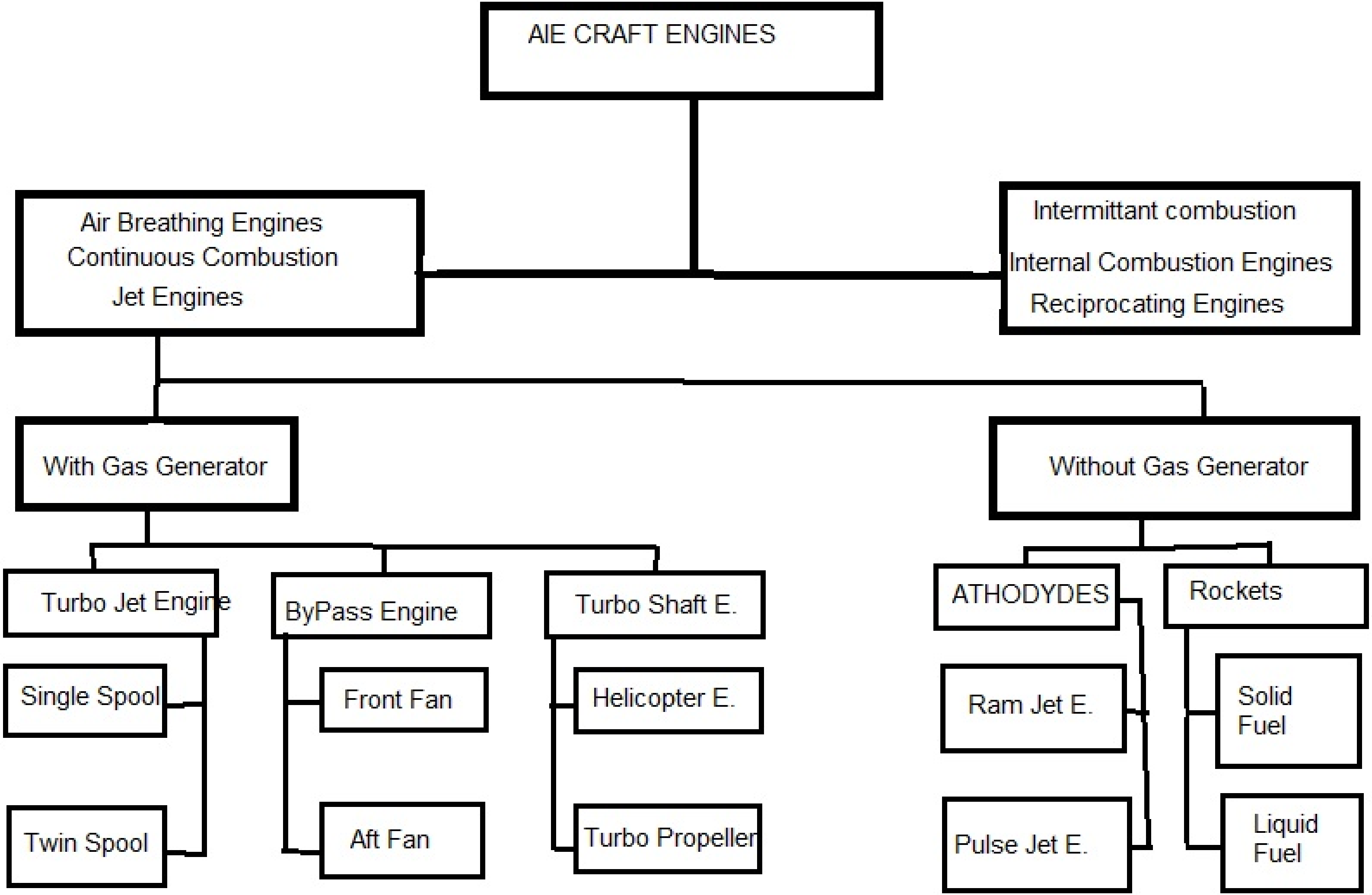


AIE CRAFT ENGINES
Air Breathing Engines
Continuous Combustion
Jet Engines
Intermittant combustion
Internal Combustion Engines
Reciprocating Engines
With Gas Generator
Without Gas Generator
Turbo Jet Engine
ByPass Engine
Turbo Shaft E.
ATHODYDES
Rockets
Single Spool
Twin Spool
Front Fan
Aft Fan
Helicopter E.
Turbo Propeller
Ram Jet E.
Pulse Jet E.
Solid Fuel
Liquid Fuel

Internal Combustion Engines:

Assisting books:

1. Basic Engineering Thermodynamics in SI units…by:Joel.
2. ICE and air pollution…..by: Edward F. Obert.
3. Motor Vehicle Engine……by: M. Khovakh.

