Scientific Journal Impact Factor (SJIF) : 3.518
International Journal of Modern Trends in Engineering and Research
IJMTER
ISSN (ONLINE) : 2349 - 9745
ISSN (PRINT) : 2393 - 8161

PERFORMANCE ANALYSIS OF SINGLE STAGE CENTIFUGALOMPRESSOR BY CHANGING ITS INLET BLADE ANGLE

Dhruv joshi[1], Nawaz Qureshi [2] and Jigar V. Bhalodiya[3]
[1]mechanical engineer, nobel engineering College
[2,3] Dr. subhash technical campus

Abstract-This Project shows the change in Performance of the Centrifugal Compressor by changing its inlet blade angle with medium pressure ratio centrifugal Compressor. Compute of parameter which got influenced by change of inlet blade angle. Determine the performance of centrifugal compressor by change of its blade angle. The design code provides a basis on which the design of the Compressor can be modeled by varying the key parameters which include both aerodynamic and geometric details. Code than will predict first cut solution. The design which models the flow in an Impeller, and an annular bend takes into consideration various loss models occurring in the complex flow of a Centrifugal Compressor. Whole compressor is modeled & Assemble in Creo 3.0 and validated in ANSYS14.5.
Keywords—Centrifugal compressor, blade, impeller

I. INTRODUCTION

Compressor is mechanical device used to increase the pressure of the gas. Compressors used for producing high pressure air are called air compressors. Air is drawn from the atmosphere during suction, compressed to require pressure and then delivered to the receiver.

If the compression is done in conventional cylinder with closely fitted piston making reciprocating motion, then compressor is called reciprocating type of compressor. External work must be supplied from prime mover to the compressor to achieve the require compression. Since The Process of Compressing Fluid Requires The Work Should Be Done On It, Compressor Has To Be Driven By Prime Mover, Such As Electric Motor Or Engine. A Machine Which Takes In Air During Suction Stroke At Low Pressure, Compresses It To High Pressure In Piston Cylinder Arrangement And Then Delivers It To Some Storage Vessel Is Known As Reciprocating Engine.

In the various ways in the generation of mechanical power turbine is best. Absence of piston engines and wrapping rubber, this means that problem in equilibrium condition during the process is reduced. The requirement for lubricating oil is also getting smaller and reliability of a system can be high.

Advantage of using this first in an effort to realize an electric power generation with water as the working fluid from the system. The system is better known as the Steam Turbine. The use of steam turbine is the first time in the early 20th century and has become the primary tool used in the generation of electrical power source. The power generated can reach 1000MW with efficiency reaching 40 percent at this time. Although there are advantages in this steam turbines, but there are also some lacks.

Production of high-pressure steam and high temperature are very high costs involved in purchasing equipment and installation of boiler systems either conventional or nuclear reactors. In addition the use of gas fuel in the boiler is not used directly by the gas turbine fuel but is used as fuel to heat water to produce a fluid that is in the form of steam, based on these two ideas for direct use of fuel gas is constantly being developed. The development of this new system begins in the moments before the Second World War, called the Gas Turbine.

II. OBJECTIVE

The absolute velocity of the flow is axial and relative velocity is at an angle β_1 from the tangential direction. Thus the swirl or whirl component c_{t1}=0

$$\mathrm{Tan}\alpha_1 = \frac{c_1}{u_1} = \frac{c_{a1}}{u_1} \ldots\ldots\ldots(1.1)$$

Flow through axially straight inducer blades in the presence of IGV. The air angle at the inlet is called IGVs angle.

$$C_{t1} = u_1 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1.2)$$

$$work\ supply = u_2^2 - V_{w1} * u_1 .(1.3)$$

$$E = mu_1 \left(r_p \wedge (\gamma - 1/\gamma) - 1\right)..(1.4)$$

$$V_{w1} = \frac{u_2 \times 0.7125}{u_1 r_p^{0.4}} + C \ldots\ldots(1.5)$$

$$R_p = \frac{P_{02}}{P_{01}} = \text{pressure ratio}...(1.6)$$

Above equation has shown that by reducing the whirl angle velocity increases for the equal RPM. So by increasing the inlet blade angle from the 60 to 90 We can have reduce in whirl velocity and increase in power consumption.

