CFD Analysis Of Centrifugal Compresor For Acoustics Prediction

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Abstract*- When the centrifugal compressor operates at low mass flow rates (close to the unstable operating condition called surge), flow instabilities may develop and severe flow reversal may occur in the wheel passage. Under such conditions, noise generation has been reported resulting in a notable discomfort induced to the passengers in the cabin.*

The aim with this study is to predict the flow field associated with a centrifugal compressor and characterize the acoustic near-field generation and propagation under stable and off-design (near-surge) operating conditions. The Large Eddy Simulation (LES) approach is employed. The unsteady features in the flow field leading to acoustic noise generation are quantified by means Cfd analysis by designing a 3d blade profile in solidworks and simulation in the ansys. The Simulation method is performed inside the rotating impeller region for several stable and off-design (including surge and near-surge) operating conditions. The acoustic near-field data are presented in terms of noise directivity maps and sound pressure level spectra.

Keywords- Centrifugal compressor, Acoustics , CFD

I. INTRODUCTION

Liquid air and liquid nitrogen when used as cryogen energy carriers are different to other conventional heat storage media. The energy storage in a cryogen happens through decreasing the cryogen internal energy while increasing its exergy [1-2]. As for energy density, cryogens have shown higher energy density than other thermal energy storage media. Also with their low critical temperatures cryogen are efficient working fluids for recovering low grade heat, thus leading to higher overall cycle efficiencies [1].The stored cryogen liquid can then be expanded through a Turbine to convert the stored energy to work [3]. There are several types of process plants, like air separation units, helium and hydrogen liquefiers, low temperature refrigerators, and cryogen based Energy Storage (CES) systems [4]. Almost every system needs many components like turbine, heat exchanger, expansion Turbine, instrumentation, vacuum vessel etc. [5-6]. The expansion Turbine constitutes the most critical component of these cycles and its performance can significantly affects the overall cycle efficiency [7]. All process plants run under varying operating conditions, and for each plant setting, the Turbine inlet temperature, inlet pressure and flow rate will vary, leading to significant changes in Turbine performance. Therefore, in order to predict the overall performance of any plant under various operating conditions, it is necessary to calculate the Turbine inlet conditions corresponding the cycle operating conditions and to predict the performance of the Turbine under such inlet conditions. This requires the study of the performance of the Turbine at conditions away from its design point [7].

In this work the Mean line method and ANSYS CFD were used to develop a small scale axial Turbine and predict its performance at differentoperating conditions using Nitrogen as the working fluid. Operating parameters like inlet temperature, inlet pressure, mass flow rate and rotational speed were investigated.

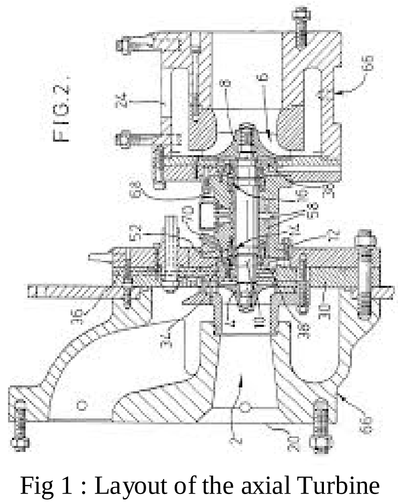


Fig 1 : Layout of the axial Turbine

Theory of operation

In an ideal Centrifugal compressor, gases undergo three thermodynamic processes: an isentropic compression, an isobaric (constant pressure) combustion and an isentropic expansion. Together, these make up the Brayton cycle.

Brayton cycle

In a practical Centrifugal compressor, mechanical energy is irreversibly transformed into heat when gases are compressed (in either a centrifugal or axial turbine), due to internal friction and turbulence. Passage through the combustion chamber, where heat is added and the specific volume of the gases increases, is accompanied by a slight loss in pressure. During expansion amidst the stator and rotor blades of the Turbine, irreversible energy transformation once again occurs.

