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Design and Optimization of Axial Flow Compressor

Shaik Abdul Salam[1], C Lakshmaiah[2] & M Venkata Ramudu[3]
[1]P.G. Scholar, [2]M.Tech Asst Professor, [3]Head Of The Department
Branch : Machine Design
[1,2,3]Geethanjali College Of Engineering & Technology
Email: [1]salamshaik126@gmail.com, [2]lakshmaiahcherukupalle@gmail.com

Abstract:

An axial flow compressor is one in which the flow enters the compressor an axial way (parallel with the hub of rotation), and ways out from the gas turbine, likewise in an axial course. The axial-flow compressor packs its working liquid by first quickening the liquid and after that diffusing it to acquire a weight increment. In an axial flow compressor, air goes starting with one phase then onto the next, each stage raising the weight marginally. The vitality dimension of air or gas flowing through it is expanded by the activity of the rotor cutting 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, an axial flow compressor is designed and displayed in 3d modeling programming genius/build. The present design has 30 cutting edges, in this theory it is replaced with 20 sharp edges and 12 edges. The present utilized material is chromium steel; it is replaced with titanium alloy and nickel alloy. Auxiliary examination is done on the compressor models to check the quality of the compressor. Cfd examination is done to check the flow of air..

Keywords : Axial Flow, Ansys, Compressor, CFD, Gas Turbine, PRO-E

INTRODUCTION
Axial Compressor

An axial compressor is a machine that can constantly pressurize gases. It is a pivoting, airfoil-based compressor in which the gas or working liquid essentially flows parallel to the hub of rotation. This varies from other turning compressors, for example, divergent compressors, axi-outward compressors and blended flow compressors where the liquid flow will incorporate an "outspread segment" through the compressor.

Transonic Axial Compressor : Transonic axial flow compressors are today generally utilized in airplane motors to get most extreme weight proportions per single-organize. High stage weight proportions are vital on the grounds that they make it conceivable to lessen the motor weight and estimate and, in this way, speculation and operational expenses.

Fig 1 Transonic lpc (left) and hpc (right) eurofighter typhoon engine EJ200

Three-dimensional shaped blades : The first section has demonstrated that a specific development in transonic compressors has been come to with respect to the general airfoil aerodesign. In any case, the flow field in a compressor isn't just affected by the two-dimensional airfoil geometry. The three-dimensional state of the edge is likewise of extraordinary significance, particularly in transonic compressor rotors where an advancement of stun structure and its obstruction with auxiliary flows is required. Numerous exploratory and numerical works can be found in the writing on the design and investigation of three-dimensional formed transonic bladings.

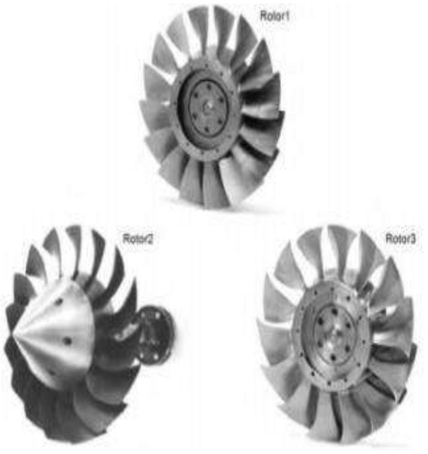
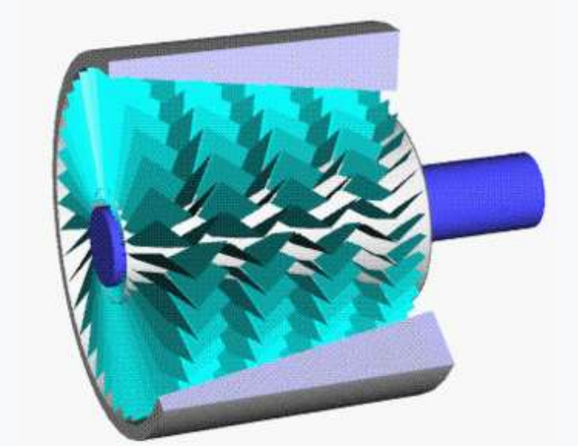


Fig 2 Transonic Compressor Test Rotors

AXIAL COMPRESSOR

From Wikipedia, the free encyclopedia
For more details on this topic, see Axial fan design.



An animated simulation of an axial compressor. The static blades are the stators.

An axial compressor is a compressor that can persistently pressurize gases. It is a pivoting, airfoil-based compressor in which the gas or working liquid mainly flows parallel to the hub of rotation, or axially. This varies from other pivoting compressors, for example, divergent compressors, axi-outward compressors and blended flow compressors where the liquid flow will incorporate an "outspread part" through the compressor.

The vitality dimension of the liquid increments as it flows through the compressor because of the activity of the rotor sharp edges which apply a torque on the liquid. The stationary cutting edges moderate the liquid, changing over the circumferential part of flow into weight. Compressors are commonly determined by an electric engine or a steam or a gas turbine.[1]

Axial flow compressors deliver a nonstop flow of compacted gas, and have the advantages of high effectiveness and expansive mass flow rate, especially in connection to their size and cross-segment. They do, in any case, require a few lines of airfoils to accomplish an extensive weight rise, making them mind boggling and costly with respect to different designs (e.g. radial compressors).

Axial compressors are essential to the design of expansive gas turbines, for example, fly motors, rapid ship motors, and little scale control stations. They are additionally utilized in modern applications, for example, substantial volume air partition plants, impact heater air, liquid synergist breaking air, and propane dehydrogenation. Because of superior, high unwavering quality and adaptable task amid the flight envelope, they are likewise utilized in aviation engines.

Typical application	Type of flow	Pressure ratio per stage	Efficiency per stage

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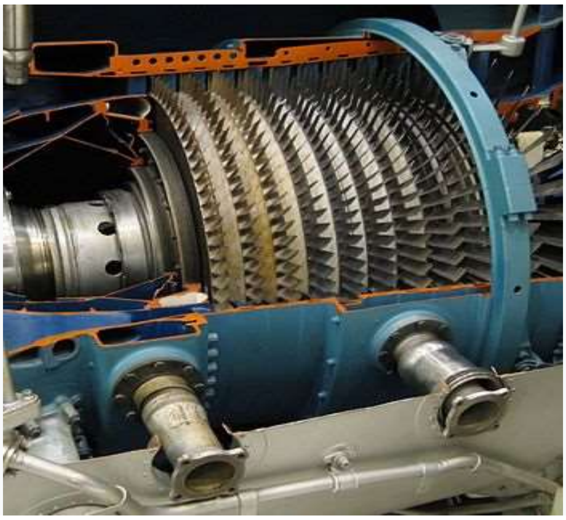
Industrial	Subsonic	1.05–1.2	88–92%
Aerospace	Transonic	1.15–1.6	80–85%
Research	Supersonic	1.8–2.2	75–85%

DESCRIPTION



The compressor in a Pratt and Whitney TF30 turbofan motor.

Axial compressors comprise of turning and stationary parts. A pole drives a focal drum, held by direction, which has various annular airfoil columns appended more often than not in sets, one turning and one stationary joined to a stationary cylindrical packaging. A couple of pivoting and stationary airfoils is known as a phase. The pivoting airfoils, otherwise called sharp edges or rotors, quicken the liquid. The stationary airfoils, otherwise called stators or vanes, convert the expanded rotational dynamic vitality into static weight through dissemination and divert the flow course of the liquid, setting it up for the rotor edges of the following stage.[3] The cross-sectional zone between rotor drum and packaging is lessened in the flow heading to keep up an ideal Mach number utilizing variable geometry as the liquid is compacted.