Impellore is the hart of the machine, Impellor include complicated blade geometry.this geometry include many angle. Each and every angle has great influence on the performance.our concern is for the inlet angle.

Basically blade geometry can be classifide in three different class

(1) Inlet guide vaes
(2) Marideonal
(3) End trail(outlet guide vane)

Our mater of concern is IGV which have inlet angle. Table given below shows the changing property by change in the pre whirl angle.

Inlet Angle	V_1(m/s)	V_{r1}(relative velocity)	V_{w1}(whirl velocity)	V_{f1}(flow velocity)	Work Supply	Mach No.	R(pressure ratio)
60	15.7	27.2053	23.557	13.6027	75.201	0.86	1.89
65	13.6922	24.86	21.447	12.4093	75.227	0.83	1.89667
70	11.481	22.9621	19.877	11.481	75.251	0.6	1.897
75	11.1065	21.4561	18.577	10.7289	75.272	0.576	1.9
80	10.252	20.1925	17.487	10.09	75.289	0.576	1.89
85	9.587	19.1017	16.537	9.55	75.303	0.55	1.89

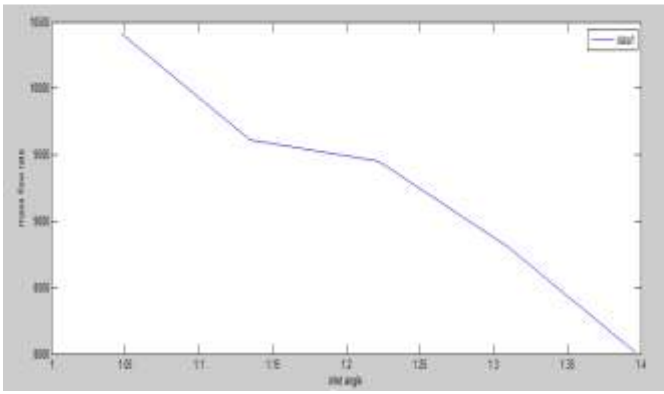

Fig 1.1 change of mass flow rate

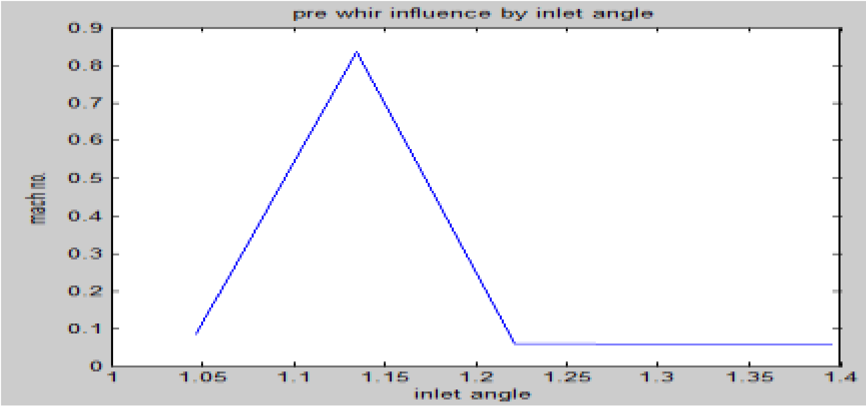

Fig1.2 change of mach no.