If the device has been designed to power a shaft as with an industrial generator or a turboprop, the exit pressure will be as close to the entry pressure as possible. In practice it is necessary that some pressure remains at the outlet in order to fully expel the exhaust gases. In the case of a jet engine only enough pressure and energy is extracted from the flow to drive the turbine and other components. The remaining high pressure gases are accelerated to provide a jet that can, for example, be used to propel an aircraft.

As a general rule, the smaller the engine, the higher the rotation rate of the shaft(s) must be to maintain tip speed. Blade-tip speed determines the maximum pressure ratios that can be obtained by the Turbine and the turbine. This, in turn, limits the maximum power and efficiency that can be obtained by the engine. In order for tip speed to remain constant, if the diameter of a rotor is reduced by half, the rotational speed must double. For example, large jet engines operate around 10,000 rpm, while micro Turbines spin as fast as 500,000 rpm.[10]

Mechanically, Centrifugal compressors can be considerably less complex than internal combustion piston engines. Simple Turbines might have one moving part: the shaft/turbine/Turbine/alternative-rotor assembly (see image above), not counting the fuel system. However, the required precision manufacturing for components and temperature resistant alloys necessary for high efficiency often make the construction of a simple Turbine more complicated than piston engines.

More sophisticated Turbines (such as those found in modern jet engines) may have multiple shafts (spools), hundreds of Turbine blades, movable stator blades, and a vast system of complex piping, combustors and heat exchangers.
Thrust bearings and journal bearings are a critical part of design. Traditionally, they have been hydrodynamic oil bearings, or oil-cooled ball bearings. These bearings are being surpassed by foil bearings, which have been successfully used in micro Turbines and auxiliary power units.[citation needed]

Creep

A major challenge facing Turbine design is reducing the creep that is induced by the high temperatures. Because of the stresses of operation, Turbine materials become damaged through these mechanisms. As temperatures are increased in an effort to improve Turbine efficiency, creep becomes more significant. To limit creep, thermal coatings and superalloys with solid-solution strengthening and grain boundary strengthening are used in blade designs. Protective coatings are used in to reduce the thermal damage and to limit oxidation. These coatings are often stabilized zirconium dioxide-based ceramics. Using a thermal protective coating limits the temperature exposure of the nickel superalloy. This reduces the creep mechanisms experienced in the blade. Oxidation coatings limit efficiency losses caused by a buildup on the outside of the blades, which is especially important in the high-temperature environment.[11] The nickel-based blades are alloyed with aluminum and titanium to improve strength and creep resistance. The microstructure of these alloys is composed of different regions of composition. A uniform dispersion of the gamma-prime phase – a combination of nickel, aluminum, and titanium – promotes the strength and creep resistance of the blade due to the microstructure.[12] Refractory elements such as rhenium and ruthenium can be added to the alloy to improve creep strength. The addition of these elements reduces the diffusion of the gamma prime phase, thus preserving the fatigue resistance, strength, and creep resistance.[13]

MicroTurbines are touted to become widespread in distributed power and combined heat and power applications. They are one of the most promising technologies for powering hybrid electric vehicles. They range from hand held units producing less than a kilowatt, to commercial sized systems that produce tens or hundreds of kilowatts. Basic principles of microTurbine are based on combustion

Part of their claimed success is said to be due to advances in electronics, which allows unattended operation and interfacing with the commercial power grid. Electronic power switching technology eliminates the need for the generator to be synchronized with the power grid. This allows the generator to be integrated with the Turbine shaft, and to double as the starter motor.

MicroTurbine systems have many claimed advantages over reciprocating engine generators, such as higher power-to-weight ratio, low emissions and few, or just one, moving part. Advantages are that microTurbines may be designed with foil bearings and air-cooling operating without lubricating oil, coolants or other hazardous materials. Nevertheless, reciprocating engines overall are still cheaper when all factors are considered. MicroTurbines also have a further advantage of having the majority of the waste heat contained in the relatively high temperature exhaust making it simpler to capture, whereas the waste heat of reciprocating engines is split between its exhaust and cooling system.[22]

However, reciprocating engine generators are quicker to respond to changes in output power requirement and are usually slightly more efficient, although the efficiency of microTurbines is increasing. MicroTurbines also lose more efficiency at low power levels than reciprocating engines.