Working

As the liquid enters and leaves the axial way, the divergent part in the vitality condition does not become an integral factor. Here the pressure is completely founded on diffusing activity of the passages.The diffusing activity in stator changes over total active 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 rotation 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 builds its speed (supreme) and along these lines lessens the relative speed between the liquid and the rotor.

To put it plainly, the rotor expands the outright speed of the liquid and the stator changes over this into weight rise. Designing the rotor section with a diffusing capacity can create a weight ascend notwithstanding its ordinary working. This produces more noteworthy weight rise per arrange which establishes a stator and a rotor together. This is the response guideline in turbomachines. On the off chance that half of the weight ascend in a phase is acquired at the rotor area, it is said to have a half response.

Design

The airfoil profiles are improved and coordinated for explicit speeds and turning. In spite of the fact that compressors can be kept running at different conditions with various flows, speeds, or weight proportions, this can result in an effectiveness punishment or even a fractional or finish breakdown in flow (known as compressor slow down and weight flood individually). Subsequently, a reasonable limit on the quantity of stages, and the general weight proportion, originates from the communication of the diverse stages when required to work far from the design conditions. These "off-design" conditions can be moderated to a specific degree by giving some adaptability in the compressor. This is accomplished regularly using customizable stators or with valves that can drain liquid from the principle flow between stages (between stage drain). Present day fly motors utilize a progression of compressors, running at various velocities; to supply air at around 40:1 weight proportion for burning with adequate adaptability for all flight conditions

Assume the underlying working point D ( ) at some rpm N. On diminishing the flow-rate at same rpm along the trademark bend by incomplete shutting of the valve, the weight in the pipe builds which will be taken consideration by increment in info weight at the compressor. Further increment in weight till point P (flood point), compressor weight will increment. Further moving towards left keeping rpm consistent, weight in pipe will increment however compressor weight will diminish prompting back wind stream towards the compressor.

Because of this reverse, weight in pipe will diminish in light of the fact that this unequal weight condition can't remain for an extensive stretch of time. In spite of the fact that valve position is set for lower flow rate say point G yet compressor will work as indicated by ordinary stable task point say E, so way E-F-P-G-E will be pursued prompting breakdown of flow, subsequently weight in the compressor falls further to point H( ). This expansion and decline of weight in pipe will happen over and again in pipe and compressor following the cycle E-F-P-G-H-E otherwise called the flood cycle.

This wonder will cause vibrations in the entire machine and may prompt mechanical disappointment. That is the reason left part of the bend from the flood point is called precarious area and may make harm the machine. So the prescribed activity go is on the correct side of the flood line.

Slowing down

Slowing down is a critical wonder that influences the execution of the compressor. An examination is made of pivoting slow down in compressors of numerous stages, discovering conditions under which a flow contortion can happen which is enduring in a voyaging reference outline, despite the fact that upstream aggregate and downstream static weight are consistent. In the compressor, a weight rise hysteresis is assumed.[6] It is a circumstance of partition of wind stream at the air thwart cutting edges of the compressor. This wonder contingent on the cutting edge profile prompts lessened pressure and drop in motor power.

Positive slowing down

Flow partition happen on the suction side of the sharp edge.

Negative slowing down

Flow partition happen on the weight side of the sharp edge. Negative slow down is insignificant contrasted with the positive slow down on the grounds that flow partition is most drastically averse to happen on the weight side of the cutting edge.

In a multi-organize compressor, at the high weight stages, axial speed is little. Slowing down esteem diminishes with a little deviation from the design point causing slow down close to the center point and tip areas whose estimate increments with diminishing flow rates. They become bigger at low flow rate and influence the whole sharp edge tallness. Conveyance weight altogether drops with huge slowing down which can prompt flow inversion. The stage productivity drops with higher misfortunes.

Pivoting stalling[edit]

Non-consistency of wind stream in the rotor sharp edges may exasperate neighborhood wind stream in the compressor without disquieting it. The compressor keeps on working typically yet with diminished pressure. In this way, turning slow down declines the adequacy of the compressor.

In a rotor with sharp edges moving say towards right. Let a few sharp edges gets flow at higher occurrence, this edge will stop emphatically. It makes impediment in the entry between the sharp edge to one side and itself. Along these lines the left sharp edge will get the flow at higher occurrence and the cutting edge on its right side with diminished frequency. The left edge will encounter more slow down while the cutting edge to its correct will encounter lesser slow down. Towards the correct slowing down will diminish while it will increment towards its left. Development of the pivoting slow down can be watched relying on the picked reference outline.

Impacts

- This diminishes proficiency of the compressor
- Forced vibrations in the cutting edges because of section through slow down compartment.
- These constrained vibrations may coordinate with the characteristic recurrence of the sharp edges causing reverberation and thus disappointment of the edge.

Improvement

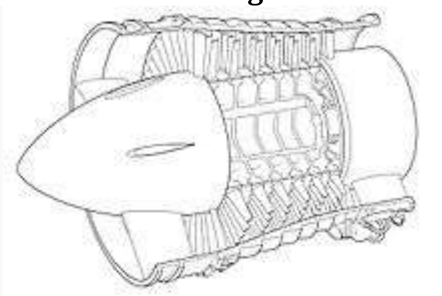
Early axial compressors offered poor proficiency, so poor that in the mid 1920s various papers guaranteed that a handy stream motor would be difficult to develop. Things changed after A. A. Griffith distributed a fundamental paper in 1926, taking note of that the explanation behind poor people execution was that current compressors utilized level sharp edges and were basically "flying slowed down". He demonstrated that the utilization of airfoils rather than the level sharp edges would expand effectiveness to the point where a down to earth fly motor was a genuine probability. He finished up the paper with a fundamental graph of such a motor, which incorporated a second turbine that was utilized to control a propeller.

Despite the fact that Griffith was notable because of his prior work on metal exhaustion and stress estimation, little work seems to have begun as an immediate consequence of his paper. The main clear exertion was a proving ground compressor worked by Hayne Constant, Griffith's associate at the Royal Aircraft Establishment. Other early stream endeavors, eminently those of Frank Whittle and Hans von Ohain, depended on the more strong and better comprehended outward compressor which was generally utilized in superchargers. Griffith had seen Whittle's work in 1929 and rejected it, taking note of a scientific blunder, and proceeding to guarantee that the frontal size of the motor would make it futile on a rapid air ship.

Genuine work on axial-flow motors began in the late 1930s, in a few endeavors that all began at about a similar time. In England, Hayne Constant achieved a concurrence with the steam turbine organization Metropolitan-Vickers (Metrovick) in 1937, beginning their turboprop exertion dependent on the Griffith design in 1938. In 1940, after the effective keep running of Whittle's diffusive flow design, their exertion was re-designed as an unadulterated fly, the Metrovick F.2. In Germany, von Ohain had created a few working diffusive motors, some of which had flown including the world's first stream airplane (He 178), however improvement endeavors had proceeded onward to Junkers (Jumo 004) and (BMW 003), which utilized axial-flow designs on the planet's first fly contender (Messerschmitt Me 262) and fly aircraft (Arado Ar 234). In the United States, both Lockheed and General Electric were granted contracts in 1941 to create axial-flow motors, the previous an unadulterated stream, the last a turboprop. Northrop additionally began their own task to build up a turboprop, which the US Navy in the long run contracted in 1943. Westinghouse additionally entered the race in 1942, their undertaking ended up being the main effective one of the US endeavors, later turning into the J30.

