A NOVEL APPROACH FOR AXIAL FLOW TURBINE SUBJECTED TO STRESSES AND DEFORMATION

MANIKALA GOPALA KRISHNA[1], PAGADALA SIDDIRAJU[2], T.CHAKRAVARHI[3]

[1]Student, Department of Mechanical Engineering, SRI VANI EDUCATIONAL SOCIETY GROUP OF INSTITUTIONS, A.P., India.

[2]Assistant Professor, Department of Mechanical Engineering, SRI VANI EDUCATIONAL SOCIETY GROUP OF INSTITUTIONS, A.P., India.

[3] Assistant Professor & Head of the Department, Department of Mechanical Engineering, SRI VANI EDUCATIONAL SOCIETY GROUP OF INSTITUTIONS, A.P., India.

***Abstract*—** The axial-flow mechanical device compresses its operating fluid by initial fast the fluid and so dispersive it to get a pressure increase. This entire work is done in various steps like firstly modeling the turbine in SOLIDWORKS software and importing the geometry into the ANSYS software for process of analysis. Here five types of analysis are being done namely, static structural analysis, transient thermal analysis, structural to thermal coupled analysis, fatigue analysis and modal analysis. Axial flow compressor in which helps in passing a high compressor air and is which can be alternatethe air. In a pivotal stream compressor, air goes starting with one phase then onto the next, each stage rising the somewhat. The vitality level of air or gas coursing through it is expanded by the activity of the rotor sharp edges which apply a torque on the liquid which is provided by an electric engine or a steam or a gas turbine. In this postulation, a hub stream compressor is outlined and displayed in 3D demonstrating programming CATIA and Ansys 15.0 The present plan has 30 cutting edges, in this theory it is supplanted with 29 sharp edges and to discover the shear stresses, disfigurement of a compressor and a disappointment investigation through single edge. A solitary cutting edge investigation is improved the situation disappointment conditions. The present utilized material is Aluminum and magnesium combination. Static Structural investigation is done on the compressor models to confirm the quality of the compressor.

keywords: Axial Flowcompressor, Ansys, CATIA.

INTRODUCTION

Structural system of axial flow turbine can be easily analyzed using finite element analysis techniques. From these techniques the FEA is done in order to find out the stresses in the existing turbine for the given loads and boundary conditions using Finite Element Analysis Software ANSYS Workbench. The fluid is accelerated by a row of rotating airfoils (blades) referred to as the rotor, and so subtle in an exceedingly row of stationary blades (the stator). The diffusion within the mechanical device converts the rate increase gained within the rotor to a pressure increase. A mechanical device consists of many stages: 1) a mix of a rotor followed by a mechanical device make-up a stage in an exceedingly mechanical device; 2) a further row of stationary blades ar often used at the compressor recess and are referred to as recess Guide Vanes (IGV) to turn out that air enters the first-stage rotors at the required flow angle, these vanes also are pitch variable so are often adjusted to the variable flow necessities of the engine; and 3) additionally to the stators, another diffuser at the exit of the mechanical device consisting of another set of vanes more diffuses the fluid and controls its speed coming into the combustors and is commonly referred to as the Exit Guide Vanes (EGV). In AN axial flow mechanical device, air passes from one stage to subsequent, every stage raising the pressure slightly. By manufacturing nonaggressive will increase on the order of one.1:1 to 1.4:1, terribly high efficiencies are often obtained. the employment of multiple stages permits overall pressure will increase of up to 40:1 in some region applications and a pressure quantitative relation of 30:1 in some Industrial applications. like alternative kinds of rotating machinery, AN axial mechanical device are often delineate in an exceedingly cylindrical frame of reference. The z axis is on the axis of rotation that is on the running length of the mechanical device shaft, the radius r is measured outward from the shaft, and therefore the angle of rotation θ is that the angle turned by the blades . These frames of reference are used throughout this discussion of axialflow compressors

WORKING PRINCIPAL OF AXIAL FLOW COMPRESSOR

As the liquid enters and leaves in the hub heading, the radial part in the vitality condition does not become possibly the most important factor. Here the pressure is completely in light of diffusing activity of the entries. The diffusing activity in stator changes over supreme motor leader of the liquid into ascend in weight. The relative motor head in the vitality condition is a term that exists simply because of the pivot of the rotor. The rotor decreases the relative motor leader of the liquid and adds it to irrefutably the active leader of the liquid i.e., the effect of the rotor on the liquid particles expands its speed (total) and in this manner lessens the relative speed between the liquid and the rotor. So, the rotor expands the outright speed of the liquid and the stator changes over this into weight rise. Planning the rotor section with a diffusing ability can create a weight ascend notwithstanding its ordinary working. This produces more noteworthy weight rise per organize which constitutes a stator and a rotor together. This is the response rule in turbo machines. In the event that half of the weight ascend in a phase is acquired at the rotor segment, it is said to have a half response.

