World Journal of Mechanics, 2014, 4, 332-347
Published Online November 2014 in SciRes. http://www.scirp.org/journal/wjm
http://dx.doi.org/10.4236/wjm.2014.411033

Structural and Conceptual Design Analysis of an Axial Compressor for a 100 MW Industrial Gas Turbine (IND100)

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Received 2 September 2014; revised 28 September 2014; accepted 21 October 2014

Abstract

The structural design of the IND100 axial compressor requires a multistage interrelationship between the thermodynamic, aerodynamic, mechanical design and structural integrity analysis of the component. These design criteria, sometimes act in opposition, hence engineering balance is employed within the specified design performance limits. This paper presents the structural and conceptual design of a sixteen stage single shaft high pressure compressor of IND100 with an overall pressure ratio of 12 and mass flow of 310 kg/s at ISOSLS conditions. Furthermore, in order to evaluate the conceptual design analysis, basic parameters like compressor sizing, load and blade mass, disc stress analysis, bearings and material selections, conceptual disc design and rotor dynamics are considered using existing tools and analytical technique. These techniques employed the basic thermodynamic and aerodynamic theory of axial flow compressors to determine the temperature and pressure for all stages, geometrical parameters, velocity triangle, and weight and stress calculations of the compressor disc using Sagerser Empirical Weight Estimation. The result analysis shows a constant hub diameter annulus configuration with compressor overall axial length of 3.75 m, tip blade speed of 301 m/s, maximum blade centrifugal force stress of 170 MPa, with major emphasis on industrial application for the structural component design selections.

Keywords

Gas-Turbine, Structural Design, Conceptual Design Analysis, Axial Compressor, Mechanical Integrity

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How to cite this paper: Aziaka, D.S., Osigwe, E.O. and Lebele-Alawa, B.T. (2014) Structural and Conceptual Design Analysis of an Axial Compressor for a 100 MW Industrial Gas Turbine (IND100). *World Journal of Mechanics*, **4**, 332-347.
http://dx.doi.org/10.4236/wjm.2014.411033

1. Introduction

The IND100 is a single shaft industrial gas turbine with a recuperator delivering a shaft output power of 100 MW at a combustor exit temperature of 1600 K, overall pressure ratio of 12 and inlet air flow of 310 kg/s at 3000 rpm. The IND100 has a sixteen stage axial compressor and a VIGV for part load operation, with a constant hub diameter annulus configuration and operating at ISOSLS conditions as shown in **Figure 1**.

Air flows axially into the compressor which has rotating blades on the discs and stationary vanes to direct the air stream into the next rows of blade [1]. The air stream moves in through the rotor blades, and the flow is later reduced in the stator blade passages whereas there is an increase in the pressure due to the shaft work done by the rotating blade which increases the total temperature and pressure, and there is continuous conversion of kinetic energy to static pressure by the rotor. The process is continued in similar manner in the sixteen stages to yield the required overall pressure ratio [2].

The design of multistage axial compressors have been investigated since 1808 by John Dumball, but his idea considered only moving blades without the stator aero foils to turn the flow into the subsequent stages. Further work done in 1885 presented an axial compressor design with low efficiency because the blades were not designed for conditions of pressure gradient rise in the flow direction [3].

The difficulties that categorized the development of axial flow compressors stemmed mainly due to flow processes [4], hence developmental investigations into fluid mechanics and aerodynamics advanced the design of axial compressors especially during World War 1 for the aviation industry as recorded by Cox (1946) and Constant (1945) and subsequently in industrial application.

The performance requirement of the axial compressor depends on their usage. In aircraft gas turbines, emphasis is on maximum work done per stage; high flow capacity in frontal areas, responds to demand of rapid acceleration over a wide range of operating conditions, and minimizes weight and length with high overall performance efficiency. Whereas in industrial application with lower rotational speed, weight and length are not major concerns but the design must meet with mechanical requirement and cost [4].

To establish the compressor structural design objectives, several aspects of design considerations are developed to provide for stable mechanical operations. Some of these considerations includes: thermodynamic and aerodynamic design, mechanical design, and structural and rotor dynamics. There is an interrelationship between these different design considerations; hence a multistage flow of information is required [5]. Axial compressors design is a major turbomachinery problem because of the adverse effect of pressure gradient. This paper has more emphasis on the structural mechanics/integrity, although a summary of the thermodynamic and aerodynamic design considerations has been put forward as part of this work. However, the scope of this paper is concerned with the conceptual design, sizing of the compressor, bearing selection, casing arrangement, blade load and stress analysis of structural integrity, shaft and disc arrangement, compressor assembly and material selections.

IND100 Schematics Description

The IND100 is a single shaft recuperated industrial gas turbine with sixteen stages of axial compressor and a VIGV with constant mean diameter annulus configuration, ten can-annular combustor, and four stages axial turbine with constant mean diameter annular configuration, operating with a combustor exit temperature of 1600 K, overall pressure ratio of 12, mass flow of 310 kg/s, 3000 rpm, and delivering 100 MW at ISOSLS conditions with generator coupled at the cold end drive.

2. Axial Compressor Structure Design Procedure

The axial compressor is designed by integrating a variety of knowledge which includes fluid mechanics and mechanical design to fulfill lower fuel consumption, higher pressure ratio and better performance efficiencies.

Figure 2 shows the compressor conceptual design procedure linking all the different disciplines.

2.1. Thermodynamic and Aerodynamic Design

The thermodynamic design analysis is used to determine the mass flow, pressure ratio and temperature distribution through the gas path. This result is employed into the aerodynamic design to determine the number of stages, rotational speed and the geometry of the compressor within the design performance limit of the de Haller num-

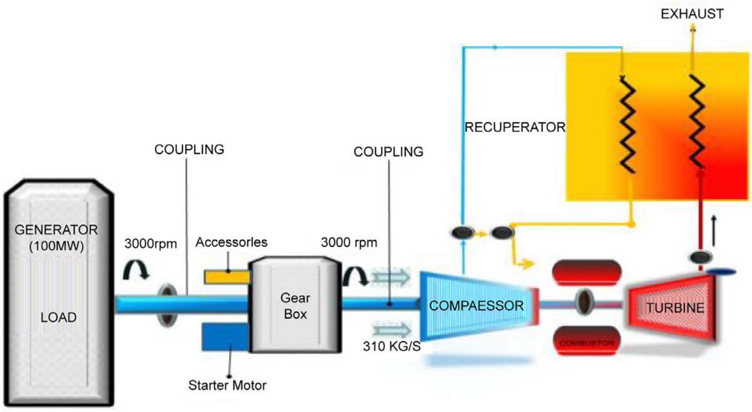

Figure 1. IND100 industrial gas turbine schematics.