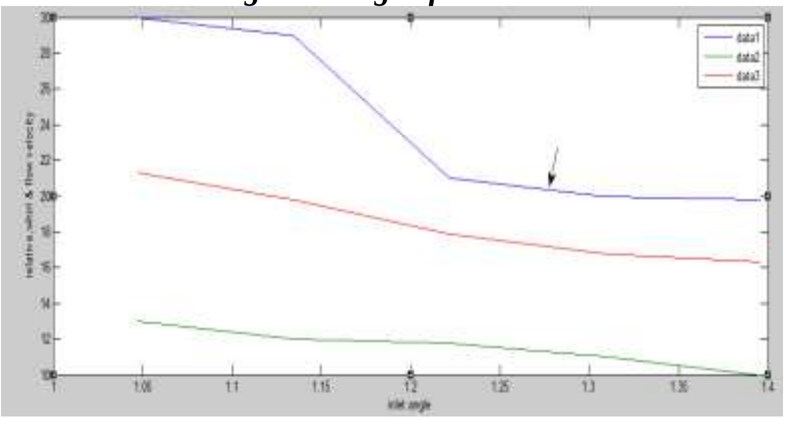

Fig1.3 change in flowvelocity,whirl velocity & relative veocity

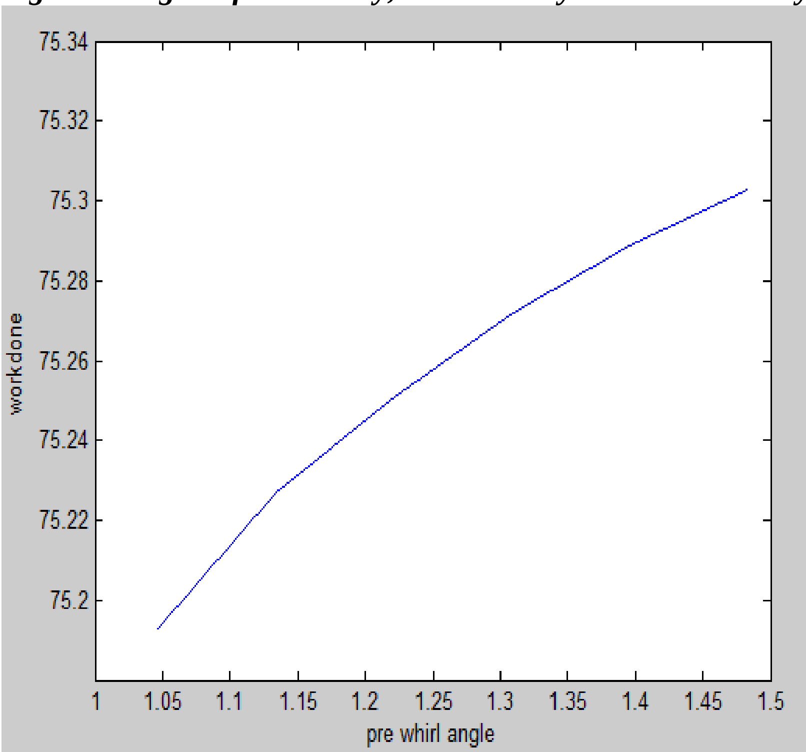

Fig1.4 change in work supply with

Th centrifugal compressor employed in gas turbines or jet engines run at very high speed and there is always a likelihood of formation of shock wave in the flow passage. To avoid this, the mach number at any point in the flow passage should not exceed more than unity. The maximum value of the Mach number corresponding to relative velocity at the inlet is $M_{r1} = \frac{C_{r1}}{\sqrt{T_1 R\gamma}} \ldots \ldots (1.8)$

III. MODELING

Fig 1.5 modeling of centrifugal compressor

IV. CFD ANALYSIS

We have proceed for the CFD analysis for the conformation we have extract CFD analysis for the five different compressor. 60^0,65^0,70^0,80^0,85^0 results are show below