Reciprocating engines typically use simple motor oil (journal) bearings. Full-size Centrifugal compressors often use ball bearings. The 1000 °C temperatures and high speeds of microTurbines make oil lubrication and ball bearings impractical; they require air bearings or possibly magnetic bearings.[23]

When used in extended range electric vehicles the static efficiency drawback is irrelevant, since the Centrifugal compressor can be run at or near maximum power, driving an alternator to produce electricity either for the wheel motors, or for the batteries, as appropriate to speed and battery state. The batteries act as a "buffer" (energy storage) in delivering the required amount of power to the wheel motors, rendering throttle response of the Centrifugal compressor completely irrelevant.

There is, moreover, no need for a significant or variable-speed gearbox; turning an alternator at comparatively high speeds allows for a smaller and lighter alternator than would otherwise be the case. The superior power-to-weight ratio of the Centrifugal compressor and its fixed speed gearbox, allows for a much lighter prime mover than those in such hybrids as the Toyota Prius (which utilised a 1.8 litre petrol engine) or the Chevrolet Volt (which utilises a 1.4 litre petrol engine). This in turn allows a heavier weight of batteries to be carried, which allows for a longer electric-only range. Alternatively, the vehicle can use heavier types of batteries such as lead acid batteries (which are cheaper to buy) or safer types of batteries such as Lithium-Iron-Phosphate.

When Centrifugal compressors are used in extended-range electric vehicles, like those planned[when?] by Land-Rover/Range-Rover in conjunction with Bladon, or by Jaguar also in partnership with Bladon, the very poor throttling response (their high moment of rotational inertia) does not matter,[citation needed] because the Centrifugal compressor, which may be spinning at 100,000 rpm, is not directly, mechanically connected to the wheels. It was this poor throttling response that so bedevilled the 1960 Rover Centrifugal compressor-powered prototype motor car, which did not have the advantage of an intermediate electric drive train to provide sudden power spikes when demanded by the driver

Centrifugal compressors accept most commercial fuels, such as petrol, natural gas, propane, diesel, and kerosene as well as renewable fuels such as E85, biodiesel and biogas. However, when running on kerosene or diesel, starting sometimes requires the assistance of a more volatile product such as propane gas - although the new kero-start technology can allow even microTurbines fuelled on kerosene to start without propane.

MicroTurbine designs usually consist of a single stage radial turbine, a single stage radial Turbine and a recuperator. Recuperators are difficult to design and manufacture because they operate under high pressure and temperature differentials. Exhaust heat can be used for water heating, space heating, drying processes or absorption chillers, which create cold for air conditioning from heat energy instead of electric energy.

Typical microTurbine efficiencies are 25 to 35%. When in a combined heat and power cogeneration system, efficiencies of greater than 80% are commonly achieved.

MIT started its millimeter size Turbine engine project in the middle of the 1990s when Professor of Aeronautics and Astronautics Alan H. Epstein considered the possibility of creating a personal Turbine which will be able to meet all the demands of a modern person's electrical needs, just as a large Turbine can meet the electricity demands of a small city.[citation needed]

Problems have occurred with heat dissipation and high-speed bearings in these new microturbines. Moreover, their expected efficiency is a very low 5-6%. According to Professor Epstein, current commercial Li-ion rechargeable batteries deliver about 120-150 W·h/kg. MIT's millimetre size Turbine will deliver 500-700 W·h/kg in the near term, rising to 1200-1500 W·h/kg in the longer term.[24]

A similar microturbine built in Belgium has a rotor diameter of 20 mm and is expected to produce about 1000 W.[23]

II. LITERATURE REVIEW

A study on Design and Analysis of Axial Flow Turbine

J H Horlock (1958), presented the two dimensional or pitch line design analysis of turbine cascades. Thermodynamic stage design relations and fluid flow relations including free and forced vortex flows, radial equilibrium conditions etc. were presented based on several experimental test procedures. These correlations are very useful in determining the important stage performance measuring parameters like stage efficiency.