Ch1. Classification:

1. Purpose: a. stationary (pumps, compressors, generator.), b. Vehicle
2. Fuel type: a. Petrol (PE), b. Diesel (DE), c. Gas (GE), d. Mixed, e. Multi (MFE)
3. Cycle: a. 2-stroke, b. 4-stroke
4. Ignition of working mixture: a. Spark (SIE), b. Compression (CIE).
5. Speed of piston: a. low speed (≤6 m/s), b. Medium ($9 \geq C_p \geq 6$), c. High (≥9 m/s).
6. Charging: a. Atmospheric, b. Supercharging.
7. Piston motion: a. Reciprocating, b. Rotary (Planetary).
8. Number of cylinders: a. Single cylinder, b. Multi cylinder.
9. Cooling: a. Air, b. Liquid
10. Cylinder arrangement: a. In line, b. V-type, c. Radial, d. Apposed.

Euler Pump and Turbine Equation

Consider the rotor of a turbomachine as being a rotating disc which takes fluid on board at station 1 with velocity V_1 and discharges the same quantity at station 2 with velocity V_2.

Let V_a…axial component of V

V_w…Whirl (tangential) componenet of V

V_r…Radial component of V.

The net turning force acting on the fluid (which is equal and opposite to that acting on the rotor), is given by Neqton's second law of motion but applied only to the tangential component, since the axial and radial components produce no turning effect.

$F_w = m\, a_w$, $a_w = (V_{w2} - V_{w1})/t$

$F_w = \dot{m}\,(V_{w2} - V_{w1})$

Torque = Force x Radius

$= \dot{m}\,(V_{w2} r_2 - V_{w1} r_1)$

= rate of change of angular momentum.

Power= Torque x angular velocity

$= \dot{m}\omega(V_{w2} r_2 - V_{w1} r_1)$ …….Euler pump and Turbine Equation.

Let us resolve the velocities into two components:

V_R….Air relative velocity

U….Blade liner velocity

Using cosine law:

$V_{R1}^2 = V_1^2 + U_1^2 - 2V_1U_1\cos\alpha_1$

$V_{R2}^2 = V_2^2 + U_2^2 - 2V_2U_2\cos\alpha_2$

$V_{w1}=V_1\cos\alpha_1$…………….$V_{w2}=V_2\cos\alpha_2$

Substituting into Euler Pump and Turbine Equation:

$$poewer = \dot{m}(\left[\frac{V_2^2-V_1^2}{2}\right]+\left[\frac{U_2^2-U_1^1}{2}\right]+\left[\frac{V_{R!}^2-V_{R2}^2}{2}\right])$$

Where:

$\frac{1}{2}\ \dot{m}\ V_2^2-V_1^2$ External Effect (absolute)

$\frac{1}{2}\ \dot{m}\ U_2^2-U_1^2$ Centrifugal Effect (Kinetic energy change due to radius change)

$\frac{1}{2}\ \dot{m}\ VR_1^2-V_{R2}^2$ Diffusion Effect (Kinetic energy change due to velocity change in the blade passage flow area).

Effect/Type of machine	External Effect in Stator	Centrifugal Effect in Rotor	Diffusion Effect in Rotor
CFC	Large effect in diffuser after rotor $V_2>V_1$	Large effect $U_2>U_1$ outward flow	Negligible $V_{R2}=V_{R1}$
RFT	Large effect in nozzle prior to rotor $V_1>V_2$	Large effect $U_1>U_2$ inward flow	Negligible $V_{R2}=V_{R1}$
AFC	Large effect in diffuser after rotor $V_2>V_1$	Constant radius Not utilized	Large in diverging blade passage $V_{R!}>V_{R2}$
AFT	Large effect in nozzle prior to rotor $V_1>V_2$	Constant radius Not utilized	Large in converging blade passage $V_{R2}>V_{R1}$

Ch.2: Compressors

2.1 Centrifugal Flow Compressors: CFC

CFCs comprise a rotating impeller and a fixed set of diffusers. The impeller is a rotating disc with radially disposed vanes, air entering the eye if the impeller is flung outwards by the vanes which impart kinetic energy to it. This energy is then converted to pressure as the air flows through the fixed diffusers which also turn the air flow from radial to axial direction.

Construction:

The impeller may be either single or double sided. Double sided impellers have smaller diameter for a given air mass flow rate, greater intake losses, more air flow instability and higher stresses by the two sets of blades hence lower rotational speed limits.

Modern impellers are machined from solid forgings of Titanium alloys and polished. They are attached to the shaft by flanged fittings and balanced statically and dynamically and tested to 5% over speed.

