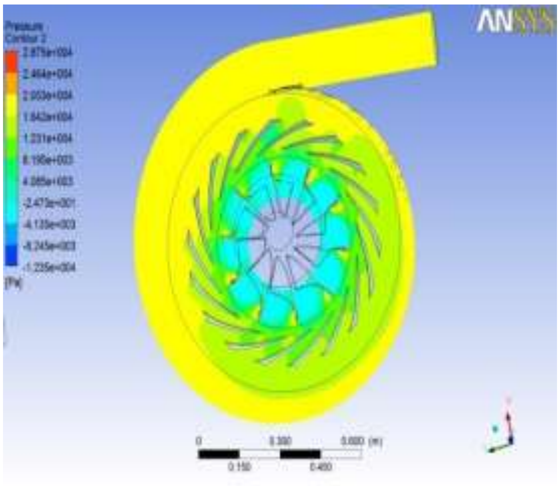
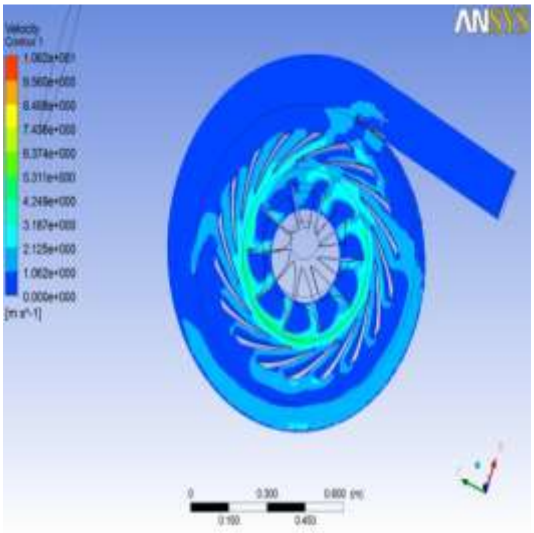


Figure: Velocity Contour Of Fluid Flow For 7 m/s At Inlet

Case 3.3 Velocity and pressure distribution at 6 m/s velocity of fluid flow for 20 degree guide vane angle and 662.08 rpm at outlet



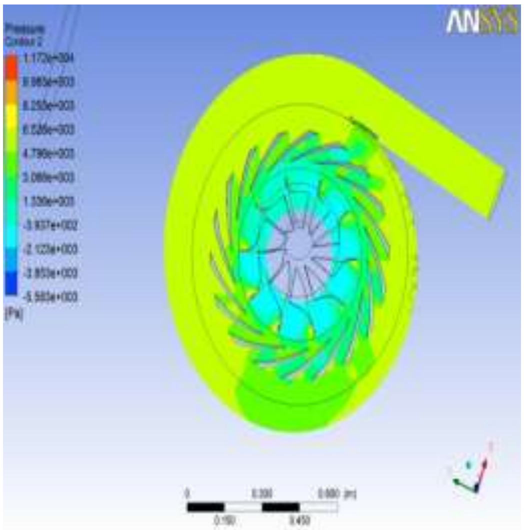


Figure: Velocity Contour Of Fluid Flow For4 m/s At Inlet

Colour Analysis:

These are the different velocity and pressure colours for three cases of velocities for guide vane angle 30, 25 and 20. These colourss are showing the distribution of velocity and pressure of fluid inside the radial turbine. According to these colours, it implies that velocity and pressure distribution in radial turbine is under acceptable condition.

STRUCTURE SIMULATION OF GUIDE VANES

Mesh Generation of Guide Vanes for Structural Domain

Mesh type	Eleme nt size	Physical preferenc e

Mapped face/tetrahedron	5 mm	Mechanical

Table : Mesh Input Detail

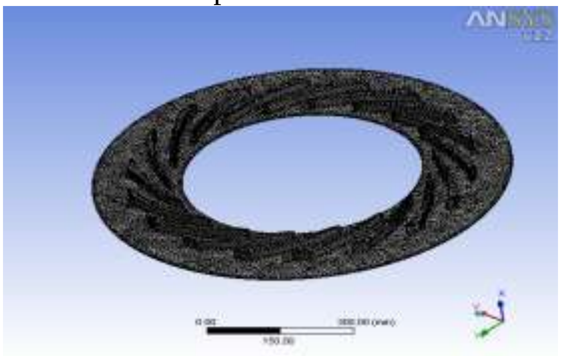


Figure : Generated Mesh Of Guide Vanes

Generated output mesh detail for guide vane in structure domain

No. of nodes	No. of elements
281396	139555

Table6.4 : Mesh Output Detail

ASSUMPTIOS

- No rotation is there.
- Linear analysis is being done.
- Static structure analysis is being done.