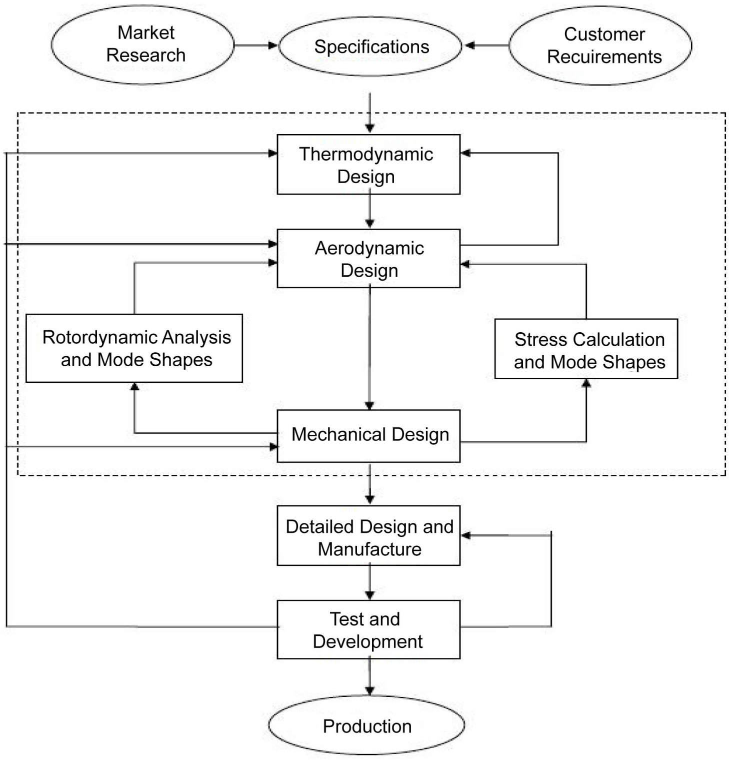

Figure 2. Compressor conceptual design procedure.

ber and other aerodynamic design performance criteria. These criteria have to be chosen for satisfactory blade loading, pressure rise at the walls and maximum Mach number [3].

The blade loading is assessed by diffusion factor derived by lieblein.

$$\mathrm{DF} = 1 - V_2/V_1 + \Delta V_\theta/2\sigma V_1 \tag{2.1}$$

where V_1 and V_2 are average velocity into and out of the blade rows in the frame of reference fixed to the blade, ΔV_θ is the change in whirl velocity in the row and σ is the solidity [3].

Decision is taken regarding the blade chord and the number of blades. Increasing the chord reduces the aspect ratio and increases solidity for the same annulus and number of blades [3]. There is need to also understand that the choice of stage loading, which has direct proportionality with the pressure rise in relation to the 16 stages and the rotational speed must be given proper considerations for a successful aerodynamic design [3].

The de Haller number characterizes the amount of diffusion over the blade and is given as;

$$V_2/V_1 \geq 0.72 \tag{2.2}$$

where, V_2 and V_1 are the velocities at the trailing and leading edge respectively.

The de Haller number is important to control the whirl velocity because a high whirl velocity implies a high fluid deflection and diffusion rate. Hence de Haller limit is set in this design as shown in **Figure 3** and is inversely proportional to the diffusion rate.

The stage loading coefficient Ψ shown in **Figure 4**, relates the blade speed to the increase in enthalpy.

$$\Psi = \Delta h/U^2 \tag{2.3}$$

where Δh is the enthalpy difference and U is the blade mid speed [1].

The pressure and temperature distribution across the stages is affected by tip speed due to the centrifugal stresses on the rotor blades. For aerodynamic reasons, axial velocity also has influence of the stage pressure ratio. These factors were considered during compressor design as shown in **Figure 5**. There was less temperature rise distribution at the first and last stage of the compressor to meet performance criteria.

From the **Figure 6**, the flow coefficient ϕ has an important influence on the compressor performance. It has direct relationship with the size of the compressor and the Mach number [2].

$$\phi = V_a/U \tag{2.4}$$

where V_a is the axial velocity. At constant blade angle, an increase in flow coefficient is equivalent to increasing the stage loading. This relationship is shown by the Equation (2.5).

$$\Psi = \phi\left\{\tan\beta_1 - \tan\beta_2\right\} \tag{2.5}$$

where β_1 and β_2 are the blade inlet and outlet angles.

The inlet Mach number distribution across the stages for both the rotor and stator was designed using the compressor flow chart table. This configuration was used to determine blade profile distributions at critical Mach numbers as shown in **Figure 7** and **Figure 8**.

2.2. Compressor Sizing

Compressor sizing could be quite tedious because it requires a comprehensive knowledge of the performance characteristics and weight. By making some assumptions on the hub to tip ratio, adiabatic efficiency, mean blade speed, pressure coefficient, and constant axial velocity for repeating stages enabled us make a preliminary deductions on the frame size as shown in **Figure 9**.

The chart above gives a correlation between the compressor inlet volume, hub to tip ratio and rotational speed. The annulus configuration for both rotor and stator from the assumptions of 160 m/s axial velocity, 0.55 hub to tip ratio is shown in **Figure 10** and **Figure 11**. The compressor length is determined using the Sagerser Empirical Weight Estimation [6].

$$C_L/D_m = 0.2 + \left\{0.234 - 0.218\left(Dh/Dt\right)\right\} * N \tag{2.6}$$

where N is the number of stages, Dm is the mean diameter C_L is the compressor length and Dh/Dt is the hub to tip ratio.

3. Mechanical Design

Mechanical design starts after the thermodynamic and aerodynamic designs have established important dimensions of the compressor. It includes the rotor system, bearings and bearings support, blade and disc design, and stress analysis.