Rotating Guide Vanes: In order for the air to enter the impeller at the correct angle: guide vanes inclined in the direction of rotation must be fitted at the eye of the impeller.

Diffusers: On leaving the impeller the air enters a clearance space before passing into the diffuser. The purpose of this space is to smooth the air flow and avoid buffeting and turbulence. The diffuser passages are divergent channels between vanes machined on to the diffuser casing. The design of the diffuser is critical since velocities at entry are very high (M0.9) and there is a risk of flow break away if the angle of divergence exceeds 10^0. The air is led from the diffuser into the ducts in which the 90^0 turn is to bring the flow back to axial direction. Cascade vanes are fitted in the ducts to ensure smooth flow through this turn.

Characteristics of CFCs:

Drawbacks:

1. Limited tip speed (460m/s) hence limited pressure ratios 4-6.

2. Multi staging possible but difficult (ex. dart turbo propeller).

3. Large frontal area.

4. Air flow per unit area is low.

5. Isentropic efficiency is less than AFCs.

Advantages:

6. Very robust.

7. Very short>

8. Less sensitive to off design operation.

Inlet Conditions: To ensure smooth entry of the air into the impeller passages, the impeller vanes must be inclined at the correct angle.

U ↑as V↑ and α↓ as r ↑

$\tan \alpha_1 = V_1/U_1$ $\quad\quad V_{w1}=0$

Exit conditions: $\tan \alpha_2 = V_{b2}/U_2$ $\quad\quad V_{w2}=U_2$

Power Input: Applying Euler Pump and Turbine Equation:

$W = \dot{m} (V_{w2}U_2 - V_{w1} U_1)$

Since $V_{w1}=0$

$W = \dot{m} V_{w2} U_2 = \dot{m} U_2^2$ as $V_{w2}=U_2$

Thus the power input $=f(\dot{m}$ and $U_2)$

Pressure Ratio: The power input is also given by the SFEE

$W = H_t = \dot{m} C_p (T_{t2}-T_{t1})$

If the isentropic efficiency is known:

<u>Slip Factor:</u> In the simple compressor we assumed that $V_{w2} = U_2$. However in the impeller passages the air will due to inertia maintain its orientation in space. Relative to the impeller the air will have a rotational velocity in the opposite direction. This rotational velocity is superimposed on the radial velocity so that V_{w2} will thus be less than U_2.

To take this slip into account a slip factor σ is introduced.

The slip factor depends on the number of impeller vanes for a typical modern centrifugal compressor with 19 vanes; the slip factor will be approximately 0.9 while for 21 vanes σ = 0.93 .However reducing σ increases the power input required and reduces the available flow area.

Based on experimental study the slip factor in general $\sigma=1-(0.63\pi/n)$

Where: n is number of vanes.

<u>Power Input Factor:</u> A second modifying term is required to account for:

- friction in impeller passages,
- leakage past impeller vanes,
- heat transfer.

Thus the power input factor:

Φ=(turbine input power to compressor)/(compressor in put power to working fluid)

so: $W=\dot{m}\sigma\Phi U_2^2$

Typical values $1.035 < \Phi < 1.04$

<u>Pre whirl:</u> $\alpha_1 \downarrow$ $as\ radius \uparrow \quad and\ Vb1 \uparrow as\ R \uparrow$

V_{b1} at the tip may become very large resulting in large frictional losses and the possibility of transonic flow with shocks and separation on the convex surfaces. This can be avoided by using fixed guide vanes at the entrance to the intake to give the air a pre whirl component in the direction of rotation. This reduces the curvature on the impeller vanes (helpful to the manufacturer), reduces the relative velocities. But also reduces the work input to: $W=\dot{m}\Phi(\sigma U_2^2 - V_{w1}U_1)$ which reduces the available pressure rise, (reduces the work capacity of the compressor)

<u>Impeller Vanes:</u> In the simple compressor we consider the impeller to have radial vanes, but these could have been swept either backwards or forwards, in relation to the direction of impeller rotation.

As a comparison consider impellers of equal mass flow rate driven at equal rotational speeds. We can see that the backward curved impeller vanes give a reduced impeller whirl velocity V_{w2} and hence reduced pressure ratio. Whilst the forward curved vanes increases V_{w2} and hence pressure ratio in comparison with the radial vane impeller. If the volume flow rate is now increased (the dotted lines) then V_{w2} is reduced for an impeller with backward curved vanes and increased for one with forward curved vanes. In comparison with the radial vaned impeller with backward curved vanes can handle a large range of flow rates at low pressure ratios. The forward curved impeller can provide high pressure ratios but is limited to a narrow range of operation. The aircraft gas turbine requires both a large range of operation and high pressure ratios. The radial vaned impeller offers a reasonable compromise and is also an easier manufacturing task, however backward curved vanes are used in the impeller of the Rolls Royse GEM to reduce the risk of surge.