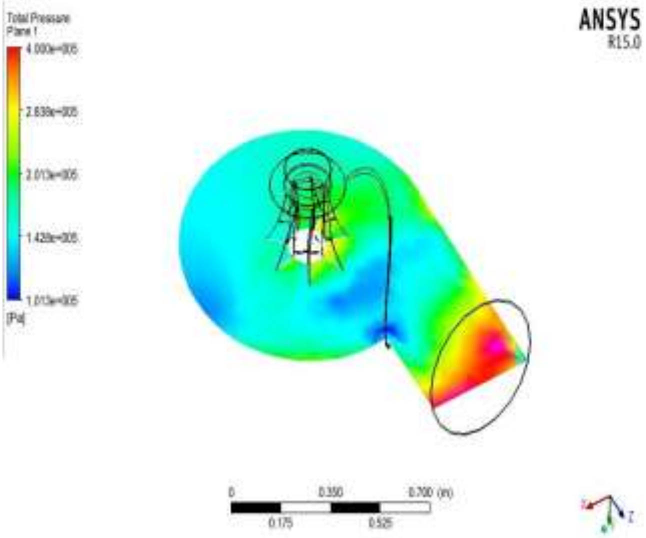

Fig 1.6 Pressure deviation Centrifugal Compressor inlet angle 60^0

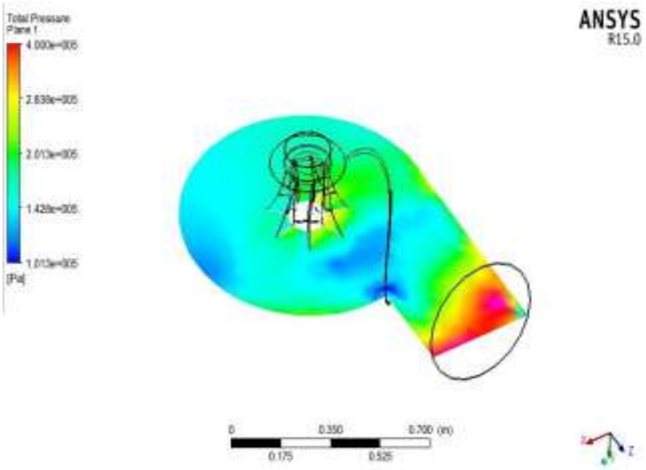

Fig 1.7 Pressure deviation Centrifugal Compressor inlet angle 65^0

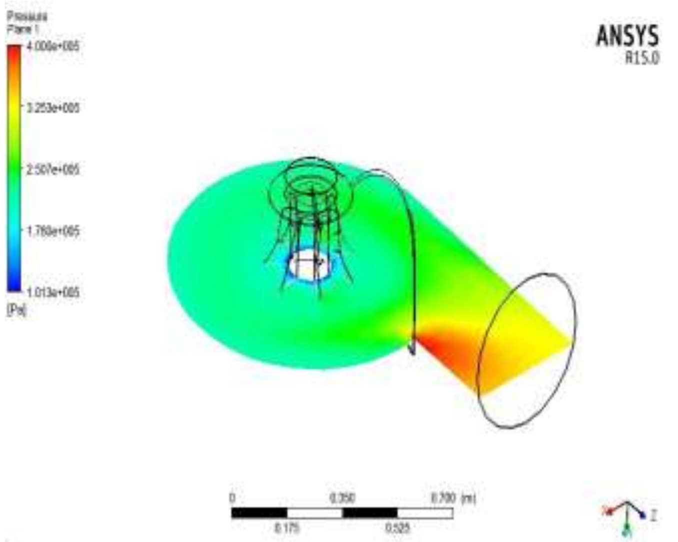

Fig 1.8 Pressure deviation Centrifugal Compressor inlet angle 70^0

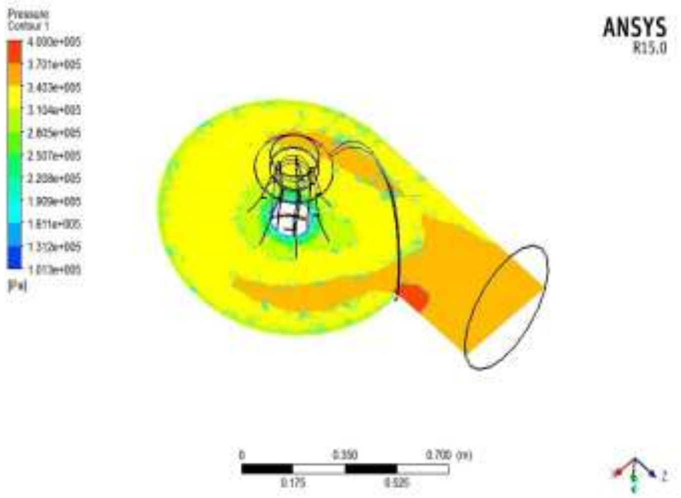

Fig 1.9 Pressure deviation Centrifugal Compressor inlet angle 80^0

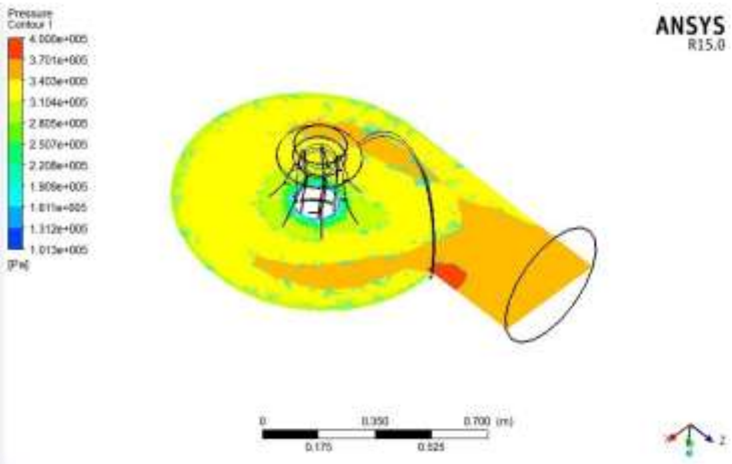

Fig 1.10 Pressure deviation Centrifugal Compressor inlet angle 85^0

Particular geometry as far the velocity triangle by increase in the pre-whirl angle whirl velocity increase from the pre whirl angle 0^0 to 90^0. After 90 & less than 0 work supply will increases.
Here increasing value of α_1 work supply increases and mass flow also increases