Sr. No.	G.V. A	RPM	Efficiency (T)	Efficiency (CFD)
1.	30	319.55	91.22%	90.28%
2.	25	638.44	96.16%	95.51%
3.	20	1105.22	98.27%	96.15%

BOUNDARY CONDITIONS

- Applied pressure on blade face is 36690 Pa.
- Environmental temperature is 22 C.

Guide VanesFor Static Structural ANSYS

Sr. No.	G.V. A	RPM	Efficiency (T)	Efficiency (CFD)
1.	30	319.55	91.22%	90.28%
2.	25	638.44	96.16%	95.51%
3.	20	1105.22	98.27%	96.15%

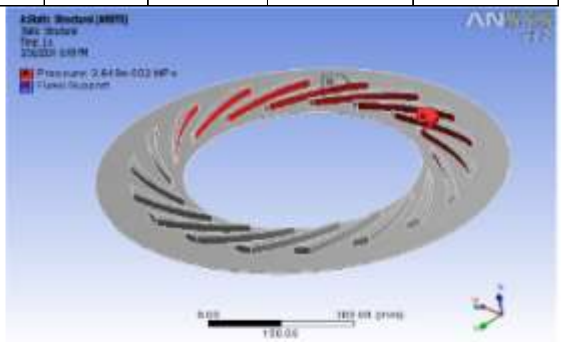


Figure : Boundary Conditions

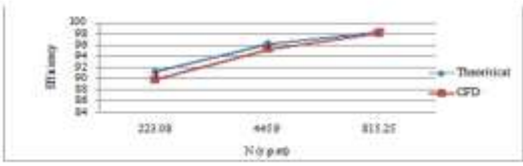
RESULTS

In structure analysis results are achieved through analyzing the von misses stress, total

Deformation and fatigue tool.

Characteristic Curve Verification and Comparison Between Theoretical And CFD Results

At 10 m/s

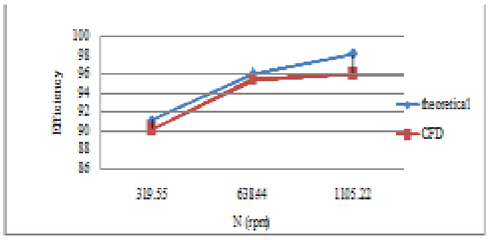


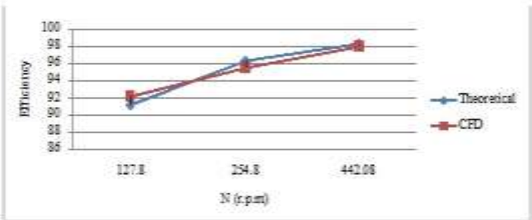
At 7 m/s

At 4 m/s

Sr. No.	G.V. A	RPM	Efficiency (T)	Efficiency (CFD)

1.	30	127.80	91.15%	92.13%
2.	25	254.80	96.28%	95.47%
3.	20	442.08	98.28%	97.97%





CONCLUSION

This paper brought out the validation of CFD results with theoretical results. The maximum efficiency regime indicated by both approaches is nearly same. Reason for slight difference of efficiency computed by theoretically and CFD method can be human errors and due to discrization of domains and solution of differential equations in computational methods. Hence the result obtained are fairly matching, however streamlines flow in some reasons have some turbulence which is due to occurrence of loses and losses are not considered very precisely. Prediction of turbine performance by CFD gives the idea to know the flow behavior inside the turbine model and get the information about the intricacy of flow pattern. After getting best model according to the CFD analysis, stress analysis on runner blades and guide vane blades is being done. In this section fatigue analysis is checked and found the life, damage and factor of safety for that model. According to structure analysis blades are having maximum life and damage is minimum. In other hand factor of safety is also maximum which implies that this model is completely safe. By observing the thermal results, thermal flux is more super alloy than titanium alloy. So using super alloy is better than titanium alloy.

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