The Diffuser: To make the maximum use of the external effect, the high velocities at the impeller circumference are diffused substantially down to velocities approximately equal to the inlet axial velocity (say 150 m/s) as the combustion chamber requires a low inlet velocity. To achieve this diffusion in a short length, the air is divided into a number of separate streams by vanes which lead into diverging passages. To avoid separation in the adverse pressure gradient, experiments have shown that the maximum included angle in the divergence should be 11^0

Sealing: Impeller sealing has been a problem characteristic of the CFC. In the past, mechanical limitations have necessitated a finite clearance between the impeller vanes and the compressor hardware. This allowed a leakage of air between the vanes and degraded the compressor performance. One solution to this problem was the shrouded impeller, a method still used for some smaller engines, e.g. the Hercules Gas turbine compressor. However weight penalties are incurred and the impeller is difficult to manufacture. A more recent solution to the problem has been to cover the inside of the compressor casing with an abradable coating and allow the impeller to wear itself out. This method is used on the RR Gem HPC.

Axial Flow Compressors:

Introduction: The function of the compressor in a Gas Turbine Engine (GTE) is to raise the pressure of air flowing into the combustion chamber so that burning occurs at higher combustion efficiency.

$$\text{as } \pi \uparrow \;\rightarrow\; \eta \uparrow \;\rightarrow\; \downarrow SFC$$

The compression ratio is 3:1 on earliest engines, to 27: on the largest types. The second function of the compressor is to draw air into the engine during static operation on the ground.

Description and Operation:

The basic components of an AFC are alternate rows of rotating and stationary aerodynamically shaped blades. The rotating blades are attached to a central rotor assembly or spool which is driven by the turbine. The stationary blades are attached to the compressor outer casing to form the stator. The rotor row increases enthalpy of air by increasing its kinetic energy [the external effect] with some internal diffusion effect. The stator row redirects the flow to the next rotor row.

Rotor row + stator row = one stage

Putting many stages together form an axial compressor depending on π required. The engine could be of one spool, double or triple spool.

Construction:

The Blades: each rotor row consists of a number of blades attached to the central rotor assembly. The blades are of aerofoil section and are twisted to maintain constant angle of attach as blade speed increases from root to tip. The blades are constructed from Al alloys unless π_c raises the temperature high enough to dictate the use of Titanium alloys or steel towards the rear of the compressor.

The Rotor: The rotor assembly may be of the drum, disc, or combination.

The Stators: The stator blades also of aerofoil section are fitted into grooves cut in the compressor casing, and are secured to prevent their rotating around the casing. Some longer blades may be shrouded to minimize vibration due to flow variations and reduce tip leakage.

Sealing: It is important to minimize tip leakage to achieve high mass flow rates, pressure ratios and efficiencies; this is done by sealing at the rotor and stator tips.

Casing Design: Compressor casings used to be made of two halves (upper and lower) which allowed easy access for blade inspection. Modern design uses full ring construction or modular construction, in which borscope inspection ports are necessary.

Blade Terminology:

Base Profile: The basic profile of compressor blades is a symmetrical aerodynamic shape. The profile is specified in terms of thickness to chord length ratio of various points along the blade; the maximum thickness being (t).

Camber: The blades are curved around a camber line which is typically a circular or parabolic arc. The camber is specified in two ways; either the fraction of chord (a/c) from the leading edge to the point of maximum chord (b) or by the camber angle (Θ).

Stagger: When the blades are installed in compressors row, they are inclined to the axial direction by the stagger angle (ζ). The stagger angle is measured from the chord line to the axial direction. The angle of the camber line from the axial direction at the leading edge is (γ_0) and at the trailing edge is (γ_1).

The air flow into a blade row will meet the blades at an angle of incidence (i) which is the difference between the air angle (α) and the blade angle (γ). At the exit from the blades row, the air will deviate from the blade angle by deviation (∂), If (i) and (∂) are equal to zero, then the air would be deflected by (Θ), otherwise the deflection (ϵ) is:

$$\epsilon = \alpha_1 - \alpha_2$$

$$= (I + \gamma_0) - (\partial + \gamma_1)$$

but: $\Theta = \gamma_o - \gamma_1$

Then $\epsilon = \Theta + i - \partial$

Blade height (h) is the length from root to tip.

Aspect Ratio = h/c

Velocity Triangles: 2-D Flow

V_1 ..absolute velocity at α_o

of the air from previous stator

row entering the rotor row.

The velocity of the air relative

to the rotor is V_{1R} at α_1.