As Griffith had initially noted in 1929, the vast frontal size of the radial compressor made it have higher drag than the smaller axial-flow type. Moreover the axial-flow design could enhance its pressure proportion just by including extra stages and making the motor somewhat more. In the radiating flow design the compressor itself must be bigger in distance across, which was considerably more hard to "fit" legitimately on the air ship. Then again, divergent flow designs stayed substantially less intricate (the significant reason they "won" in the race to flying precedents) and in this manner have a job in spots where estimate and streamlining are not all that critical. Hence they remain a noteworthy answer for helicopter motors, where the compressor lies level and can be worked to any required size without irritating the streamlining to any extraordinary degree.

Axial Flow Jet Engines



Low-pressure axial compressor scheme of the Olympus BOl.1 turbojet.

In the jet engine application, the compressor faces a wide variety of operating conditions. On the ground at takeoff the inlet pressure is high, inlet speed zero, and the compressor spun at a variety of speeds as the power is applied. Once in flight the inlet pressure drops, but the inlet speed increases (due to the forward motion of the aircraft) to recover some of this pressure, and the compressor tends to run at a single speed for long periods of time.

There is simply no "perfect" compressor for this wide range of operating conditions. Fixed geometry compressors, like those used on early jet engines, are limited to a design pressure ratio of about 4 or 5:1. As with any heat engine, fuel efficiency is strongly related to the compression ratio, so there is very strong financial need to improve the compressor stages beyond these sorts of ratios.

Additionally the compressor may stall if the inlet conditions change abruptly, a common problem on early engines. In some cases, if the stall occurs near the front of the engine, all of the stages from that point on will stop compressing the air. In this situation the energy required to run the compressor drops suddenly, and the remaining hot air in the rear of the engine allows the turbine to speed up[citation needed] the whole engine dramatically. This condition, known as surging, was a major problem on early engines and often led to the turbine or compressor breaking and shedding blades.

For all of these reasons, axial compressors on modern jet engines are considerably more complex than those on earlier designs.

1.1.1. Spools

All compressors have an optimum point relating rotational speed and pressure, with higher compressions requiring higher speeds. Early engines were designed for simplicity, and used a single large compressor spinning at a single speed. Later designs added a second turbine and divided the compressor into low-pressure and high-pressure sections, the latter spinning faster. This *two-spool* design, pioneered on the Bristol Olympus, resulted in increased efficiency. Further increases in efficiency may be realised by adding a third spool, but in practice the added complexity increases maintenance costs to the point of negating any economic benefit. That said, there are several three-spool engines in use, perhaps the most famous being the Rolls-Royce RB211, used on a wide variety of commercial aircraft.

1.1.2. Bleed air, variable stators

See also: Bleed air

As an aircraft changes speed or altitude, the pressure of the air at the inlet to the compressor will vary. In order to "tune" the compressor for these changing conditions, designs starting in the 1950s would "bleed" air out of the middle of the compressor in order to avoid trying to compress too much air in the final stages. This was also used to help start the engine, allowing it to be spun up without compressing much air by bleeding off as much as possible. Bleed systems were already commonly used anyway, to provide airflow into the turbine stage where it was used to cool the turbine blades, as well as provide pressurized air for the air conditioning systems inside the aircraft.

A more advanced design, the *variable stator*, used blades that can be individually rotated around their axis, as opposed to the power axis of the engine. For startup they are rotated to "closed", reducing compression, and then are rotated back into the airflow as the external conditions require. The General Electric J79 was the first major example of a variable stator design, and today it is a common feature of most military engines.

Closing the variable stators progressively, as compressor speed falls, reduces the slope of the surge (or stall) line on the operating characteristic (or map), improving the surge margin of the installed unit. By incorporating variable stators in the first five stages, General Electric Aircraft Engines has developed a ten-stage axial compressor capable of operating at a 23:1 design pressure ratio.

1.2. Design notes

1.2.1. Energy exchange between rotor and fluid

The relative motion of the blades to the fluid adds velocity or pressure or both to the fluid as it passes through the rotor. The fluid velocity is increased through the rotor, and the stator converts kinetic energy to pressure energy. Some diffusion also occurs in the rotor in most practical designs.

The increase in velocity of the fluid is primarily in the tangential direction (swirl) and the stator removes this angular momentum.

The pressure rise results in a stagnation temperature rise. For a given geometry the temperature rise depends on the square of the tangential Mach number of the rotor row. Current turbofan engines have fans that operate at Mach 1.7 or more, and require significant containment and noise suppression structures to reduce blade loss damage and noise.

1.2.2. Compressor maps

A map shows the performance of a compressor and allows determination of optimal operating conditions. It shows the mass flow along the horizontal axis, typically as a percentage of the design mass flow rate, or in actual units. The pressure rise is indicated on the vertical axis as a ratio between inlet and exit stagnation pressures.

A surge or stall line identifies the boundary to the left of which the compressor performance rapidly degrades and identifies the maximum pressure ratio that can be achieved for a given mass flow. Contours of efficiency are drawn as well as performance lines for operation at particular rotational speeds.

1.2.3. Compression stability

Operating efficiency is highest close to the stall line. If the downstream pressure is increased beyond the maximum possible the compressor will stall and become unstable.

Typically the instability will be at the Helmholtz frequency of the system, taking the downstream plenum into account.

1.2.4. Centrifugal impeller

The key component that makes a compressor centrifugal is the centrifugal impeller, Figure 0.1, which contains a rotating set of vanes (or blades) that gradually raises the energy of the working gas. This is identical to an axial compressor with the exception that the gases can reach higher velocities and energy levels through the impeller's increasing radius. In many modern high-efficiency centrifugal compressors the gas exiting the impeller is traveling near the speed of sound.

Impellers are designed in many configurations including "open" (visible blades), "covered or shrouded", "with splitters" (every other inducer removed) and "w/o splitters" (all full blades). Both Figures 0.1 and 3.1 show open impellers with splitters. Most modern high efficiency impellers use "backsweep" in the blade shape.[6][17][18]

Euler’s pump and turbine equation plays an important role in understanding impeller performance.

1.2.5. Diffuser

The next key component to the simple centrifugal compressor is the diffuser.[7][8][18] Downstream of the impeller in the flow path, it is the diffuser's responsibility to convert the kinetic energy (high velocity) of the gas into pressure by gradually slowing (diffusing) the gas velocity. Diffusers can be vaneless, vaned or an alternating combination. High efficiency vaned diffusers are also designed over a wide range of solidities from less than 1 to over 4. Hybrid versions of vaned diffusers include: wedge, channel, and pipe diffusers. There are turbocharger applications that benefit by incorporating no diffuser.

Bernoulli's fluid dynamic principle plays an important role in understanding diffuser performance.

1.2.6. Collector

The collector of a centrifugal compressor can take many shapes and forms.[7][18] When the diffuser discharges into a large empty chamber, the collector may be termed a *Plenum*. When the diffuser discharges into a device that looks somewhat like a snail shell, bull's horn or a French horn, the collector is likely to be termed a *volute* or *scroll*. As the name implies, a collector’s purpose is to gather the flow from the diffuser discharge annulus and

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deliver this flow to a downstream pipe. Either the collector or the pipe may also contain valves and instrumentation to control the compressor.