Hub stream compressors are normally utilized at applications with low differential weight (head) necessities and high stream rates. An average hub compressor comprises of a drum, to which cutting edges of particular geometry are connected.

Ordinary utilizations of enormous size pivotal compressors are those used to pack the air admission of gas turbines. These are commonly multistage pivotal compressors, comprising of a few phases. Each stage normally comprises of a rotor and a stator. A rotor is a set or pivoting cutting edges, whose part is to quicken the gas stream. A stator is an arrangement of stationary edges whose part is to change over the speed vitality picked up in the rotor to weight vitality, like the impeller – diffuser blend utilized as a part of the radial compressors.

AXIAL-FLOW JET ENGINES:

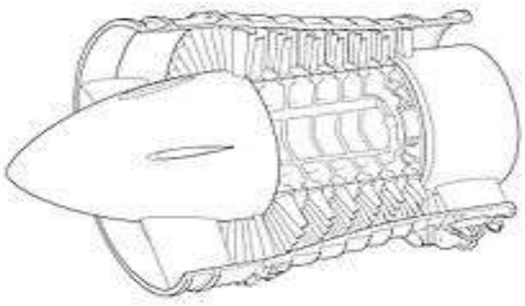

Figure: Low-pressure axial compressor scheme of the turbojet.

In the stream motor application, the compressor confronts a wide assortment of working conditions. On the ground at departure the bay weight is high, channel speed zero, and the compressor spun at an assortment of velocities as the power is connected. Once in flight the channel weight drops, however the gulf speed increments (because of the forward movement of the flying machine) to recoup some of this weight, and the compressor tends to keep running at a solitary speed for drawn out stretches of time.

There is just no "flawless" compressor for this extensive variety of working conditions. Settled geometry compressors, similar to those utilized on early stream motors, are restricted to an outline weight proportion of around 4 or 5:1. Likewise with any warmth motor, fuel productivity is firmly identified with the pressure proportion, so there is exceptionally solid money related need to enhance the compressor organizes past these sorts of proportions. Also the compressor may slow down if the delta conditions change suddenly, a typical issue on early motors. Now and again, if the slow down happens close to the front of the motor, the majority of the phases starting there on will quit compacting the air. In this circumstance the vitality required to run the compressor drops all of a sudden, and the staying hot air in the back of the motor enables the turbine to accelerate the entire motor significantly. This condition, known as surging, was a noteworthy issue on early motors and regularly prompted the turbine or compressor breaking and shedding sharp edges. For these reasons, hub compressors on present day fly motors are impressively more intricate than those on prior outlines.

Scope of the present work:

Numerous experimental and scientific programming methods are produced to outline and streamline the pivotal stream compressor arrange. In any case, the ideal outline of pivotal stream compressor arrange still remains a push territory of research for some originators. This is because of the nearness of a substantial number of clashing parameters which require the execution of effective calculations. So as to enhance the numerous clashing target works, the ordinary improvement procedures are not exactly reasonable. These methods are productive in deciding non-ruled arrangement fronts that are best as for all the goal capacities.

LITARATURE REVIEW

[1] J H Horlock (1958),

displayed the two dimensional or pitch line plan investigation of compressor falls. Thermodynamic stage plan relations and liquid stream relations including free and constrained vortex streams, spiral xxvii harmony conditions and so forth were exhibited in view of a few test systems. These connections are extremely helpful in deciding the critical stage execution measuring parameters like stage proficiency. S Lieblien (1958), led misfortune and slow down condition examination in hub stream compressor falls to decide different misfortune coefficients, for example, profile misfortune, skin contact misfortune, end divider misfortune and so on. Quantitative estimations to decide the size of misfortunes were done.

[2] S Lieblien (1960),

completed the investigation of low speed air compressor with customary cutting edges to decide the liquid stream attributes as far as rate and deviation plots for least misfortune. Course hypothesis of compressors and cutting edge streamlined relations were used to bring knowledge into the conduct of liquid at various occurrence and deviation edges. Lakshmi narayana and J H Horlock (1963), built up the articulation for stream model to decide the freedom between the tip of the sharp edges and compressor packaging divider amid a blocked stream condition. The model predicts the abatement in organize productivity because of tip leeway impact.