S Lieblien (1958), conducted loss and stall condition analysis in axial flow turbine cascades to determine various loss coefficients such as profile loss, skin friction loss, end wall loss etc. Quantitative measurements to determine the magnitude of losses were carried out.

S Lieblien (1960), carried out the analysis of low speed air turbine with conventional blades to determine the fluid flow characteristics in terms of incidence and deviation angles for minimum loss. Cascade theory of turbines and blade aerodynamic relations were utilized to bring insight into the behavior of fluid at different incidence and deviation angles.

B.Lakshmi narayana and J H Horlock (1963), developed the expression for flow model to determine the clearance between the tip of the blades and turbine casing wall during a blocked flow condition. The model predicts the decrease in stage efficiency due to tip clearance effect

B.Lakshminarayana (1970), presented a review on secondary flows and various loss sources that cause profile loss, skin friction loss, endturbine annulus region. These losses were estimated by conducting wind tunnel tests on turbines with different geometrical configurations.

C C Koch and L H Smith (Jr) (1976), determined various loss sources causing skin friction loss, end wall loss, profile loss etc., and their influence on the performance of axial flow turbine stage.

Tesch W.A, Moszee R.H et al (1976), applied stability and frequency response analysis techniques to provide a more economical approach to surge line and frequency response determination in blade rows of turbo machinery. The model was extended for turbines with inter stage cross flows.

Steinke R J (1976), presented an aerodynamic design of five stage core turbine with 9.271:1 pressure ratio and 29.17 kg/sec of mass flow rate. The first three stages in the design of core turbine were fabricated and tested experimentally. An optimal inlet guide vane set was determined to improve the adiabatic efficiency.

MC Kenzie AB (1980), formulated the Semi empirical relations and correlations for axial flow turbine blades, based on the tests conducted on a low speed axial flow turbine.

E.Macchi and A.perdichizzi (1981), presented a reliable method for estimation of efficiency of a Turbine stage. The Turbine stage performance was found to be a function of three main parameters, i.e.; the expansion ratio, specific speed and a dimension less parameter which accounts for actual Turbine dimensions.

C C Koch (1981), presented an engineering approach to the problem of predicting maximum pressure rise capability (or) predicting the maximum value of stall margin coefficient. A semi-empirical model was developed based on the tests conducted on a low speed axial flow turbine.

EM Greitzer and F K moore (1986), presented a theory based on rotating stall and surge phenomena in case of axial flow turbines was proposed. A theoretical compression model was presented in that work.

A Sehra, J Bettner et al (1992), applied the design techniques developed for aircraft turbines to the turbines used in low utility Centrifugal compressor. The objective was to develop an aero dynamic design with a level of stage efficiency, which is higher than that of the turbines used in large Centrifugal compressors.

I J Day (1993), analyzed the occurrence of stall phenomena including rotating stall and surge in case of axial flow turbines. He described the discovery and importance of short length scale disturbances in the stall inception process.

Mansoux, C.A , Gysling, D.L et al (1994), developed a nonlinear Moore-Greitzer rotating stall model suitable for control, analysis and design of axial flow turbine against rotating stall and surge phenomena. The nonlinear turbine characteristic obtained from the model was shown to be the primary determinant of stall inception transient behavior.

Adnan M. Abdel Fattah and Peter.C.Frith (1995), proposed an approximate procedure for the derivation of individual stage characteristics of a multi stage axial flow turbine. The stage characteristics were derived from the steady state overall performance map, geometry of turbine annulus and blade section profile data. The one dimensional simulation model developed by NASA-LEWIS research centre was used in predicting the overall performance of turbine by applying the stage stacking technique.