Applications

Below, is a partial list of centrifugal compressor applications each with a brief description of some of the general characteristics possessed by those compressors. To start this list two of the most well-known centrifugal compressor applications are listed; gas turbines and turbochargers.[5]

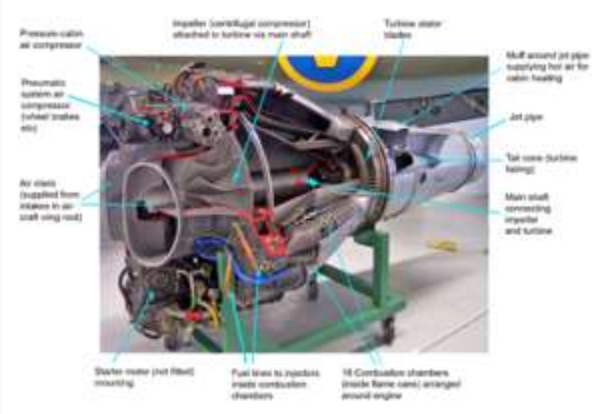

Figure 4.1 – Jet engine cutaway showing the centrifugal compressor and other parts.

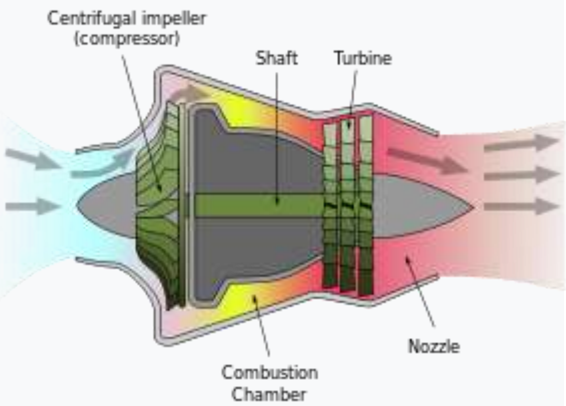

Figure 4.2 – Jet engine cross section showing the centrifugal compressor and other parts.

- In gas turbines and auxiliary power units.

In their simple form, modern gas turbines operate on the Brayton cycle. (ref Figure 5.1) Either or both axial and centrifugal compressors are used to provide compression. The types of gas turbines that most often include centrifugal compressors include turboshaft, turboprop, auxiliary power units, and micro-turbines. The industry standards applied to all of the centrifugal compressors used in aircraft applications are set by the FAA and the military to maximize both safety and durability under severe conditions.

Centrifugal impellers used in gas turbines are commonly made from titanium alloy forgings. Their flow-path blades are commonly flank milled or point milled on 5-axis milling machines. When tolerances and clearances are the tightest, these designs are completed as hot operational geometry and deflected back into the cold geometry as required for manufacturing. This need arises from the impeller's deflections experienced from start-up to full speed/full temperature which can be 100 times larger than the expected hot running clearance of the impeller.

- In automotive engine and diesel engine turbochargers and superchargers.[20] Ref. Figure 1.1

Centrifugal compressors used in conjunction with reciprocating internal combustion engines are known as turbochargers if driven by the engine's exhaust gas and turbo-superchargers if mechanically driven by the engine. Standards set by the industry for turbochargers may have been established by SAE.[21] Ideal gas properties often work well for the design, test and analysis of turbocharger centrifugal compressor performance.

- In pipeline compressors of natural gas to move the gas from the production site to the consumer.[22]

  Centrifugal compressors for such uses may be one- or multi-stage and driven by large gas turbines. Standards set by the industry (ANSI/API, ASME) result in large thick casings to maximize safety. The impellers are often if not always of the covered style which makes them look much like pump impellers. This type of compressor is also often termed an *API-style*. The power needed to drive these compressors is most often in the thousands of horsepower (HP). Use of real gas properties is needed to properly design, test and analyze the performance of natural gas pipeline centrifugal compressors.

- In oil refineries, natural gas processing, petrochemical and chemical plants.[22]

  Centrifugal compressors for such uses are often one-shaft multi-stage and driven by large steam or gas turbines. Their casings are often termed *horizontally split* or *barrel*. Standards set by the industry (ANSI/API, ASME) for these compressors result in large thick casings to maximize safety.

  The impellers are often if not always of the covered style which makes them look much like pump impellers. This type of compressor is also often termed *API-style*. The power needed to drive these compressors is most often in the thousands of HP. Use of real gas properties is needed to properly design, test and analyze their performance.

- Air-conditioning and refrigeration and HVAC: Centrifugal compressors quite often supply the compression in water chillers cycles.[23]

  Because of the wide variety of vapor compression cycles (thermodynamic cycle, thermodynamics) and the wide variety of workings gases (refrigerants), centrifugal compressors are used in a wide range of sizes and configurations. Use of real gas properties is needed to properly design, test and analyze the performance of these machines. Standards set by the industry for these compressors include ASHRAE, ASME & API.

- In industry and manufacturing to supply compressed air for all types of pneumatic tools.

  Centrifugal compressors for such uses are often multistage and driven by electric motors. Inter-cooling is often needed between stages to control air temperature. Note that the road repair crew and the local automobile repair garage find screw compressors better adapt to their needs. Standards set by the industry for these compressors include ASME and government regulations that emphasize safety. Ideal gas relationships are often used to properly design, test and analyze the performance of these machines. Carrier's equation is often used to deal with humidity.

- In air separation plants to manufacture purified end product gases.

  Centrifugal compressors for such uses are often multistage using inter-cooling to control air temperature. Standards set by the industry for these compressors

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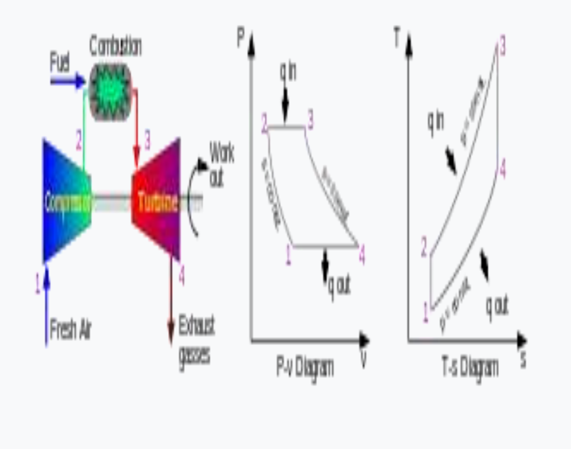
include ASME and government regulations that emphasize safety. Ideal gas relationships are often used to properly design, test and analyze the performance of these machines when the working gas is air or nitrogen. Other gases require real gas properties.

In oil field re-injection of high pressure natural gas to improve oil recovery.

Centrifugal compressors for such uses are often one-shaft multi-stage and driven by gas turbines. With discharge pressures approaching 700 bar, casing are of the barrel style. Standards set by the industry (API, ASME) for these compressors result in large thick casings to maximize safety. The impellers are often if not always of the covered style which makes them look much like pump impellers. This type of compressor is also often termed *API-style*. Use of real gas properties is needed to properly design, test and analyze their performance.

Performance

Below Figure – Brayton cycle. Illustration of the Brayton cycle as applied to a gas turbine.



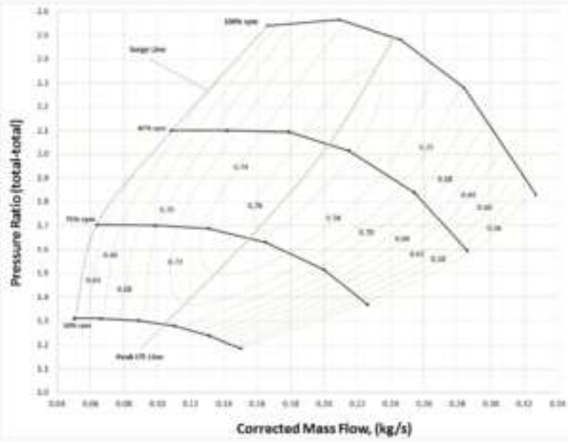


Figure 5.2 – Example centrifugal compressor performance map.