3.1. Typical Life and Integrity Requirement

Mechanical design is focused on obtaining low-weight, high-integrity and balanced solutions for the compressors based on the aerodynamic performances and blade shapes required. The selections of material and component designs must match the life requirements at both steady and unsteady load conditions. Detailed vibration analysis

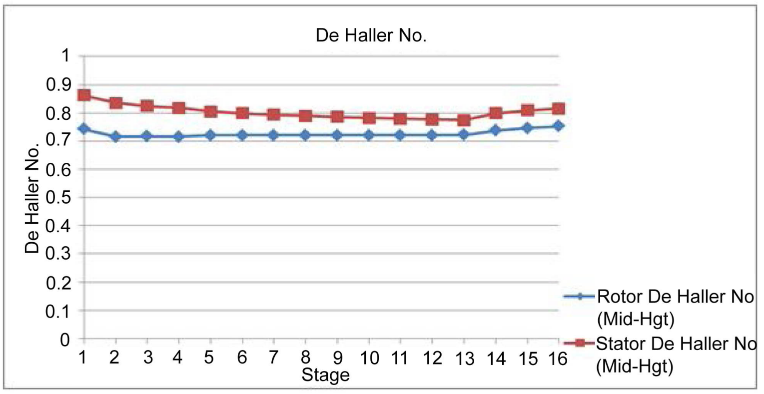

Figure 3. De Haller Number across the stages.

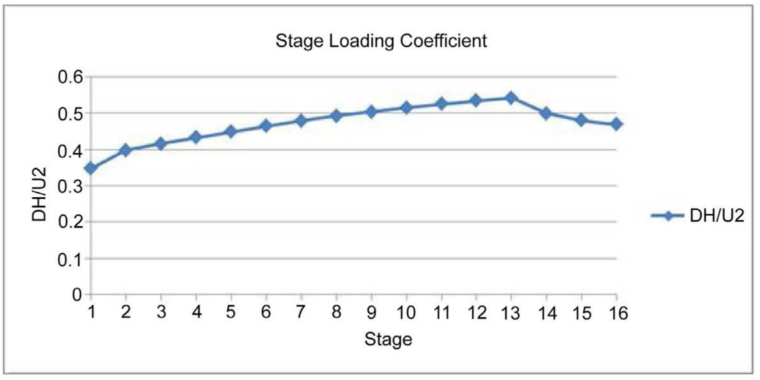

Figure 4. Stage loading coefficient across the 16 stages.

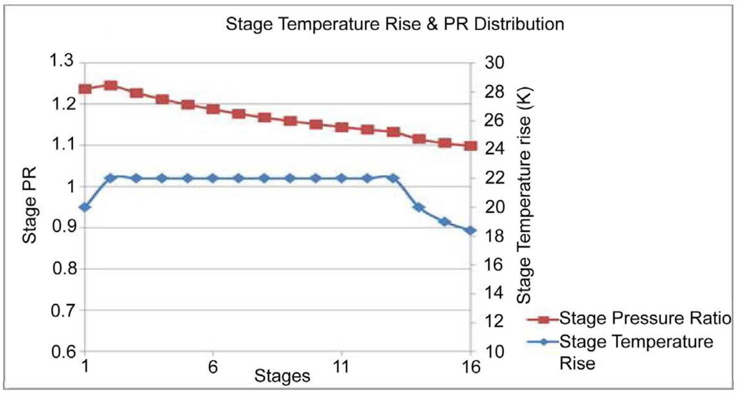

Figure 5. Temperature and pressure distribution across the stages.

is considered in order to ensure that potential resonance is avoided and when present, is within controllable limits. These can then be verified via telemetry testing on the engines over the whole operating range. Parts are also designed considering the manufacturing capability to ensure a seamless transition from design to manufacture [7].

3.2. Compressor Casing

The design of axial compressor casings could be difficult because of the variations in the inlet and outlet nozzles [8]. Casing can be made by casting or fabrication with steel or iron. Some designs have an inner shell that carries the stator vanes [8] while some have the outer casing carrying the stators directly. For these designs, the inlet section, center body with the stators and the discharge section are bolted at two vertical joints.

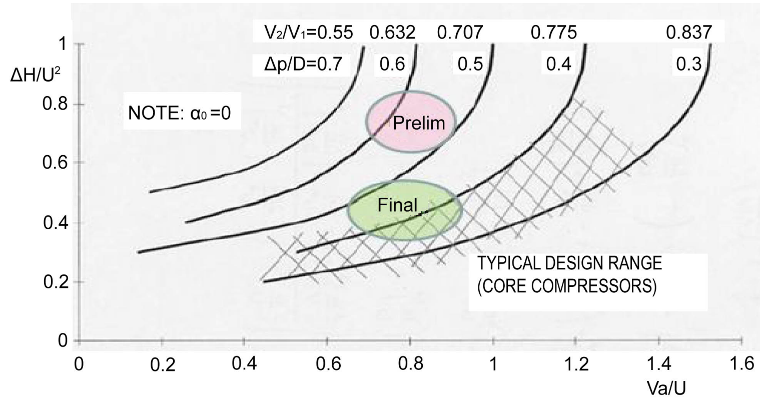

Figure 6. Design stages loading chart.

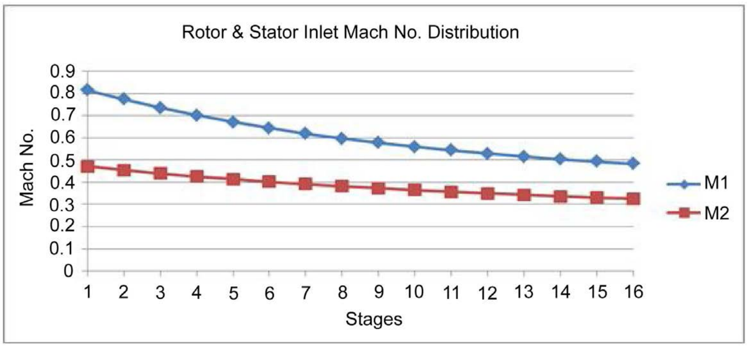

Figure 7. Inlet Mach number across the rotor and stator stages.