[3] B.Lakshminarayana (1970)

The exhibited an audit on optional streams and different misfortune sources that reason profile misfortune, skin erosion misfortune, end xxviii compressor annulus district. These misfortunes were assessed by directing breeze burrow tests on compressors with various geometrical setups. C Koch and L H Smith (Jr) (1976), decided different misfortune sources causing skin erosion misfortune, end divider misfortune, profile misfortune and so forth., and their impact on the execution of pivotal stream compressor organize. Tesch W.A, Moszee R.H et al (1976), connected strength and recurrence reaction investigation strategies to give a more temperate way to deal with surge line and recurrence reaction assurance in edge lines of turbo apparatus. The model was reached out for compressors with entomb organize cross streams. Steinke R J (1976), introduced a streamlined outline of five phase center compressor with 9.271:1 weight proportion and 29.17 kg/sec of mass stream rate. The initial three phases in the outline of center compressor were created and tried tentatively. An ideal gulf control vane set was resolved to enhance the adiabatic proficiency.

[4] MC Kenzie AB (1980)

The Semi observational relations and connections for pivotal stream compressor cutting edges, in view of the tests directed on a low speed hub stream compressor. execution was observed to be a component of three fundamental parameters, i.e.; the development proportion, particular speed and a measurement less parameter which represents real turbine measurements. C Koch (1981), introduced a building way to deal with the issue of anticipating greatest weight rise capacity (or) foreseeing the most extreme estimation of slow down edge coefficient. A semi-observational model was created in view of the tests directed on a low speed pivotal stream compressor.

DESIGN OF AXIAL FLOW COMPRESSOR USING CATIA

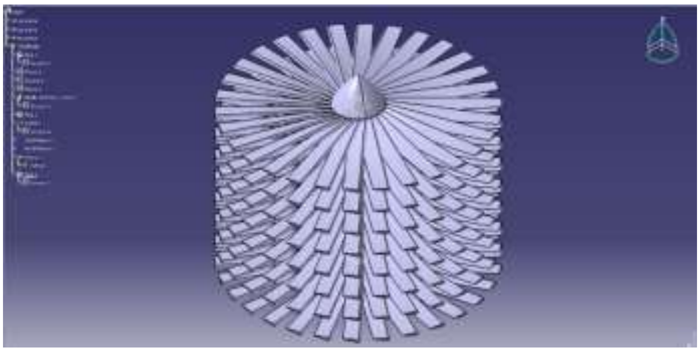

Figure 3.1 30 blade design axial flow compressor.