III. METHODOLOGY

Mesh

To analyze fluid flows, flow domains are split into smaller subdomains (made up of geometric primitives like hexahedra and tetrahedra in 3D and quadrilaterals and triangles in 2D). The governing equations are then discretized and solved inside each of these subdomains. The aim of the simulation was to analyse the fluid flow within the computational domain. Considering that the quality and resolution of the mesh have a great impact on the results, a fine hybrid tetrahedral element mesh of about 1.5 million cells weused. (Fig 7 to Fig 10 below

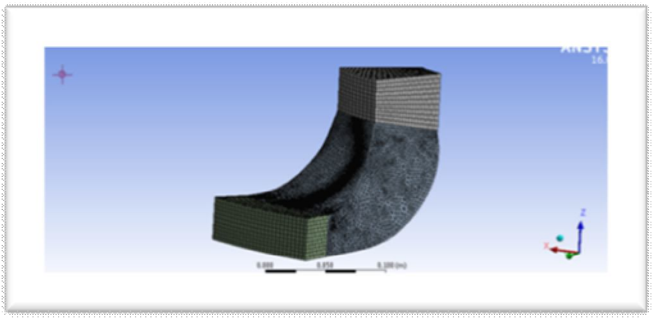


Figure 2 : Computational domain (Plan)

Solution and Post-processing

The process of solving a complex system is inherently difficult and requires high-end computing machines. For stability and convergence several hundred iterations were performed in ANSYS FLUENT 16.0 SOFTWARE. The post processing of results were done using ANSYS CFD POST and other tools like MS EXCEL.

IV. RESULTS

Results of the axial Centrifugal compressor

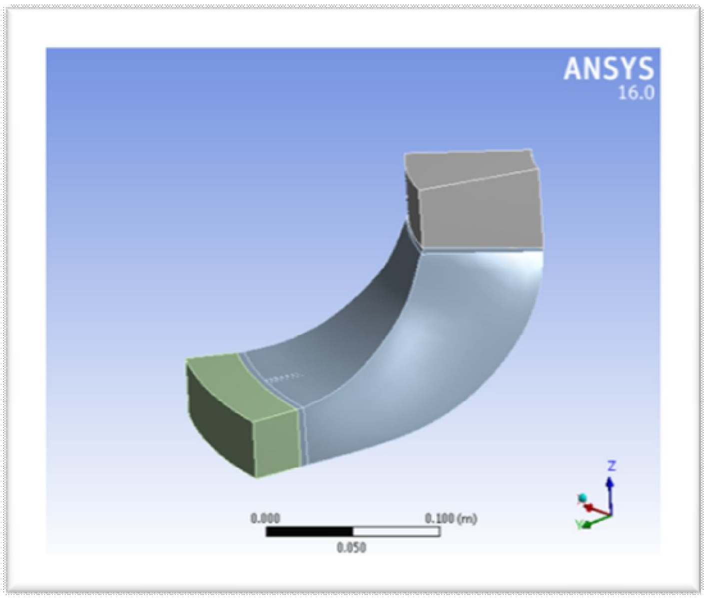
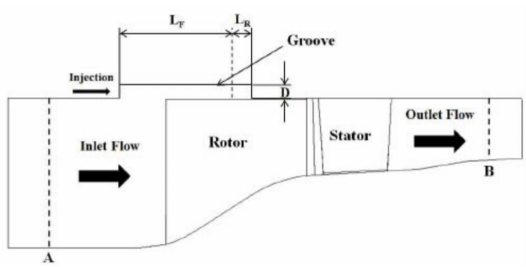


Figure 3: Meridonial region of an axial Centrifugal compressor



Stage Plots

The following plots show, for each stage, a meridional view of the geometry, blade-to-blade contour and vector views, and circumferentially averaged meridional views.