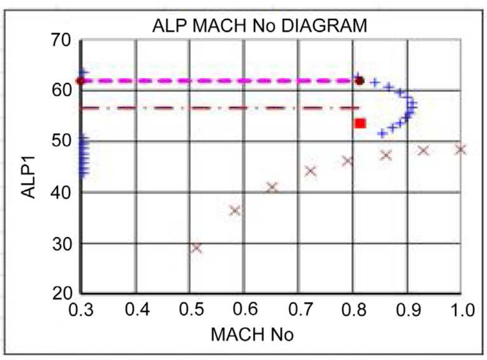

Figure: Alpha Mach No Diagram for 1st Stage

Stages	M1/M3	Mcrit
1	0.814	0.885
2	0.774	0.897
3	0.735	0.882
4	0.701	0.841
5	0.671	0.832
6	0.643	0.796
7	0.619	0.817
8	0.597	0.788
9	0.578	0.785
10	0.560	0.804
11	0.544	0.802
12	0.529	0.800
13	0.516	0.798
14	0.503	0.777
15	0.493	0.738
16	0.484	0.716

- Blading Limit Mcrit>M1

Figure 8. Blade profile distribution at critical Mach number.

The inlet casing is to channel the air stream into the compressor. The centre casing transfers structural loads from the adjoining casing to the forward support. The compressor discharge section is the final portion of the compressor section. It contains the final compressor stages, and forms a compartment of inner and outer walls of the compressor diffuser.

Compressor casing could have a horizontal split casing or a vertical split casing. The horizontal split consist of half casings joined along the horizontal centerline while the vertical split is formed by a cylinder closed at the end covers For our industrial application, horizontal split casing was considered to allow for ease of maintenance since the industrial engine requires a longer operating hours before maintenance.

The mounting feet are fixed to the outer casing to provide for centerline support. The design uses a rectangular inlet casing to make available more axial limit clearance. The purpose of this is to allow for strength and

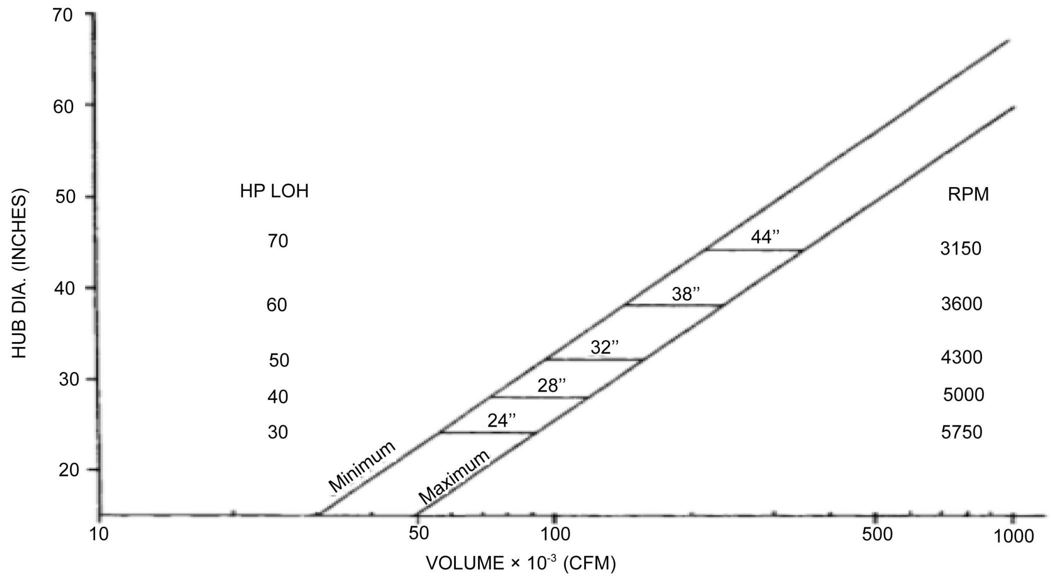

Figure 9. Compressor sizing chart (Royce Brown 1997).

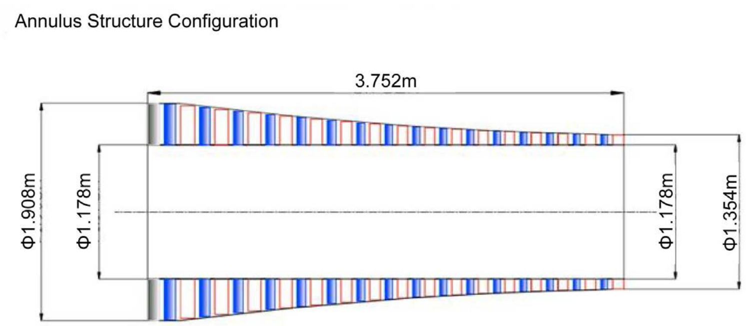

Figure 10. IND100 compressor annulus.

leakage testing [8].

The materials used for the casing design depends majorly on the operating temperature, pressure, and sizing. In general, Meehanite GD cast iron with 25 - 30 kg/mm tensile strength and 70 kg/mm compressive strength is used [9]. **Figure 11** shows the compressor casing assembly from the inlet to discharge and **Figure 12** describes a typical variable inlet guide vane casing for IND100 application.

3.3 Compressor Stator Design

The stators can be carried by separate inner casings or by the outer casing. The outer casing arrangement exposes the ends of the shank that are used as shaft to couple to linkages that controls the movement [8]. For movable stators, the vanes are passed through the wall of the carriers. Sometimes due to leakages at the bushings mount in the stator liners; single casings have sealing problems which can be handled using double casing configuration. Most single casing uses a lagging over the linkages, which acts as a collector.

Cast iron are used for the stator vanes if a separate inner casing design is employed and if the expected temperature is above 500 K, multipart liners with the discharge end made of steel are used.

3.4 Rotor Disc Design Configuration

The disc is to carry the rotating compressor; hence the energy contained in the disc is very high and must have high integrity. This integrity is achieved by careful design, highly controlled manufacture and thorough testing. A description of rotor disc design is shown in **Figure 13**.

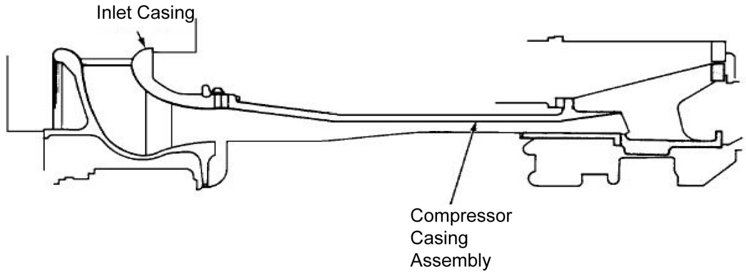

Figure 11. Casing structure (courtesy GE).