While illustrating a gas turbine's Brayton cycle,[12] Figure 5.1 includes example plots of pressure-specific volume and temperature-entropy. These types of plots are fundamental to understanding centrifugal compressor performance at one operating point. Studying these two plots further we see that the pressure rises between the compressor inlet (station 1) and compressor exit (station 2).

At the same time, it is easy to see that the specific volume decreases or similarly the density increases. Studying the temperature-entropy plot we see the temperature increase with increasing entropy (loss). If we assume dry air, and ideal gas equation of state and an isentropic process, we have enough information to define the pressure ratio and efficiency for this one point. Unfortunately, we are missing several other key pieces of information if we wish to apply the centrifugal compressor to another application.

Figure 5.2, a centrifugal compressor performance map (either test or estimated), shows flow, pressure ratio for each of 4 speed-lines (total of 23 data points). Also included are constant efficiency contours. Centrifugal compressor performance presented in this form provides enough information to match the hardware represented by the map to a simple set of end-user requirements.

Compared to estimating performance which is very cost effective (thus useful in design), testing, while costly, is still the most precise method.[9] Further, testing centrifugal compressor performance is very complex. Professional societies such as ASME (i.e. PTC–10, Fluid Meters Handbook, PTC-19.x),[25] ASHRAE (ASHRAE Handbook) and API (ANSI/API 617-2002, 672-2007)[22][24] have established standards for detailed experimental methods and analysis of test results. Despite this complexity, a few basic concepts in performance can be presented by examining an example test performance map.

1.2.7. Performance maps

Pressure ratio and flow are the main parameters[12][22][24][25] needed to match the Figure 5.2 performance map to a simple compressor application. In this case, it can be assumed that the inlet temperature is sea-level standard. Making this assumption in a real case would be a significant error. A detailed inspection of Figure 5.2 shows:

- Corrected mass flow: 0.04 – 0.34 kg/s
- Total pressure ratio, inlet to discharge ($PR_{t-t} = P_{t,discharge}/P_{t,inlet}$): 1.0 – 2.6

As is standard practice, Figure 5.2 has a horizontal axis labeled with a flow parameter. While flow measurements use a wide variety unit specifications, all fit one of 2 categories:

Mass flow per unit time

Mass flows, such as kg/s, are the easiest to use in practice as there is little room for confusion. Questions remaining would involve inlet or outlet (which might involve leakage from the compressor or moisture condensation).

For atmospheric air, the mass flow may be wet or dry (including or excluding humidity). Often, the mass flow specification will be presented on an equivalent Mach number basis. It is standard in these cases that the equivalent temperature, equivalent pressure and gas is specified explicitly or implied at a standard condition.

Volume flow per unit time

In contrast, all volume flow specifications require the additional specification of density. Bernoulli's fluid dynamic principal is of great value in understanding this problem. Confusion arises through either inaccuracies or misuse of pressure, temperature and gas constants.

Also as is standard practice, Figure 5.2 has a vertical axis labeled with a pressure parameter. The variety of pressure measurement units is also vast. In this case, they all fit one of three categories:

- *The delta increase or rise from inlet to exit* (Manometer style)

- *The measured discharge pressure* (force)
- *The force ratio* (ratio, exit/inlet)

Other features common to performance maps are:

Constant speed lines

The two most common methods used for testing centrifugal compressors is to test along lines of constant shaft speed or along lines of constant throttle. If the shaft speed is held constant, test points are taken along a constant speed line by changing throttle positions. In contrast, if a throttle valve is held constant, test points are established by changing speed (common gas turbine practice). The map shown in Figure 5.2 illustrate the most common method; lines of constant speed. In this case we see data points connected via straight lines at speeds of 50%, 71%, 87%, and 100% RPM. The first three speed lines have 6 points each while the highest speed line has five.

Constant efficiency islands

The next feature to be discussed is the oval shaped curves representing islands of constant efficiency. In this figure we see 11 contours ranging from 56% efficiency (decimal 0.56) to 76% efficiency (decimal 0.76). General standard practice is to interpret these efficiencies as isentropic rather than polytropic.

The inclusion of efficiency islands effectively generates a 3-dimensional topology to this 2-dimensional map. With inlet density specified, it provides a further ability to calculate aerodynamic power. Lines of constant power could just as easily be substituted.

Design or guarantee point(s)

Regarding gas turbine operation and performance, there may be a series of guaranteed points established for the gas turbine's centrifugal compressor. These requirements are of secondary importance to the overall gas turbine performance as a whole. For this reason it is only necessary to summarize that in the ideal case, the lowest specific fuel consumption would occur when the centrifugal compressors peak efficiency curve coincides with the gas turbine's required operation line.

In contrast to gas turbines, most other applications (including industrial) need to meet a less stringent set of performance requirements. Historically, centrifugal compressors applied to industrial applications were needed to achieve performance at a specific flow and pressure. Modern industrial compressors are often needed to achieve specific performance goals across a range of flows and pressures; thus taking a significant step toward the sophistication seen in gas turbine applications.

If the compressor represented by Figure 5.2 is used in a simple application, any point (pressure and flow) within the 76% efficiency would provide very acceptable performance. An "End User" would be very happy with the performance requirements of 2.0 pressure ratio at 0.21 kg/s.

Surge

Surge - is the point at which the compressor cannot add enough energy to overcome the system resistance or backpressure.[26] This causes a rapid flow reversal (i.e., surge). As a result, high vibration, temperature increases, and rapid changes in axial thrust can occur. These occurrences can damage the rotor seals, rotor bearings, the compressor driver and cycle operation. Most turbomachines are designed to easily withstand occasional surging. However, if the machine is forced to surge repeatedly for a long period of time, or if it is poorly designed, repeated surges can result in a catastrophic failure. Of particular interest, is that while turbomachines may be very

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durable, the cycles/processes that they are used within can be far less robust.

Surge line

The surge-line shown in Figure 5.2 is the curve that passes through the lowest flow points of each of the four speed lines. As a test map, these points would be the lowest flow points possible to record a stable reading within the test facility/rig. In many industrial applications it may be necessary to increase the stall line due to the system backpressure. For example, at 100% RPM stalling flow might increase from approximately 0.170 kg/s to 0.215 kg/s because of the positive slope of the pressure ratio curve.

As stated earlier, the reason for this is that the high-speed line in Figure 5.2 exhibits a stalling characteristic or positive slope within that range of flows. When placed in a different system those lower flows might not be achievable because of interaction with that system. System resistance or adverse pressure is proven mathematically to be the critical contributor to compressor surge.

THEORETICAL CALCULATIONS

Pressure = 0.904N/mm^2 ;

Temperature = 288K

Absolute Velocity

$$C_1 = \frac{C_{a_1}}{\cos \alpha_1} = \frac{150}{\cos(12)} = 177.75 \text{m/s}$$