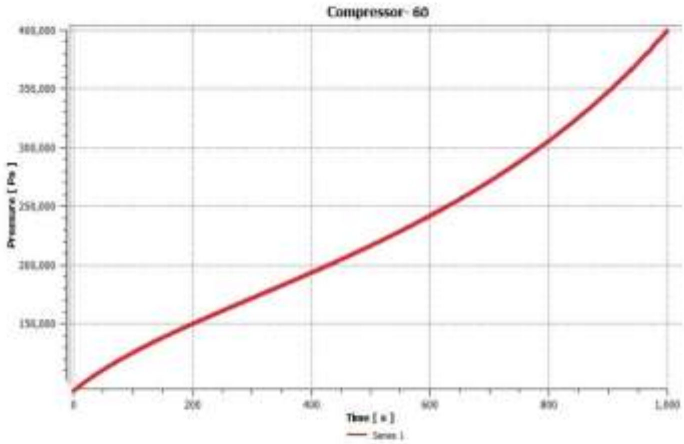

Figure 1.10 Pressure at outlet for blade angle 60^0

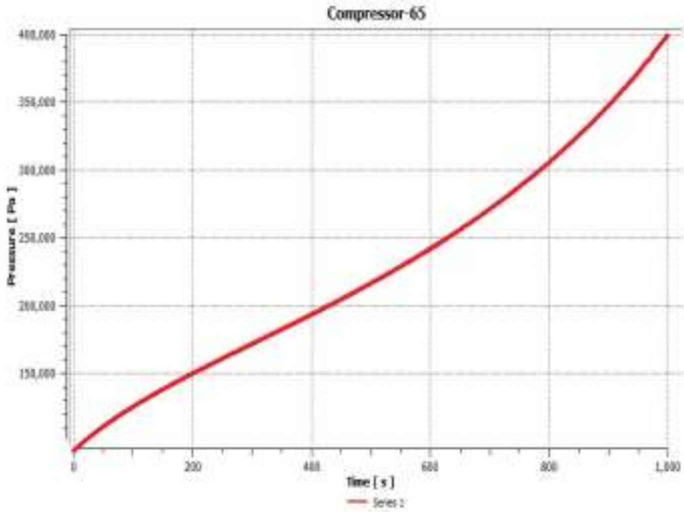

Figure 1.11 Pressure at outlet for blade angle 65^0

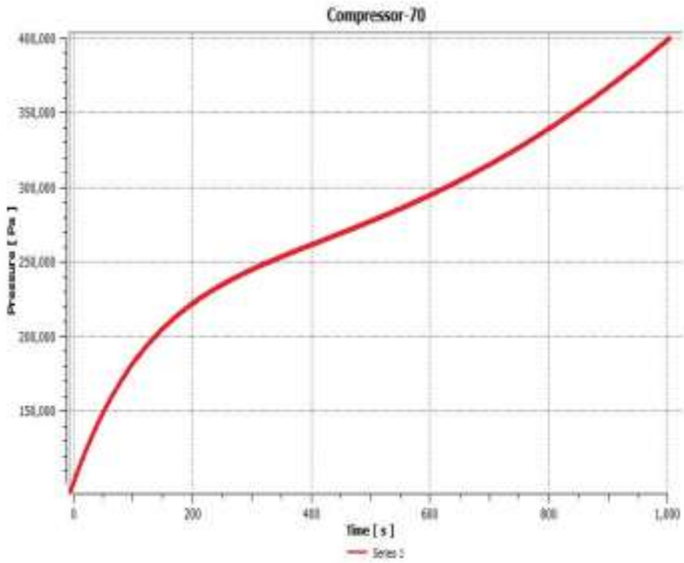

Figure 1.12 Pressure at outlet for blade angle 70^0

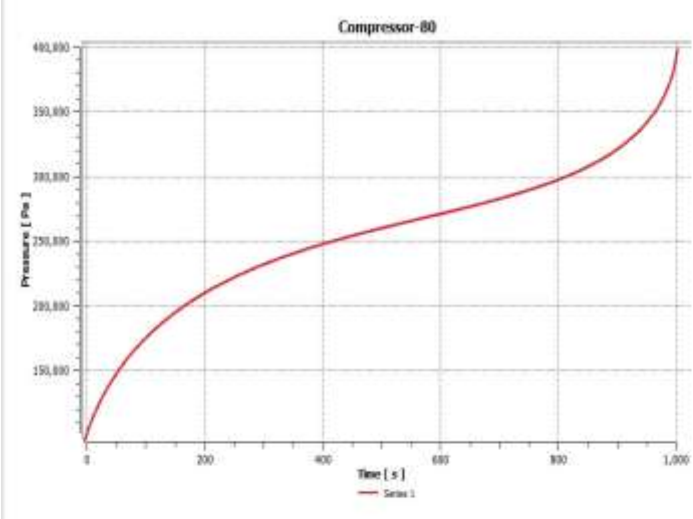

Figure 1.13 Pressure at outlet for blade angle 80^0

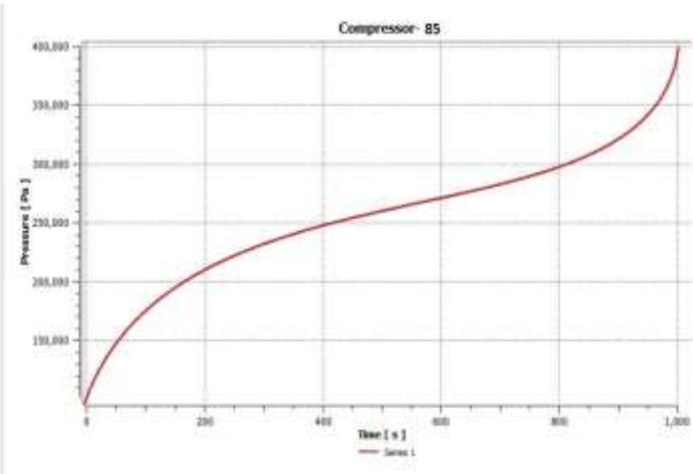

Figure 1.14 Pressure at outlet for blade angle 85^0

As from the diagram given above shows that by changing the pre whirl angle there is no change in the outlet pressure we can have the same pressure ratio at every pre-whirl angle, but for the each pre whirl angle work supply get increases means we can have the same pressure ratio with reduced work supply

V. OPTIMIZATION

Objective function z= (u^2-($v_{w1}^2 * \cos(\alpha_1)$)(minimize)
Function is trigonometric so it is continuous
α_1>30^0 because below 30^0 standard $\frac{r}{r_1}$ >0.5 which is not satisfied below 30^0.So our domain of experiment is [30 85]
By differentiating equation with respect to the α_1
dz/dα_1=($v_{w1}^2 * \sin(\alpha_1)$)…..(1.9)
For local maximum and minimum
z is maximum at 90^0 and minimum at 30^0