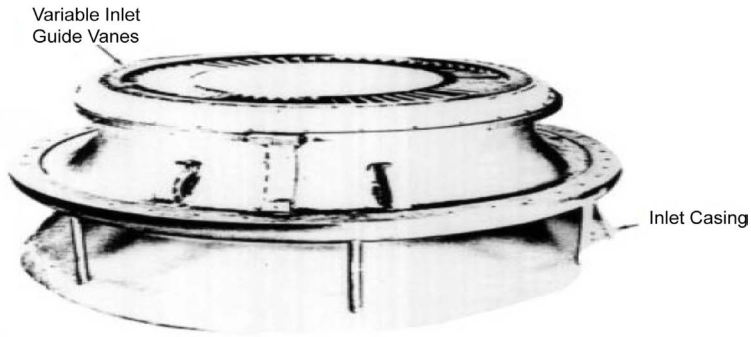

Figure 12. Variable inlet guide vane casing (GE).

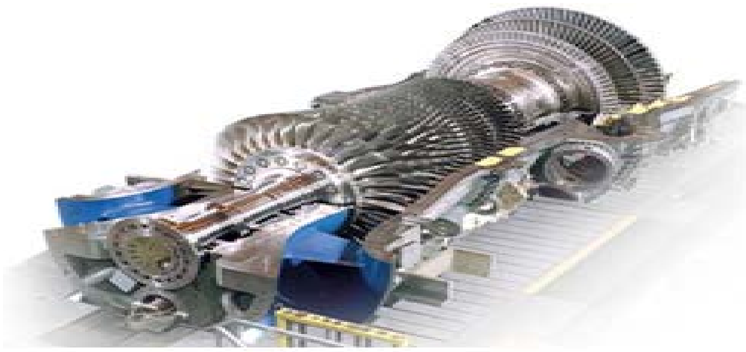

Figure 13. Rotor disc (GE).

The compressor blade is coupled to the outside surface of the rotor. There are basically two configurations of the rotor design; the basic disc configuration and the drum type configuration. The disc configurations also have variations with shrunk disc on a shaft design or stacked disc design which are bolted together.

The IND100 is to employ the disc construction method which has the blades linked by a single dovetail lobe root design [8]. The blade root is positioned in the slots and keyed in place while the slot is holed into the rotor.

For the shrunk disc design, they are arranged in stack into the shaft by pre-heating the disc to enlarge the bore and are allowed to cool, hence attaching themselves to the shaft after cooling.

The drum type rotor configurations, uses hollow drum and conical roots of bolted construction to allow for stagger adjustment when the need arises.

Low alloy steel with specified heat treatment is used for rotors in order to match the stresses imposed by the blades and rotor weight. Chrome Molybdenum alloy like AISI 4140 or AISI 4340 are used and are manufactured by forging [8].

3.5 Shaft Design

The need for high integrity in shaft designs is to avert catastrophe on the engine when they fail [10]. The design should be able to predict all possible stress sources so that the adequate shrink can be applied to the interface.

The shaft design for IND100 consists of central section with constant diameter as shown in **Figure 14**. Matching the shaft to the rotor construction for concentricity and to have precise control on the round shaft is very important, for a balance, and unconstrained running of the compressor.

A special type of shaft is used for the bolted disc or drum design configurations. The disc connects together with the shaft to form a hub and at the end of the hub is the stub shaft. Stub shafts are attached to both ends of the center and the bearings are fitted to it. The shaft is design to be as stiff as possible to avoid whirling and to reach its best flexural behavior. The shaft is connected to a gear box as shown in **Figure 15** via a flexible coupling and the function of the gearbox is to drive the accessories only.

Interference fit is used to ensure concentricity during operations and transient temperature variation at start up, shut down and hot restarts [8].

3.6. Shaft Manufacture

Shaft is commonly made of forged alloy materials like steel 40NiCrMo7 UNI suitable for hardening and tempering [9].

The shaft is long and has hollowed part; they need a high level of accuracy when manufacturing.

Traditionally, shafts manufacture methods poses considerably machining and checks challenges. Flow methods already in on some shaft drivers, are been developed for mainstream shafts. This cold extrusion process yields a preformed shape that is desired, which minimizes waste of materials. The pre-form are combined to give seamless components of improved hardness and strength [7].

3.7. Compressor Blade Design

The design specification of the compressor blade is of major importance in the overall compressor design. The

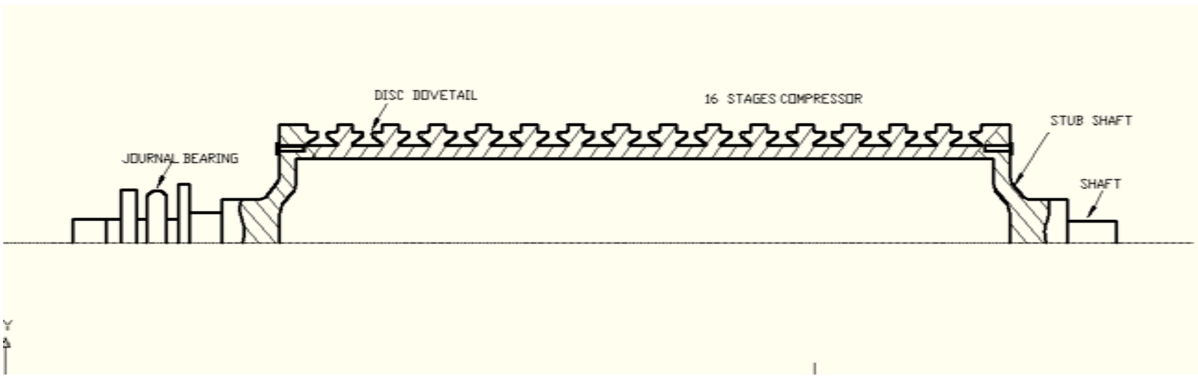

Figure 14. Shaft constant hub configuration.

Figure 15. Stub shaft.

axial compressor blade is an airfoil shape and is designed for compression of air efficiently at a high tip blade speed. To achieve this blade design objective, a blade to blade surface calculation is used to define the blade shape and is a way to describe the flow unlike in the past where correlations from series of standard profile were used [8].

The blade must be design not to fail due to gross over stressing or high cycle fatigue, hence a dovetail which is precise in size and position are used to keep the blades in their desired position and locations on the wheel. Both the stator and rotor blade is mounted in similar dovetail arrangement.