$V_{1R} = V_1 – U_1$

V_1 and V_{1R} may be resolved

into axial V_a and whirl velocities

V_{w1} and V_{w1R}

$V_{w1} = V_{a1} \tan \alpha_0$

$V_{w1R} = V_{a1} \tan \alpha_1$

The velocity of the air leaving rotor

relative to the rotor V_{2R} at α_2.

The absolute vale of this velocity

entering the stator is V_2 at α_3

$V_2 = V_{2R} + U_2$

V_{2R} and V_2 may be resolved into

V_{a2}, V_{w2R} and V_{w2}

$V_{w2} = V_{a2} \tan \alpha_3$

$V_{w2R} = V_{a2} \tan \alpha_2$

velocity V_3 at α_4 is the absolute

velocity of the air leaving the stator.

In passing through the rotor the relative velocity has decreased from V_{R1} to V_{R2} (some internal diffusion has occurred hence the static pressure will have increased). In passing through the rotor the absolute velocity has increased from V_1 (or V_3) to V_2 so the flow has been accelerated (i.e. an external effect). In passing through the stator the absolute velocity has decreased from V_2 to V_3 as the static pressure increases, but the total pressure remains constant in the stator and only rises in the rotor. Each stator row of blades acts as a diffuse for converting kinetic energy of the air leaving the rotor into pressure energy and to guide the air at the correct angle for the next rotor.

Basic Theory:

Consider a single stage of compression in which the flow is a 2-D thus no radial effects. U is constant, axial velocity is constant. $\dot{m} = \rho V_a A$

As the pressure and hence density increases, if the area decreases in inverse proportion then V_a can be held constant throughout the compressor.

Geometrical Considerations: From the velocity triangles

$U = V_{w1} + V_{w1R}$ and $U = V_{w2R} + V_{w2}$

$U = V_a (\tan \alpha_0 + \tan \alpha_1) = V_a (\tan \alpha_2 + \tan \alpha_3)$

$\tan \alpha_0 + \tan \alpha_1 = \tan \alpha_2 + \tan \alpha_3$ ………………………………………….1

$\tan \alpha_1 - \tan \alpha_2 = \tan \alpha_3 - \tan \alpha_0$ ………………………………………….2

Euler Pump and Turbine Equation: applying Euler equation to the rotor

$W = \dot{m}\,(UV_{w2} - UV_{w1})$ ………………………………………………………….3

$= \dot{m}\,UV_a\,(\tan \alpha_3 - \tan \alpha_0)$ ……………………………………………….4

Using equation 2

$W = \dot{m}\,UV_a\,(\tan \alpha_3 - \tan \alpha_0)$ ……………………………………………….5

Temperature Rise: applying the SFEE to the rotor

$Ht = Q - W$

$W = \dot{m}\,C_p\,(T_{t3} - T_{t0})$ ……………………………………,,………………6

and to the stator

$0 = \dot{m}\,C_p\,(T_{t4} - T_{t3})$ since W and Q are both zero in the stator.

Let $T_{tstage} = T_{t3} - T_{t0}$

$T_{tstage} = W/(\dot{m}\,C_p)$

Substituting equations 4 and 5

$T_{tstage} = (UV_a/C_p)\,(\tan \alpha_3 - \tan \alpha_0)$ …………………………………7

or

$T_{tstage} = (UV_a/C_p)\,(\tan \alpha_1 - \tan \alpha_2)$ …………………………………8

Pressure Ratio:

Work Done Factor: Because of variations in the axial velocity profile across the annulus; due to wall friction, and leakage across the ends of both the rotor and stator blades, the work done in a stage is less than that estimated from the basic theory. An empirical correction factor called work done factor Ω is used to modify the previous equations:

Equation 4 becomes:

$W = \dot{m}\Omega\, UV_a (\tan \alpha_3 - \tan \alpha_0)$ ……………………………………………….12

Equation 7 becomes:

$T_{tstage} = (\Omega UV_a/C_p) (\tan \alpha_3 - \tan \alpha_0)$ ………………………………………..13

Equation 10 becomes:

stage number	1	2	3	4and thereafter
Ω	0.98	0.93	0.88	0.83

Degree of Reaction: ^0R

We have seen that the increase in static pressure occurs partly in the rotor(due to internal diffusion effect) and partly in stator (due to external effect). It is useful to have a measure of the division between these two effects. Such a measure is provided by the degree of reaction which is defined as:-

$$^{0}R = \frac{Energy\ transfer\ due\ to\ pressure\ rise\ in\ rotor}{Total\ Energy\ Transfer}$$

Which can be written as:

$$^{0}R = \frac{Internal\ Diffusion\ effect}{Internal\ diffusion\ effect + External\ Effect} \qquad 16$$