C_{a_1}= Constant axial velocity

α_1 = Radius between blade to blade

MODELING IN CATIA

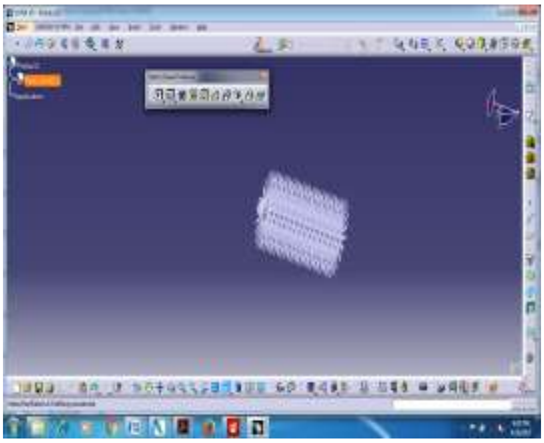

FIG 3: 30 BLADES

Fig 4:20 blades

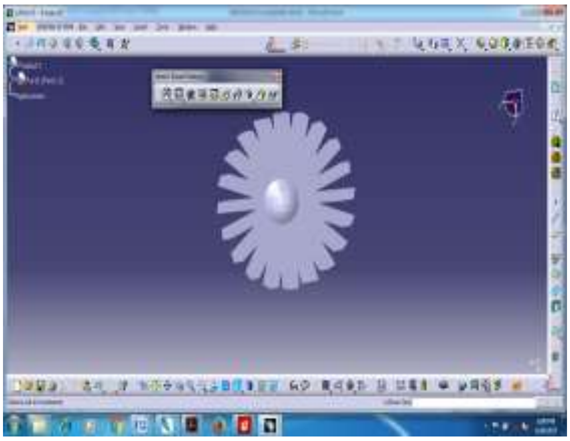

Fig 5: 12 blades

RESULTS & DISCUSSION

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Analysis of Compressor 30 Blades
Nickel Alloy

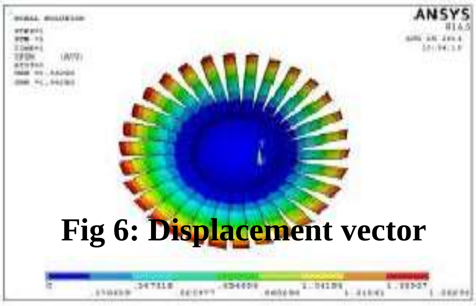

Fig 6: Displacement vector

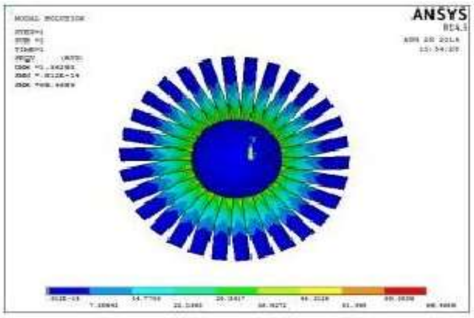

Fig 7: Stress vonmises vector

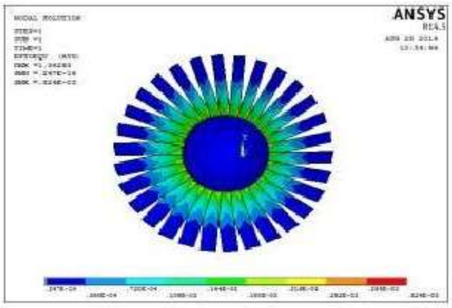

Fig 8: Strain Vonmises Vector

Titanium

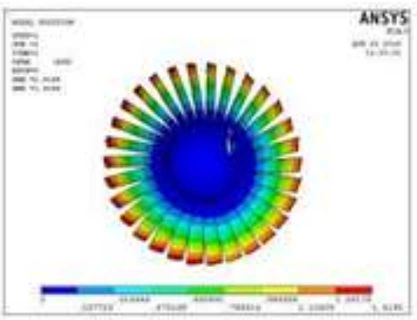

Fig 9: Displacement Vector Vector

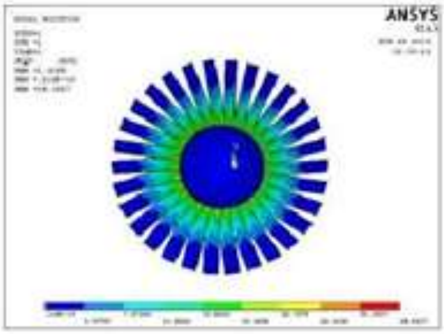

Fig 10: Stress Vonmises Vector

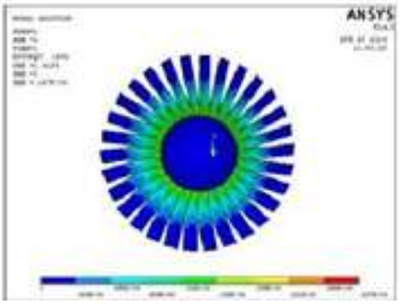

Fig 11: Strain vonmises

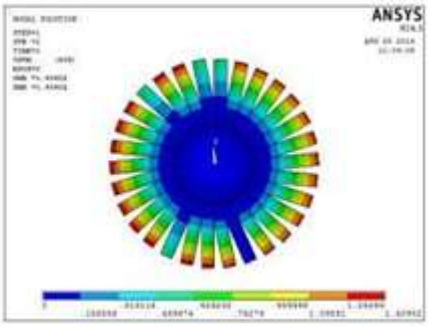

Fig 12: Displacement Vector Vector

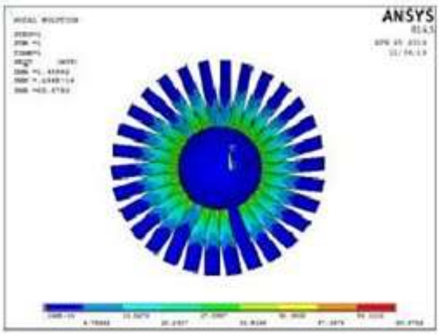

Fig 13: Stress Vonmises Vector

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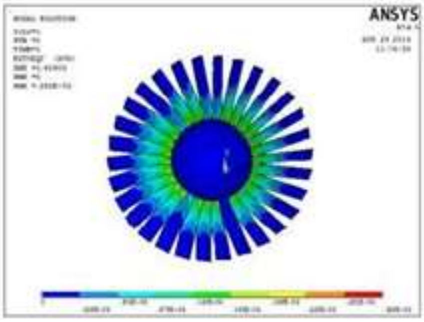