The aerodynamic blade profile design gives the stagger, solidity, camber angle, camber line shape (chord ratio) and thickness distribution. The chord line is the line passing through the trailing and leading edges while the camber line is curved and runs through the middle of the blade profile. Having obtained the blade radius, inlet and outlet angles and the nominal deflections, the next step is to determine the chord length which depends on the pitch and the number of blades in the row. During the choice of the number of blades the aspect ratio is considered because of its effect on secondary losses.

Some of the governing equations used to determine the blade profile are as follows;

$$\delta = m\theta\sqrt{(s/c)} \tag{3.1}$$

$$m = 0.23(2b/c)^2 + 0.1(a_2/50) \tag{3.2}$$

$$n = 2\Pi * r_m/s \tag{3.3}$$

where, δ is the blade deviation, θ is the camber angle, n is the number of blades, a is the distance of the point of maximum camber, c is the chord, s is the pitch.

The number of blades to be chosen is usually odd number to reduce the impact of vibration frequency. Lashing wires are also used on the rotor blades to solve blade vibration stress problems. The basic resonance and direct excitation source should be considered using stator or splitter and guide vanes to pass the frequencies [8].

3.8. The Effect of Blade Shape

The known ways of blade shape design is to base the design on families of profile such as the C-series, NACA-65 series and the Double Circular Arc (DCA). These blade profiles have their advantages and disadvantages at different Mach number operation and at blade thickness. The effect of the blade thickness and Mach number are associated with losses which could lead to flow separation. **Figure 16** and **Figure 17** show comparison

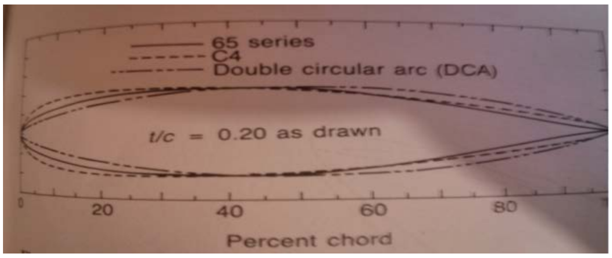

Figure 16. Comparison of thickness distribution on three common profiles (Royce Brown 1997).

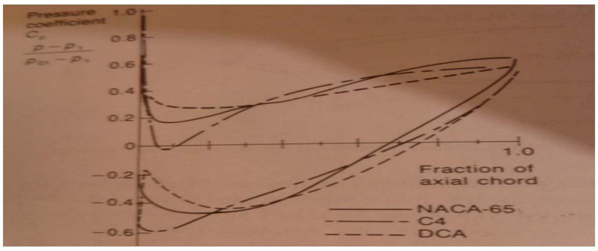

Figure 17. Pressure distribution on the three blade profile (Royce Brown 1997).

on thickness distribution and pressure distribution among the blade profiles.

3.9. Blade Manufacture

The axial compressor blade is made mainly by précised hot forging of extruded billet, which is continued with machining process on the root feature and finishing operations. After forging, the surfaces of the blades are polished to obtain highly unconstrained finishing which is important to achieve better aerodynamic design performance. When the shape of the blade becomes too complex with optimized leading profile edges, other manual techniques are employed for the finishing by automated processes. The reason for the technologies is to achieve better quality and repeatability of the leading profile edge features and to allow the increase of production quantity. Achieving these techniques is based upon advances in the capability of modeling the manufacturing process, and also on-condition monitoring processes, like the polishing to adapt to wearing on the wheel and for self-dressing [7].

3.10. Blade Materials

The common materials used for compressor blades are the 12 chrome steel AISI 400 which is also called stainless steel 400 series and type 403 [8]. These groups of alloys have properties that include good ductility at high strength levels, at temperature up to 755 K. High resistance to corrosion is also obtained because of the high contents of molybdenum and chromium.

3.11. Bearing Selections

Bearings are devices that permits constrained relative movement on the surface of two parts, either rotational or linearly. Axial Compressors uses majorly the journal bearings and their designs consist of partial arc pads with circular geometry.

The industrial gas turbine IND100 uses three journal bearings to support the rotor and a thrust bearing to keep the rotor and stator axial position. The journal bearing have the plain sleeve type for larger compressor running at low speed and the tilting pad type for smaller compressor at high rpm. The thrust bearing are mostly the tilting pad type because of high thrust load [8]. A typical of journal and thrust bearings used in this design is shown in **Figure 18**.

The bearing design and configuration are crucial because bearings must be able to run smoothly and reliably under varying engine conditions. They transfer the shaft loads unto the main engine structure, hence are to be reliably strong to adapt with shaft unbalancing, which could occur when blades fail.

Aviation engines use the ball and roller bearings in its applications. High-precision bearings and bearings load management systems are employed for weight and reliability demands on the aircraft engines.

Bearings and seal are incorporated in the compressor inlet casing, discharge casing and exhaust frame. Lubrication oil is passed into the bearing housing and flows into an annulus and to the bearing rotor interface through the machined slots for smooth and cooling operations on the bearings.

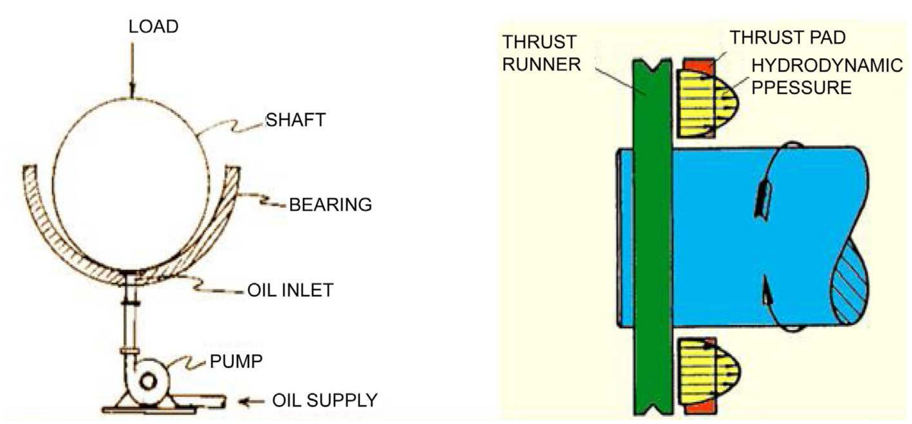

Figure 18. Journal and thrust bearings.

3.12. Bearing Materials

Bearings are made of carbon steel materials with tin alloy coatings.