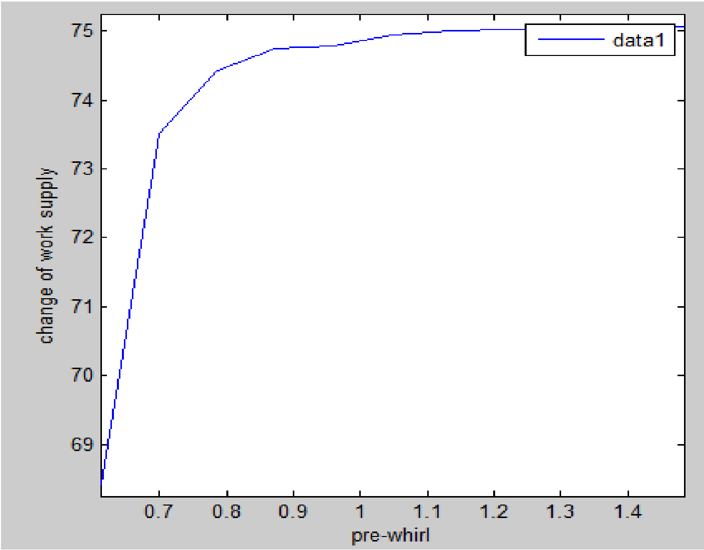

Fig 1.14 Change in work supply by change in pre-whirl

VI. CONCLUSION

By considering the compression pressure ratio don't shows any deflection by changing the pre whirl angle from the 30^0 to the 90^0. We have taken impeller with pre whirl angle 60^0, 65^0,70^0,80^0,85^0 and produce CFD analysis in the ANSYS 15.0. It shows similar result. We can evaluate the work supply for the centrifugal compressor. From the result we can see that by increasing pre whirl work supply increases but compression or pressure ratio don't change. We can have the best pre whirl by reducing pre whirl angle. Pre whirl is also able to reduce the shock losses in the centrifugal compressor.

REFERENCES

[1] "Optimal Design of a Centrifugal Compressor Impeller Using Evolutionary Algorithms" by Soo-Yong Cho,1 Kook-Young Ahn,2 Young-Duk Lee,2 and Young-Cheol Kim
[2] "Numerical Studies on the Effect of Impeller Blade Skew on Centrifugal Compressor Performance" by V.C. Arunachalam..
[3] "Empirical Design Considerations for Industrial Centrifugal Compressors" by Cheng Xu and Ryoichi S. Amano
[4] "Meridional Considerations of the Centrifugal Compressor Development" by C. Xu and R. S. Amano
[5] "Impellor diffuser interaction in centrifugal compressor" by yu kwong..
[6] "Effect of Area Ration on The Performance of The Centrifugal Compressor"
[7] Lawrence F. Schumann and David A. Clark
[8] "Design of a High Efficiency Compact Centrifugal Compressor for Rotorcraft Applications" by E.A. Lurie..
[9] "Computational Study on the Effect of Blade Cutback on the Performance of the Centrifugal Compressor" by Seralathan S
[10] "Guide-Vanes Upstream Impeller of Centrifugal Compressor" by Toussaint Michel…
[11] "Modeling for control of Centrifugal Compressors" By Oskar Leufvén
[12] "Performance Analysis of Refrigerant Centrifugal Compressor" By Sharad Chaudhary
[13] "Investigation of Performance Analysis of Compressor Cooling in Vapor Compression system" By Mr.V.M.Jamadar
[14] "Centrifugal Compressor Optimization Final Report" by Xavier Medina
[15] "Design Procedure of Centrifugal Compressors" By Adnan Hamza Zahed
[16] "Experimental and numerical analysis of a classical bleed slot system for a turbocharger compressor" by Bahram Nikpour
[17] "Study of the Flow in Centrifugal Compressor" by Cheng Xu
[18] "Design optimisation of an impeller with CFD and Meta-Model of optimal Prognosis (MoP)" by Florian Frese
[19] "Rotor Dynamics of Centrifugal Compressors in Rotating Stall" by Donald benty.
[20] "The Development of a High Speed Centrifugal Compressor Research Facility" by Ryan Fleming1, Fangyuan Lou1, and Nicole L. Key2 Purdue University, West Lafayette, IN, 47907