Fig 14: Strain Vonmises

Analysis of Compressor 20 blades Nickel alloy

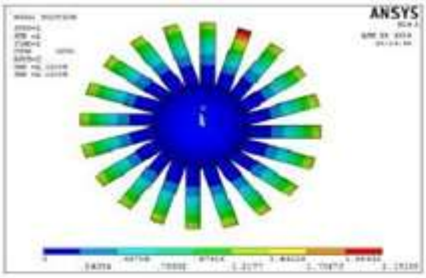

Fig 15: Displacement Vector Vector

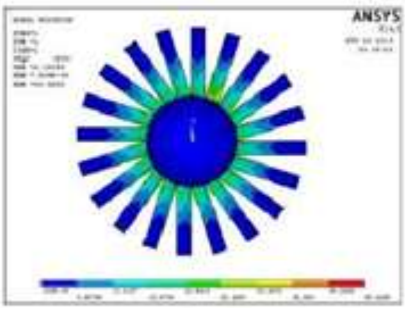

Fig 16: Stress Vonmises Vector

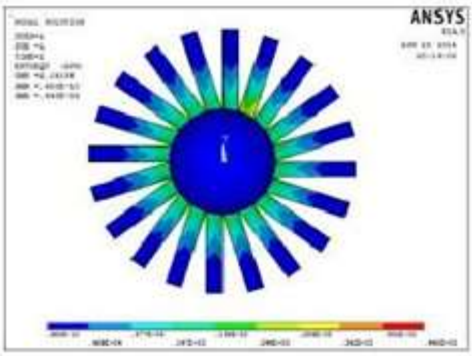

Fig 17: Strain vonmises

Titanium

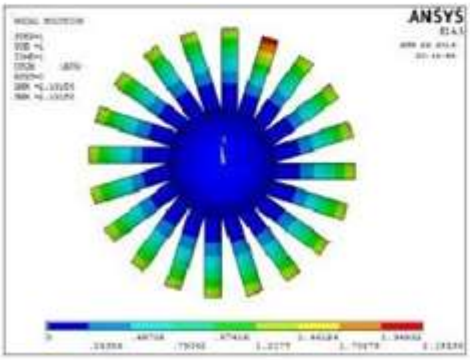

Fig 18: Displacement vector Vector

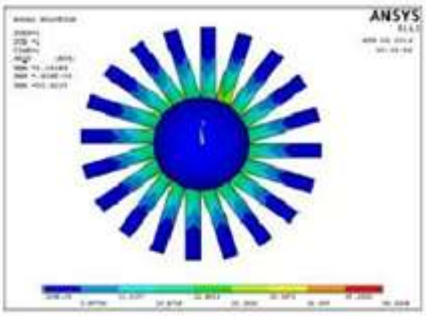

Fig 19: Stress vonmises vector

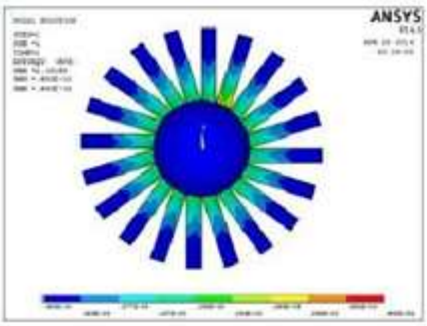

Fig 20: : Strain vonmises

Steel

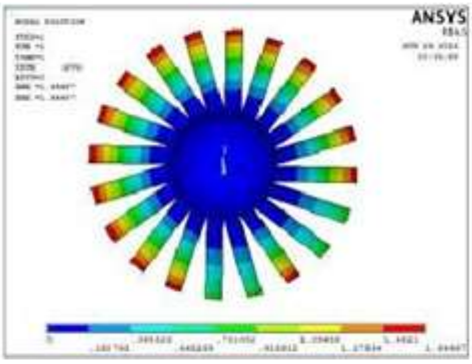

Fig 21: Displacement vector

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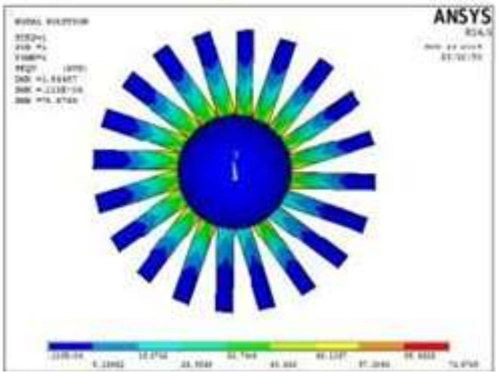


Fig 22: Stress vonmises vector

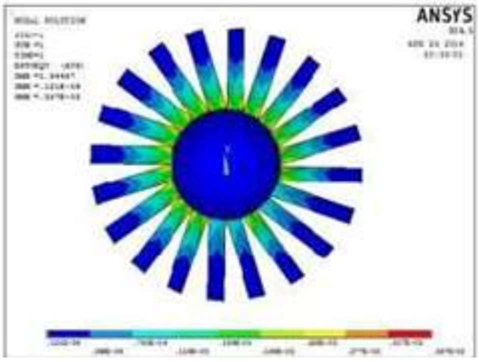


Fig 23: Strain vonmises vector

12 Blades

Nickel alloy

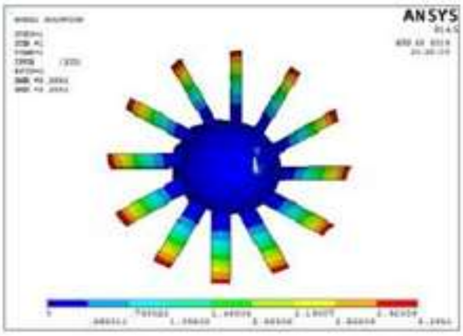
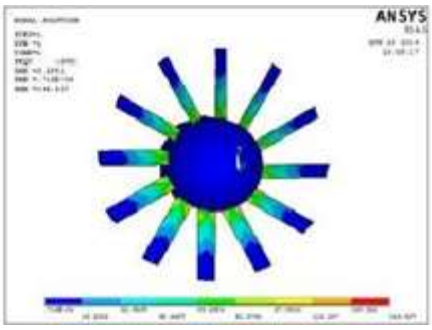


Fig 24:Displacement vector



or Fig 25: Stress vonmises vect

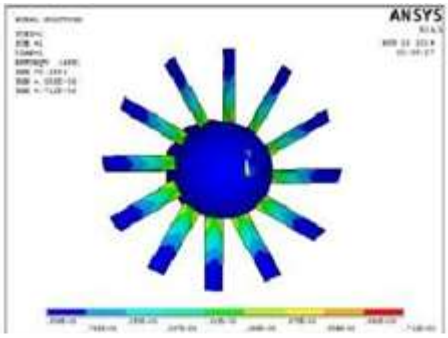


Fig 26: Strain vonmises vector

Titanium

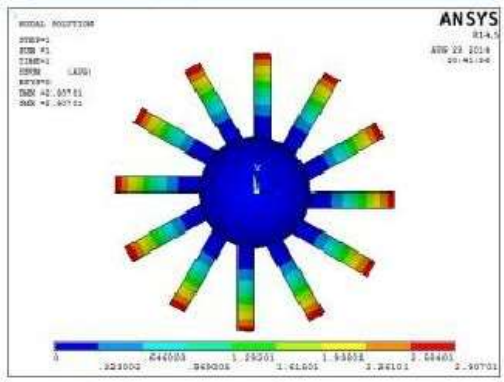


Fig 27: Displacement vector

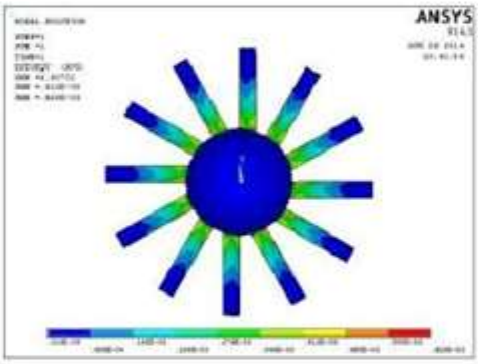


Fig : Strain vonmises vecto

Steel

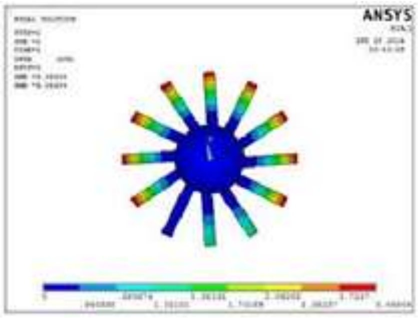


Fig 30: Displacement vector

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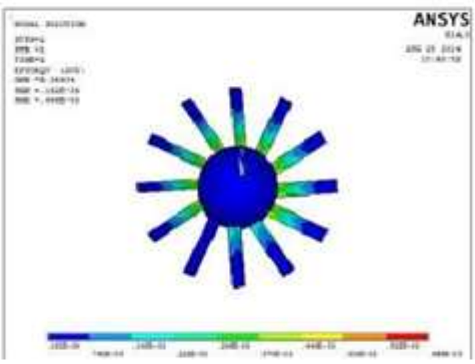