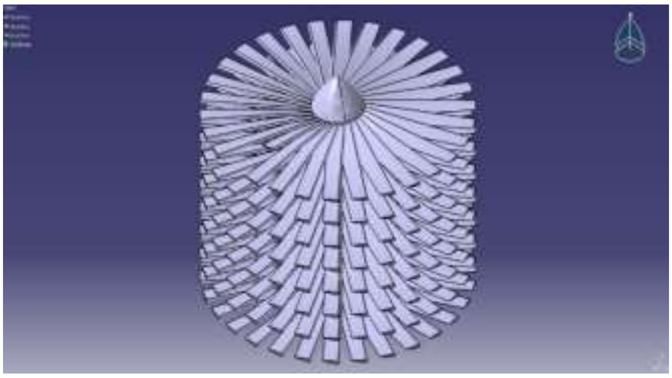

Figure 3.2 design of 29 blade axial compressor

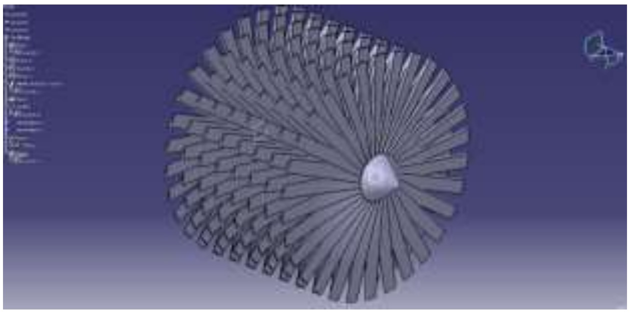

Figure 3.3 schematic view of axial flow compressor

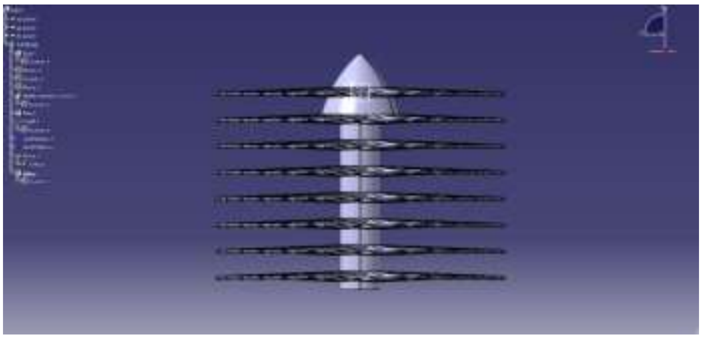

Figure 3.4 schematic view of design of blades

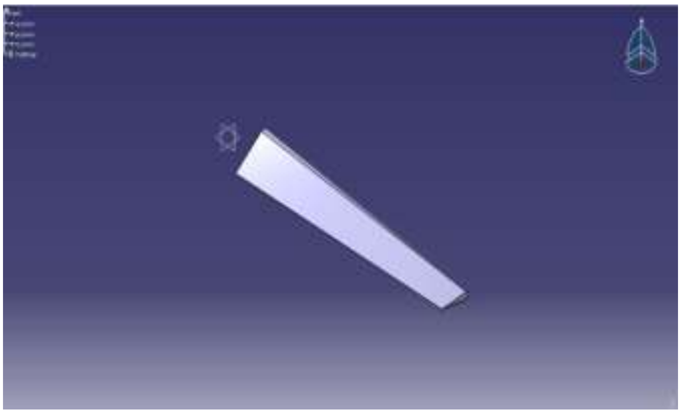

Figure 3.5 design of single blade of axial flow compressor

CONCLUSION

This paper presented The conclusions thus extracted from these analyses are the numerical values of stress concentrations, displacement, thermal stress, various modes of failure for different natural frequencies, and its life in cycles and safety factor. And it is found out that maximum stress is 194.61MPa, and in the thermal to structural coupled analysis the maximum stress is 217.65MPa. The other stresses outside the localized stress region are well within the yield strength value 1035Mpa, the life of the turbine chosen is 1x106 cycles. Hence the Axial Flow Turbine is safe.

REFERENCES

[1] J H Horlock (1958) Axial Flow Compressors, Butterworths Scientific Publications, London, ISSN 1650379 Pages 157-167

[2] S Lieblien (1960), Gostelow, J. P., Cascade Aerodynamics, PergamonPress),ISSN: 2348 – 8360 Page 85 ...

[3] B.Lakshminarayana (1970) Aerodynamics of Turbines and Compressors, Princeton University Press, Princeton, NJ,.),ISSN: 2348 – 8360 Page 85 ...

[4] MC Kenzie AB (1980) Axial-Flow Compressors, ASME Press, New York, ISSN'S (1-50 of 89) pages 300-305

[5] EM Greitzer and F K moore (1986), Axial Flow Fans and Compressors-Aerodynamic Design and Performance, Ashgate, England,), ISSN 1941-7020, Page 2 ..

[6] R.Schulze and D.K.Hennecke (1998) Aerodynamics Aircraft Engine Components, AIAA, New York, ISSN 2032-6653 , page 1265,

[7] R.Schulze and D.K.Hennecke (1998) Aircraft Engine Design, AIAA, New York,) ISSN 2353-6845 Page 1 ... 7

[8] Tomita JT and Barbosa JR (2004) A., Principles of Turbo-hardware in Air-Breathing Engines, Cambridge University Press,ISSN 1307-5179 pages 99-105

[9] P.V.Ramakrishna and M.Govardhan (2010) optimal design, Longman Scientific and Technical, Essex,ISSN 1819-6608 Page 1.

[10] S.S.Rao and R.S.Guptha (1980), "A Preliminary Analysis of the Magnitude of Shock Losses in Transonic Compressors," NASA Technical PaperNACA-RM-E57A30, 1957.

[11]. A Massado and A Satta (PART-I) (1990), "Counts of Two-and Three-Dimensional Transonic Cascade Flow Fields Using the NavierStokes Equations,: J. Eng. Gas Turbines and Power, Vol. 108, Jan., pp. 93-102,

[12] Jeffrey C. Lagarias, James A.reeds et al (1998), "Forecast of Compressor Cascade Performance Using a Navier-Stokes Technique," ASME Paper 88-GT-96, International Gas

Turbine Conference, Amsterdam, The Netherlands,

[13] BehroozFarshi, Reza Taghavi-Zenouz et al (2004). "Two-Dimensional Computations of Multi-organize Compressor Flows Using a Zonal Approach," AIAA Paper 89-2452, Monterey, Calif.,

[14] Etter D M, Hicks M J et al (1982), "Calculation of Internal Flows at High Reynolds Number by Numerical Solution of the Navier-Stokes Equations," Rech . Aerosp., No. 1986-6, pp. 27-44, 1986.

[15] S Rajeev and C S Krishna murthy (1992) "Exploratory and Computational Investigation of the Tip Clearance Flow in a Transonic Axial Compressor," ASME Paper 94-GT-365, June 1994