Stage 1 Plots

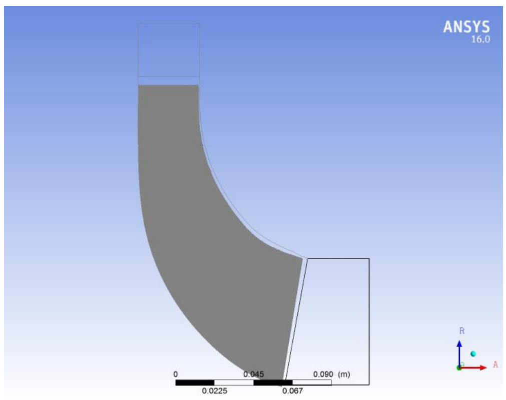


Figure 4: Meridonial geometry

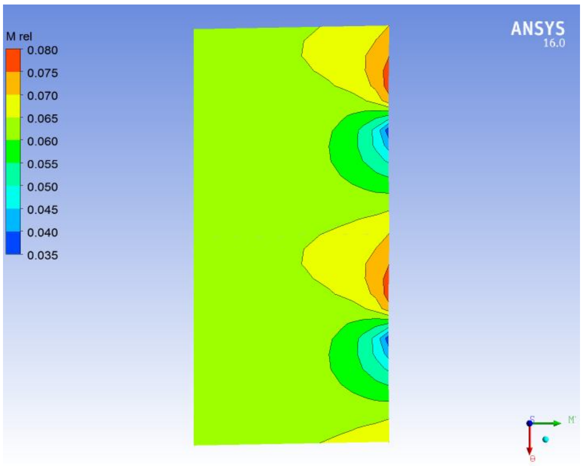


Figure 5: Stage 1 contours of M rel at 50% span

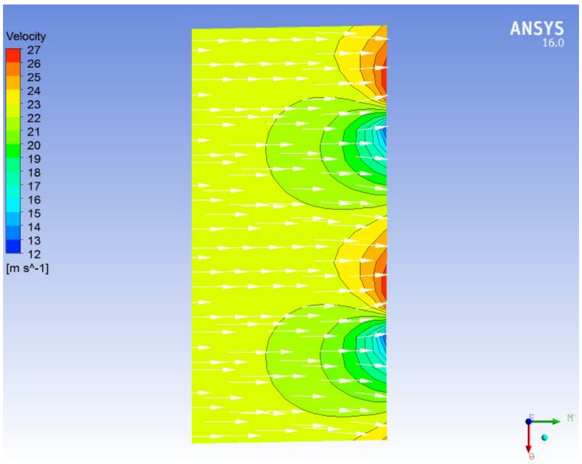


Figure 6: Stage 1 velocity vectors at 50% span

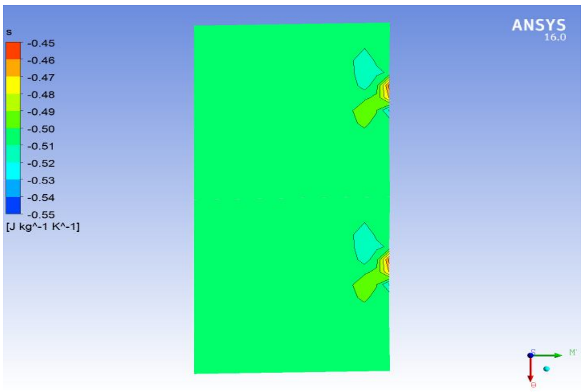


Figure 7: Stage 1 contours of s at 50% span

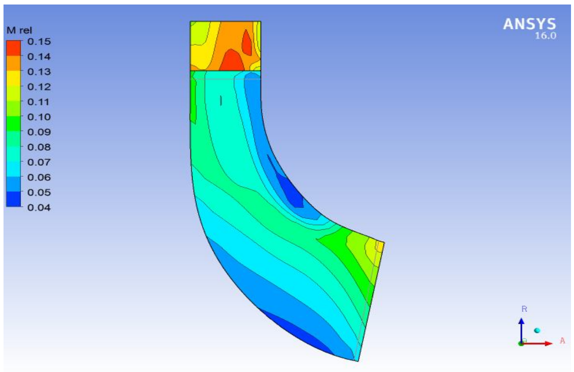


Figure 8: Stage 2 contours of circumferentially area-averaged M rel

Component Charts

The following charts show blade loading and spanwise-averaged quantities for each component.
The following charts show the blade loading for each component.