3.13. Seals

Axial compressor is most equipped with labyrinth type end seals. The seal is design to restrict flow by making the fluid pass through series of ridges and intricate paths. Labyrinth seals are categorized into the ring type oil seal, thread type, fluid abradable and continuous groove interstage air seal

3.14. Weight and Stress Analysis

The axial compressor weight is affected by the rotor and stator blades, seals, rotor/disc, casings and tip speed. Using the Sagerser Empirical Weight Estimation [6]

$$C_w = K\left(\dot{D}_m\right)^{2+a} * \left(N\right)^{1+b} * U_t^c \tag{3.4}$$

where, $\dot{D}_m$ is the average mean diameter and the equation assumes that the ratio of compressor length to inlet mean diameter is relatively constant.

Figure 19 shows weight correlation of compressor using the Sagerser Emprical Weight Estimation

Typical axial load estimation on the IND100 is shown in **Table 1**.

The major sources of load on the compressor are:

- Centrifugal forces acting on sections of the aero foil and are caused by actions of inertia.
- Gas bending moment that is caused by changes in momentum and pressures of the fluid passing through the blades.

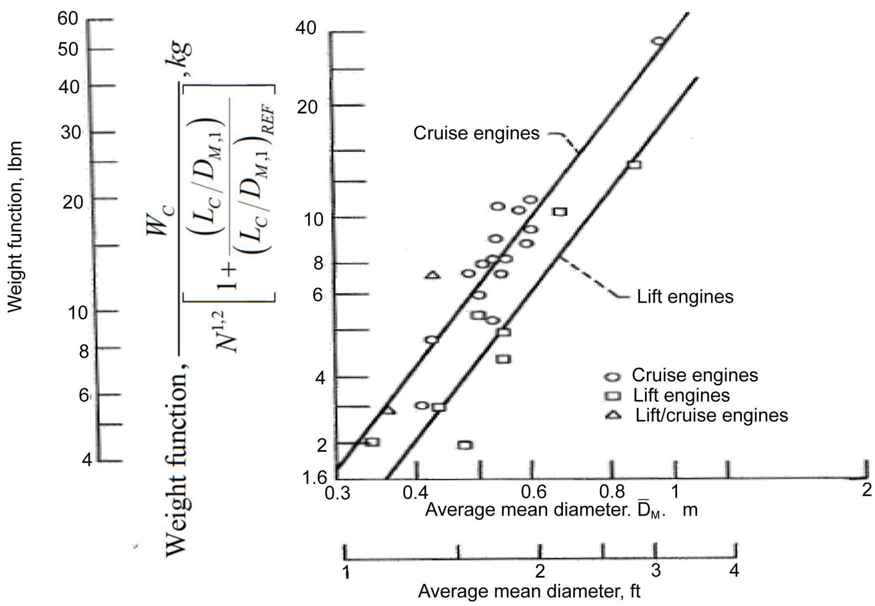

Figure 19. Compressor weight correlation chart.

Table 1. Axial rotor force of IND100.

<table>
<tr><th></th><th>1</th><th>2</th><th>3</th><th>4</th></tr>
<tr><td>Pressure (Kpa)</td><td>101</td><td>1089</td><td>1035</td><td>103.02</td></tr>
<tr><td>Area (Sq Meter)</td><td>1.88</td><td>0.8</td><td>0.8</td><td>1.88</td></tr>
<tr><td>Axial Force (KN)</td><td>+189.88</td><td>−413.80</td><td>+393.3</td><td>−193.67</td></tr>
<tr><td>Net Force (KN)</td><td colspan="2">−223.93</td><td colspan="2">199.63</td></tr>
<tr><td>Net Force on Shaft (KN)</td><td colspan="4">−24.30</td></tr>
</table>

- Bending moment caused by centrifugal load on points that are radially above the centroid.
- Shear load and complex load due to thermal gradients [11].

The gas passage on the axial compressor exposes it to unusual stresses which must be considered in the design of axial compressor. Some basic governing equations for stress analysis are as follows;

$$\mathrm{CF} = m * r_{cg} * \omega^2 \tag{3.5}$$

$$F = m * \mathrm{d}V/\mathrm{d}t \tag{3.6}$$

$$Mv = m * \mathrm{d}V * r \tag{3.7}$$

where CF is the centrifugal force, Mv is the bending moment, F is the force produced due to change in momentum of the gases.

4. Compressor Assembly and Operations

Considerations must be given on how each of the compressor components are assembled and fitted into the overall engine design. Each component of the turbomachinery is built in assembly and coupled to other engine component. An overview of the overall axial compressor structure design for the IND100 is shown in **Figure 20**.

Split casing along two diametrically opposite axial lines with the stators assembled in each half of the casing and the rotor built as a separate unit. The two halves of the casing are then coupled and bolted along their axial joint. Another approach is to divide the casings into series of rings, one ring for each row of stators. The assembly of the components are built in stages from one end with a support ring to a jig for the stators, rotor disc and shaft with blades fitted using the jig [10].

4.1. Couplings

A rigid hollow coupling connects the gas turbine rotor shaft to the gas generator. A bolted flanged connection forms the joint at each end of the coupling. Flexible coupling is used to connect the accessory drive to the gas turbine shaft at the compressor end. The coupling is design to transmit the starting and driving torque and also provide flexibility to accommodate both angular and parallel misalignment and allow for axial movement [7].

4.2. Spline Joint

The spline joint is one that has grooves cut length wise on the outside of the shaft joint for short distance and other shaft joint with corresponding groove cut along its inside. The shafts are slid together and secured axially by bolting.

4.3. Blade Fixing

The various methods used in fixing blades to disc includes; circumferential dovetail which involves making circumferential slot round the rim of the disc into which the blades slide and feeding the blades in slot using grub screw to prevent them from sliding along the rim. Pan fixing is another method employed for compressor blade. In this method, the blade is fixed to the disc by axial pin and is held in position against centrifugal forces. The Blisk or Bling use integral blade and disc arrangement. The blade and disc are single component and any failure will affect both components [10].

4.4. Bearing Position

Bearings are usually located near the ends of the component shaft. For the IND100 application, the first journal bearing is at the center of the inlet casing assembly and also has a thrust bearing attached. The second bearing is located at the compressor discharge sections**.** Its supports ledges at the horizontal and an axial key is located to the bottom centerline which permits relative growths resulting from temperature differences while the bearing are still at the centered of the discharge section. There is bearing liner, labyrinth seals, and a bearing housing included in this assembly. The bearing liner is prevented from rotating with the shaft by an antirotation pin located at the other lower half of the bearing liner [12]. **Figure 21** shows the bearing location for the IND100 gas turbine.