In terms of work done:

$$^{0}R = \frac{\dot{m}\,Cp\,(T_3 - T_0)}{\dot{m}\,Cp\,(T_{t3} - T_{t0})} \qquad 17$$

By Euler

$$^{0}R = \frac{V_{1R}^2 - V_{2R}^2}{2U\,(V_{w2} - V_{w1})}$$

Using velocity triangles:

$^{0}R = (V_a/2U)\,(\tan\alpha_1 + \tan\alpha_2)$ ……………………………….19

$= 1 - (V_a/2U)\,(\tan\alpha_3 + \tan\alpha_0)$ …………………………..20

$$= \frac{V_{w1R} + V_{w2R}}{2U}$$

a) 100% Reaction:

If the ^{0}R is 1.0 then there is no external effect and all pressure rise occurs in the rotor. The stator passages are of constant area and serves only to redirect the flow. For this case V_{1R} and V_{2R} are very high. This type is uncommon.

0% Reaction: The rotor passages are of constant area while the stator passages act as diffusers. This type is not used.

b) 50% Reaction: If the degree of reaction is (0.5) then half the pressure rise is due to the internal diffusion effect in the rotor half due to the external effect realized in the stator. From equation number 21:

$\frac{V_{1R} + V_{w2R}}{2U}$ = 1/2 then U = $V_{w1R} + V_{w2R}$

$\alpha_1 = \alpha_3$ and $\alpha_0 = \alpha_2$

The velocity triangles are therefore symmetrical and the blades are mirror images of each other. This type is used in most gas turbine engines.

Losses:

a. Primary loss: If a blade row is required to produce large deflections or extensive diffusion in a short distance then high losses occur and the possibility of breakaway or separation of the flow.

Loss coefficient = $\frac{P_1 - P_2}{\frac{1}{2}\rho V^2}$

b) Secondary loss:

Deterioration in the performance of an axial compressor stage can be related to these losses. Physical damage to blades and also dirt may cause local changes in incidence, deflection and tip clearance; these will result in an increase in the loss coefficient leading to:

1. Reduced air flow rate and pressure ratio, hence reduced thrust or Shp output.

2. Blade stalling and compressor surge.

3. Drop in efficiency of compressor, hence reduced thermal and overall efficiency and increased SFC.

Transonic Compressors:

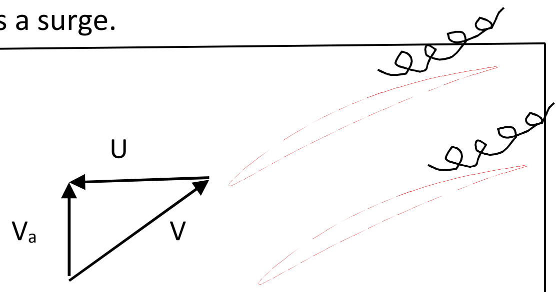
With the development of large engines, the mass flow rates have increased. This resulted in very large diameter compressors and fans with blades of large aspect ratios. It has been necessary to develop transonic compressors which have subsonic flow at the rotor hub and supersonic flow at the rotor tips. At the present rotor tip speed of M1.5 are used. One of the advantages of transonic stages is that IGVs can be dispensed, since for the same axial velocity the absence of IGVs gives higher values of V_1.

Low speed Stall:

Axial flow compressors are normally designed so that at the design condition (i.e. 100%rpm), the axial velocity component at each stage is the same, the air mass flow in must be constant and equals air density by flow area by axial velocity component.
As the density increases, the flow area must be reduced to keep axial velocity constant.
Hence the blade height (flow area) is reduced down flow of the axial flow compressors.
However, at low rotational speeds , the mass flow reduction from the design value is more marketed than the reduction in rotational velocity, so that the axial velocity at inlet is low and the incidence is high and could be high enough to stall the 1st stage, see Fig.(1) . The stalled blades produce an aerodynamic pulsation which can be recognized by the operator as banging or rumbling sounds. This is then known as a surge.

Reduction in mass flow rate ⟹
reeducation in V_a

⇨ large change
in incidence => rotor blades stall.



Stall of many stages => compressor surge.

Fig.(1): Blade Stall

To avoid surge the running line (Fig.2) must not cross the surge line, this means the πc is or will be low , which in turn leads to poor cycle efficiency , high SFC and low thrust. This could result that it is not possible to accelerate the engine rapidly, having left insufficient over fuelling allowance.