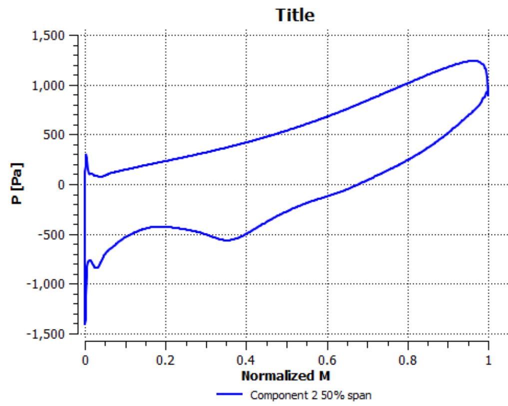


Chart 1. Component 2 blade loading chart

The following charts show circumferentially averaged quantities along hub-to-shroud lines located at the leading and trailing edges of the blade.

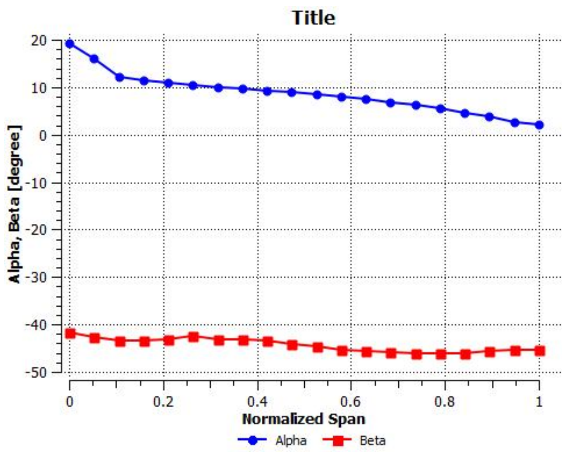


Chart 2. Component 2 chart showing circumferentially averaged flow angle at the LE

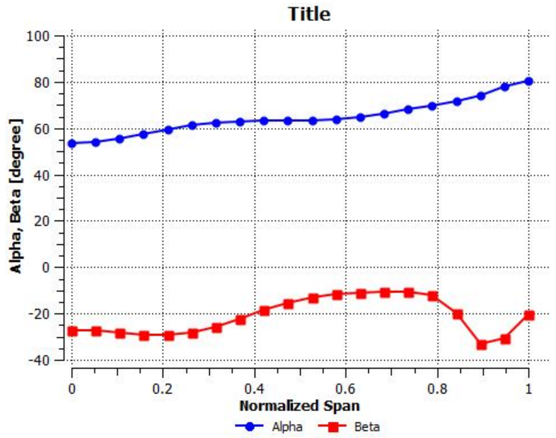


Chart 3. Component 2 chart showing circumferentially averaged relative Mach number at the LE

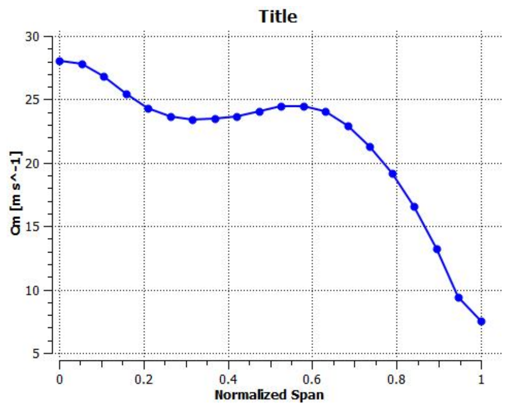


Chart 4. Component 2 chart showing circumferentially averaged flow angle at the TE

V. CONCLUSION

In this project we have done the Cfd simulation for Centrifugal gas Compressor and derived the flow patterens and pressure and velocity contours This type of simulation requires major computational Facilities to derive such pattern This flow pattern is very useful in managing the efficiency of the Compressor.

Centrifugal gas Compressors are relatively new in the market and are attracting wide attention due to their aired applications. Development of a sophisticated engineering product like micro Compressoris a continuous process. A lot of work is yet to bed one on the design aspects be fore the micro Compressor can be readied for market consumption. The design procedure has to take into various other parameters to make it suitable for practical applications .Also , manufacturing of such complex shapes of minute size is another ongoing research work. Further research in to the design and manufacture process would result in production of even better micro gasCompressors.

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