Fig :Strain vonmises vector

CFD ANALYSIS

Blades

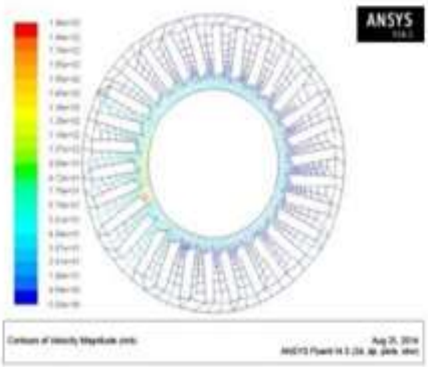

Fig 33: Velocity magnitude

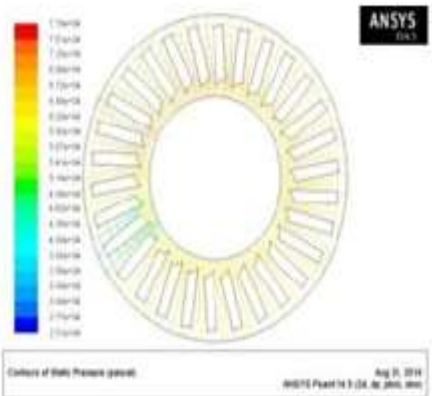

Fig 34: Static pressure

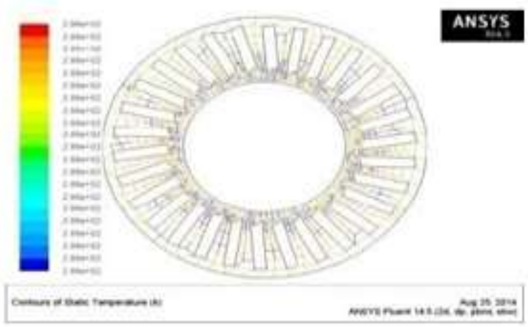

Fig 35: Temperature

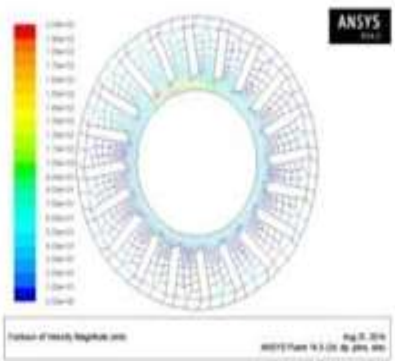

Fig 36:magnitude

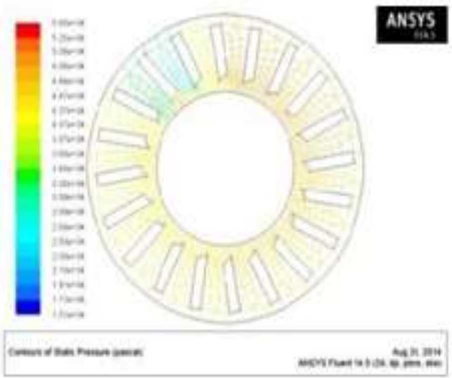

Fig 37: Static pressure

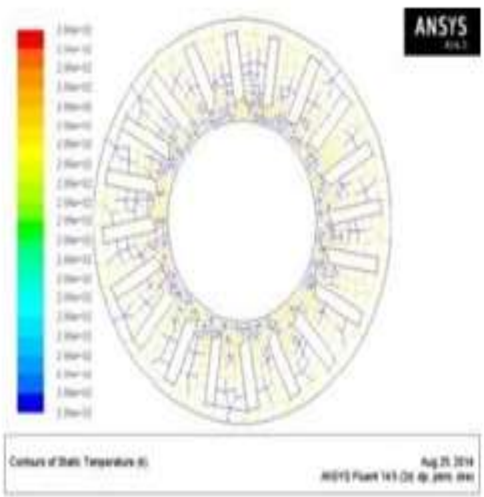

Fig 38: Temperature

12 Blades

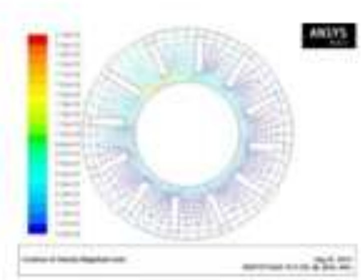

Fig 39: magnitude

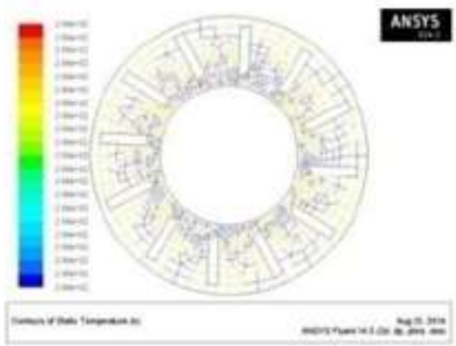

Fig 40:Static pressure

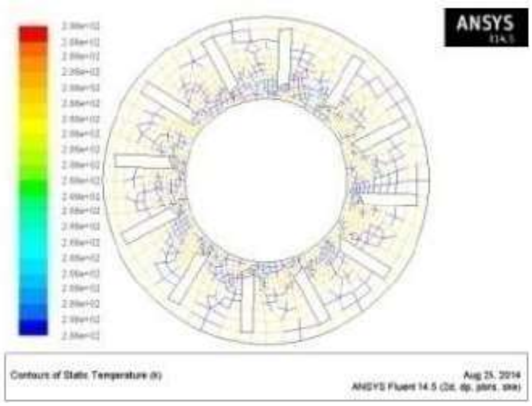

Fig 41:Temperature

RESULTS TABLE

At 30 blades

	Displacement(mm)	Stress(N/mm^2)	Strain
Nickel alloy	1.56293	66.4689	0.324e-03
Titanium	1.4195	34.0917	0.29e-03
Steel	1.40902	60.8752	0.29e-03

Table 1: Result of 30 blades

At 20 blades

	Displacement(mm)	Stress(N/mm2)	Strain
Nickel alloy	2.06664	99.2341	0.485e-03
Titanium	2.19186	50.920	0.440e-03
Steel	1.64487	73.6795	0.357e-03

Table no 2 Result of 20 blades

At 12 blades

	Displacement(mm)	Stress(N/mm^2)	Strain
Nickel alloy	3.2851	146.837	0.712e-03
Titanium	2.90701	72.5377	0.628e-03
Steel	3.06304	138.669	0.666e-03

Table no 3 Result of 12 blades

CFD Results

	30 blades	20 blades	12 blades
Velocity (m/s)	$1.94e^{+02}$	$2.00e^{+02}$	$2.14e^{+02}$
Pressure(N/mm^2)	$7.78e^{+04}$	$5.45e^{+04}$	$5.71e^{+04}$
Temperature(k)	$2.88e^{+02}$	$2.88e^{+02}$	$2.88e^{+02}$
Mass flow rate (kg/s)	0.081484798	0.1520251	0.23096226

Table no 4 CFD Result

CONCLUSION

In this thesis, an axial flow compressor is designed and modeled in 3D modeling software Pro/Engineer. The present design has 30 blades, in this thesis it is replaced with 20 blades and 12 blades. The present used material is Chromium Steel, it is replaced with Titanium alloy and Nickel alloy. Titanium alloy and Nickel alloy are high strength materials than Chromium Steel. The density of Titanium alloy is less than that of Chromium Steel and Nickel alloy. So using Titanium alloy for compressor blade decreases the weight of the compressor Structural analysis is done on the compressor models to verify the strength of the compressor. The stress values for less than the respective yield stress values for Titanium alloy and Nickel alloy. The stress value is less for titanium alloy than Nickel alloy, so using Titanium alloy is better. By using 12 blades the stresses are increasing, but are within the

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limits. CFD analysis is done to verify the flow of air. The outlet velocity is increasing for 12 blades, pressure is more for 30 blades and mass flow rate is more for 12 blades.

So it concluded that using Titanium alloy and 12 blades is better for compressor blade.

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