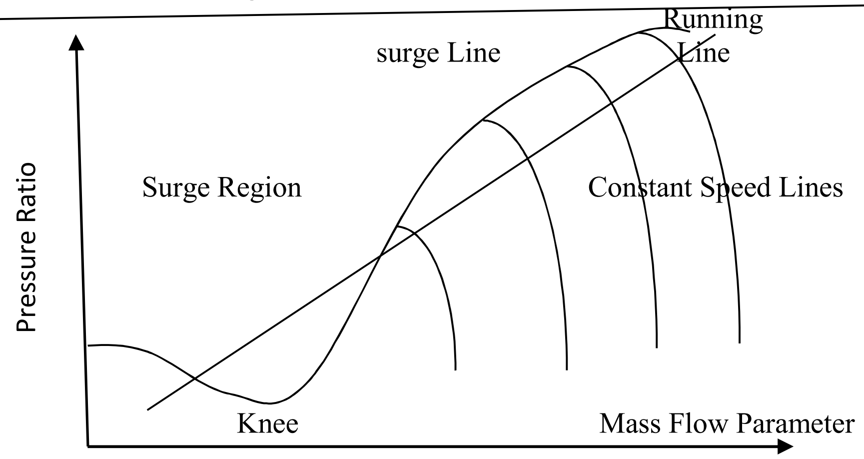


Fig. (2): Compressor Characteristics

To achieve acceptable pressure ratio it is necessary for the running line to cross the surge line which for a typical axial compressor is most likely to happen near the knee on the surge line (the knee is due to front end stall at low speeds).

Some means has to be found to prevent compressor surge at low speed, so that the engine may be operated over the full range of rotational speeds.

Stall Prevention:

To build in some form of air –Flow control , there are 4 possible methods[2]:

1- The dust characteristics may be changed by :
   1- variable area propulsion nozzle .
   2- Blow off values (Bleed Valves).
2- The compressor characteristics may be changed by :
   1- variable IGVS & stators .
   2- Multi-spooling.

Variable Area Propulsion Nozzle :

A nozzle determines the turbine running line and fixes the compressor running line Increase of nozzle area displaces the running line away from the surge line

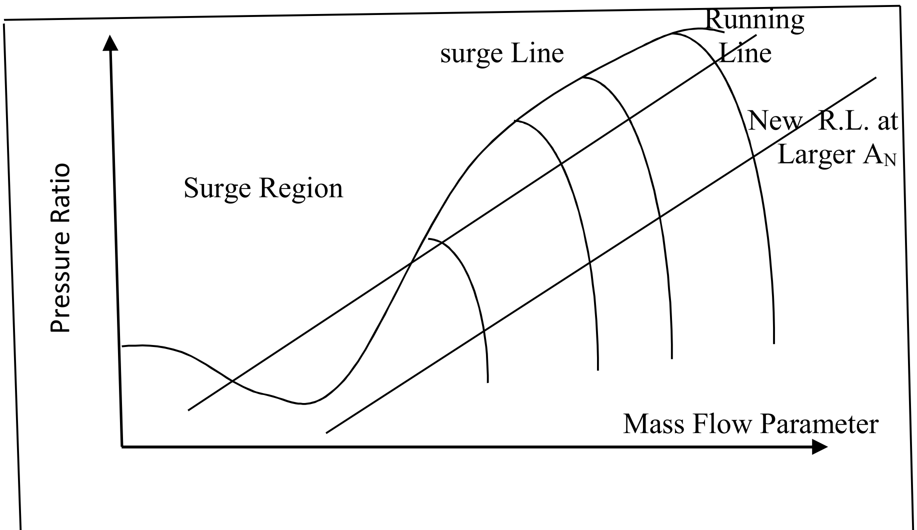


Fig.(3): Use of Variable Area Nozzle

As the nozzle area increases, see Fig.(3), the air mass flow increase too and hence increases the axial velocity component (V_a) which will correct the angle of incidence hence, prevent stall. Variable nozzles are fitted to re-heated engines but not always used for air flow control. The spey 202 (phantom) and Adour 102(Jaguar) are examples of variable nozzle used for air flow control.

Blow Off Valves (Bleed Valves).

BOVs which bleed off air from an intermediate stage of the compressor alter the duct characteristics of the engine and move the equilibrium running line away from the surge line .BOVs cause all inlet mass flow to be increased through the compressor stage prior to the BOVs. This increases the axial velocity component and cause improved incidence. BOVs are only opened at low speeds where air control is required as the bleed mass flow rate is a waste, see Fig.(4). Examples on BOVs is the pegasus11