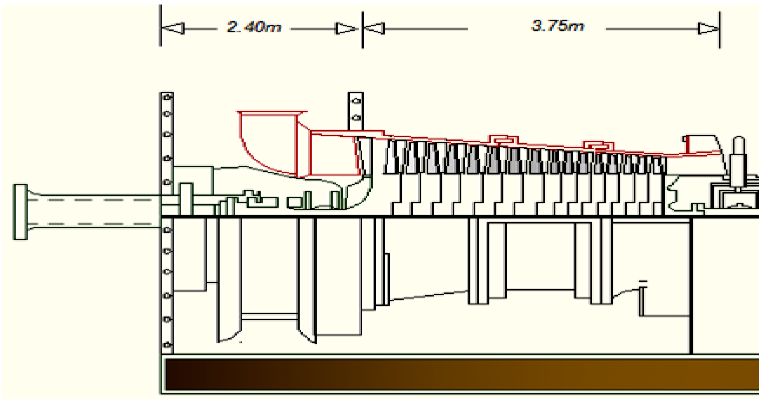

Figure 20. IND100 overall axial compressor structure.

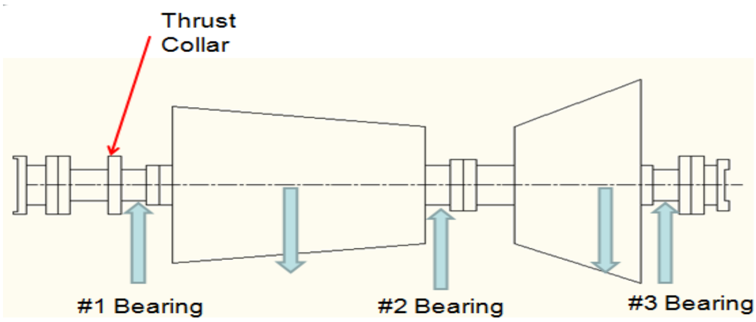

Figure 21. Bearing location.

4.5 Variable Inlet Guide Vane Operating Mechanism

It allows the flow inlet angle to be changed in order to meet the engine operating conditions [13]. This is located at the front of the axial compressor and closes during engine starting at low rpm but is opened as the engine rpm increases. It also helps to avoid compressor stalling at low rpm by keeping the low pressure blades angle of attack moderate.

The variable inlet guide vane is coupled at the shaft end of the compressor inlet section and is mechanically positioned, by a control ring and pinion gear arrangement connected to a hydraulic actuator drive and linkage arm assembly. The position of these vanes affects the quantity of compressor inlet air flow.

4.6. Effect of Tip Clearance Due to Heat and Centrifugal Force

As the axial compressor is subjected to varying thermal and centrifugal loads (mechanical), this produces small changes in both the radial and the axial dimension of the component resulting in relative movement between rotating and stationary components and causing variations in the performance efficiency and flow.

One of the reasons for these changes in tip clearance is thermal growth of the component [14]. In transient conditions, the axial compressor components expand and this expansion takes place at different rates because of the material absorption rate, heat transfer coefficient and component shape.

Another factor that effect tip clearance changes is the centrifugal expansion due to changes in the rpm. These changes could be seen in the casing, rotor disc and blade expansion rates.

5. Vibrations and Balancing

Axial compressors are reaction type of machines because of the adverse pressure gradient. It means that the rotor disc is exposed to pressure differentials along the rotating rows of blades which could cause unwanted vibration, hence can be controlled in the design.

Unbalancing is a major cause of vibrations in axial compressors and if unchecked could damage bearings, seals and in severe case causes major component failure or fatigue [8]. An unbalance occurs when the centre of the mass is displaced from the rotating centre.

$$F = mr\omega^2 \tag{5.1}$$

Balancing can be achieved by putting the shaft on knife edge. This is done by rotating the disc gravitation until the mass is at the bottom. Dynamic balancing is also employed [8].

Resonance within the axial compressor is another source of vibration. Usually gas turbines components operate at resonant frequency below the operating frequency. Measurement probes are used to monitor the frequency operations of the components to control vibrations.

Other sources of vibration are compression temperature rise, or driver induced vibration. An understanding on the nature of the sources of vibrations and how to keep them from mixing with resonance frequencies of the compressor is needed.

6. Structural Integrity

The whole compressor design modeling facilitates the complete design of structures for optimum weight, rotor dynamics ahead of physical validations, blade tip clearance to enhance the component performance and operability at different conditions. This will provide an environment for analyzing the rotor dynamics, representative engine loads distribution and the integral of the individual structural models.

Structural validation is necessary to ensure that the compressor performs optimally without sudden component failure on test bed runs and during nominal operations.

7. Conclusions

Work is transmitted from the turbine to drive the compressor through a single shaft and spline system arrangement as described in the IND100 applications, with the journal and thrust bearings providing the radial and axial positioning on the rotating component.

The vital structural design of the axial compressor includes the compressor rotor and the compressor casings. Within the compressor casings are the variable inlet guide vanes, the various stages of rotor and stator blades, and the exit guide vanes. The design of these components needs an integration of thermodynamic, aerodynamic and mechanical design considerations to form an integral structure.

TURBOMATCH of Cranfield university, a computer based software that has a graphical user interface (GUI) capable of simulating gas turbine engine is used to ascertain the inlet and outlet temperature and pressure condition of the compressor which help to achieve the change in temperature distribution across the entire stages of the compressor.

Also, analytical empirical estimation was utilized; hence design judgment and selections must be matched with structural integrity.

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Nomenclature

CF: Centrifugal Force; N/m
CW: Compressor Weight; kg
D_m : Mean Diameter; m
DF: Diffusion Factor
N : Number of Stages; Kj/kgK
h : Enthalpy; Kg
m: mass; m/s
U: Blade Speed; m/s^2
V: Velocity
HPC: High Pressure Compressor
VIGV: Variable Inlet Guide Vane
ISOSLS: International Standard Organization Sea Level Static
ϕ : Flow Coefficient
ω : Angular Velocity; rad/sec
ψ : Stage Loading Coefficient
δ : Blade Deviation