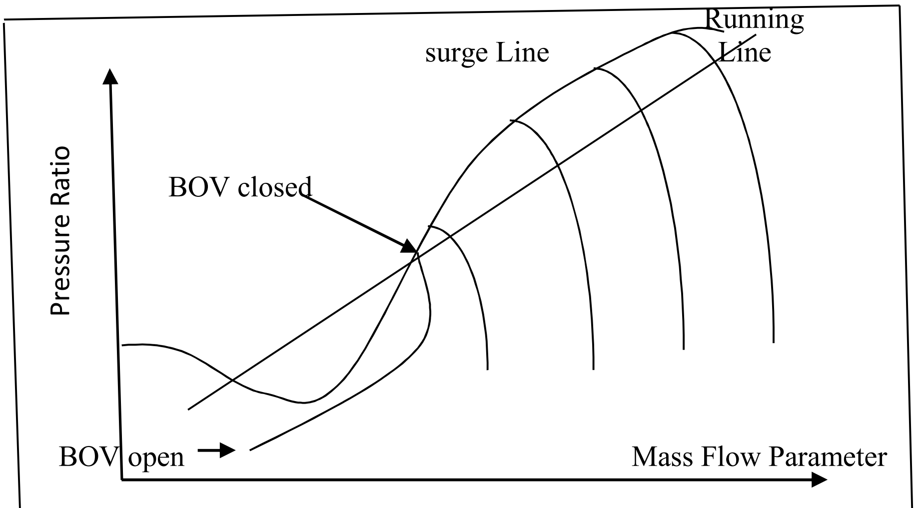


Fig.(4): Use of BOV

Variable IGVS and Stators

Front end stall at low speeds can be overcome if a set of variable inlet guide vanes (IVs) are positioned to direct the air on to the first row of moving blades at the correct incidence (i). As the axial velocity decreases the vanes must be repositioned to maintains the correct incidence, see Fig.(5).

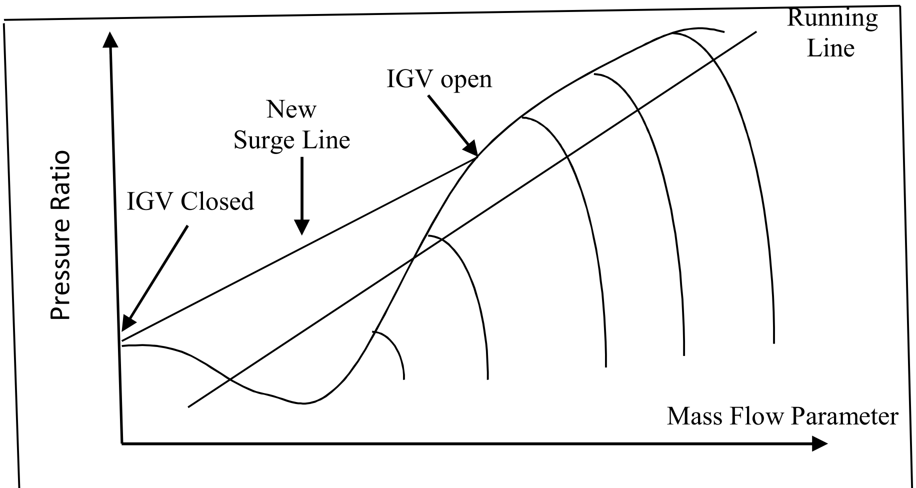


Fig.(5): Use of Variable IGV and Stators

Multi –Spooling:

The basic problem of low speed stall is that at low speeds the incidence of front stage is increased and that of the rear stage is reduced in comparison with design values. This is more noticeable in axial flow compressors of high pressure ratios. The problem can be solved by dividing the compressor into two or more parts and operating each part at different speed. The incidence can be restored by operating the front stages at reduced speeds and the rear at increased speeds . Examples of methods used to prevent compressor surge. Table (1) shows examples of methods used to prevent compressor surge.

Table (1): Examples of Methods Used to Prevent Compressor Surge [3].

Engine (Aircraft)	πc	No. of stage	No. of spools	Variable IGV	Variable Stators	BOV	Variable nozzle
Viper 200 (JP5)	4.37	7	1	No	No	No	No
Olympus301 (Vulcan)	10.5	6+7	2	No	No	No	No
Adour (jaguar)	9.6	2+5	2	No	No	Yes	No
Pegasus (harrier)	13.5	3+8	2	HP only	No	Yes	No
Spey202 (phantom)	20.5	5+12	2	HP only	No	Yes	Yes
Conway (VC10)	15.8	7+9	2	No	No	No	No
RB199 Tornado	23	3+3+6	3	No	No	Yes ip+hp	No
Turmo III C4 (Puma)	5.7	1centrifual 1 axial	1	No	No	No	No

Stability in a compressor is the ability of a compressor to recover from disturbances that alter the compressor operation about an operational equilibrium point. Disturbances may be considered as transient or deliberate changes to the operating point. In the case of transient disturbances, the system is stable if it returns to its original operating point. If the disturbances drive the compressor away from the original point, the system is unstable.