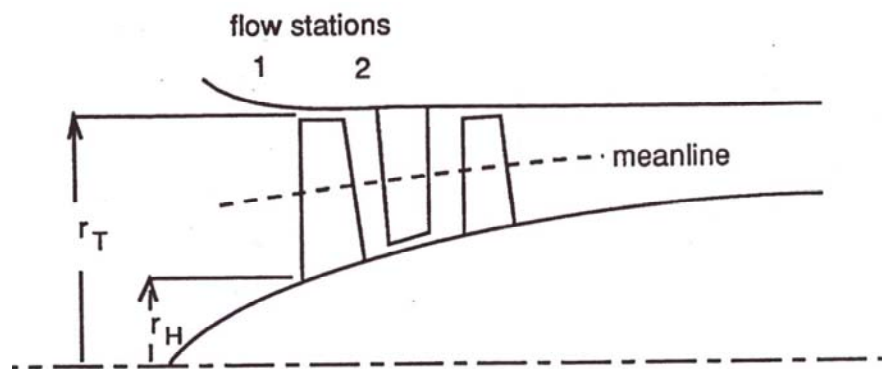


16.50 Lecture 26

Subjects: Compressor Blading; Design; Multi-Staging

Compressor blading; Radial variations

In all of the discussion so far the blade speed has been taken as a parameter. In fact the blade speed varies with the radial location in the compressor, so the “designs” that we have discussed are applicable only at one radius in any real compressor. In practice it is usual to begin a compressor design with such a treatment, called a “mean line design” carried out at some mean radius. But the effects of the blade speed variation with radius are important, so we must be aware of the constraints they impose on the design.



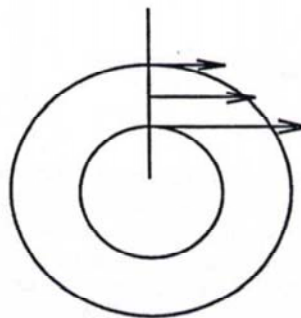
Normally it is desirable for the pressure (or temperature) ratio of a rotor blade row to be approximately constant over the radial length of the blade, so that the outlet airflow has uniform pressure. From the Euler equation

$$\frac{T_{t2}}{T_{t1}} - 1 = \frac{\omega(r_2 v_2 - r_1 v_1)}{c_p T_{t1}}$$

Supposing for simplicity that $v_1 = 0$ (no inlet guide vane), we see this condition requires

$$r_2 v_2 = \text{const.}; \quad v_2 = \frac{\text{const}}{r_2}$$

This implies that the rotor blade row should generate a Free Vortex in the flow



Now let us draw the velocity triangles for tip and hub, assuming $r_H/r_T = 1/2$, and $w_1 = w_2 = \text{const} = \omega r_T / 2$.

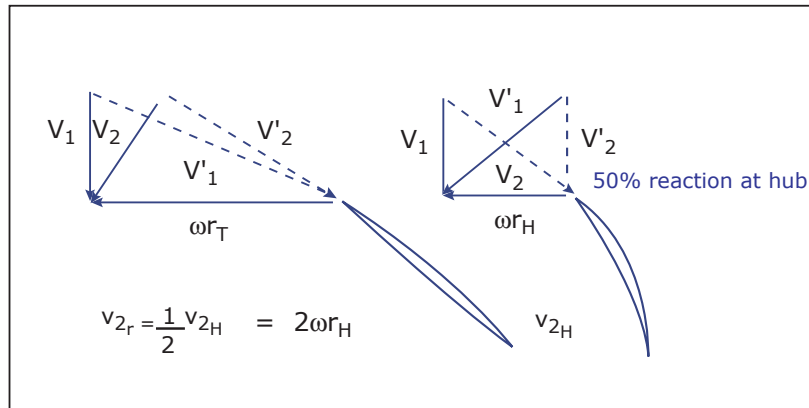


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Here it has been assumed that $v_2(r_H) = \omega r_H$, (that is that $\beta'_2 = 0$) and it follows that

$$v_2(r_T) = \omega r_H / 2 = \omega r_T / 4$$

The approximate blade shapes as sketched are determined by the condition that the leading and trailing edges are aligned with the flow. We see that the blades are strongly “twisted” from hub to tip.

Now let us see what these variations imply for the Diffusion Factor, D.

Taking $v_1 = 0$,

$$D = 1 - \frac{V'_2}{V'_1} + \frac{|v_2 - v_1|}{2\sigma V'_1}$$

Remember that these are relative to the rotor blade.

From the geometry,

TIP

$$|v_2 - v_1| = \omega r_T / 4$$

$$V'_1 = \omega r_T \sqrt{1 + 1/4} = 1.12 \omega r_T$$

$$V'_2 = \omega r_T \sqrt{\left(\frac{1}{2}\right)^2 + \left(\frac{3}{4}\right)^2} = .901 \omega r_T$$

$$D_T = 1 - \frac{.901}{1.12} + \frac{1/4}{2(1.12)\sigma_T}$$

HUB

$$|v_2 - v_1| = \omega r_H = \omega r_T / 2$$

$$V'_1 = \omega r_H \sqrt{2} = \omega r_T / \sqrt{2}$$

$$V'_2 = \omega r_H = \omega r_T / 2$$

$$D_H = 1 - \frac{1}{\sqrt{2}} + \frac{\frac{1}{2}}{(2/\sqrt{2})\sigma_H}$$

$$D_T = .195 + \frac{.116}{\sigma_T}$$

$$D_H = .293 + \frac{.353}{\sigma_H}$$

The solidities at hub and tip, σ_H and σ_T are of course design choices, but since the spacings S_H and S_T are related by

$$\frac{S_H}{S_T} = \frac{r_H}{r_T}$$

$$\frac{\sigma_H}{\sigma_T} = \frac{C_H}{C_T} \frac{S_T}{S_H} = \frac{C_H}{C_T} \frac{r_T}{r_H}$$

Let us choose $\sigma_T = 1$, $C_H/C_T = 1$, then $\sigma_H = 2$

$$D_T = .195 + .116 = .311$$

$$D_H = .293 + \frac{.353}{2} = .470$$

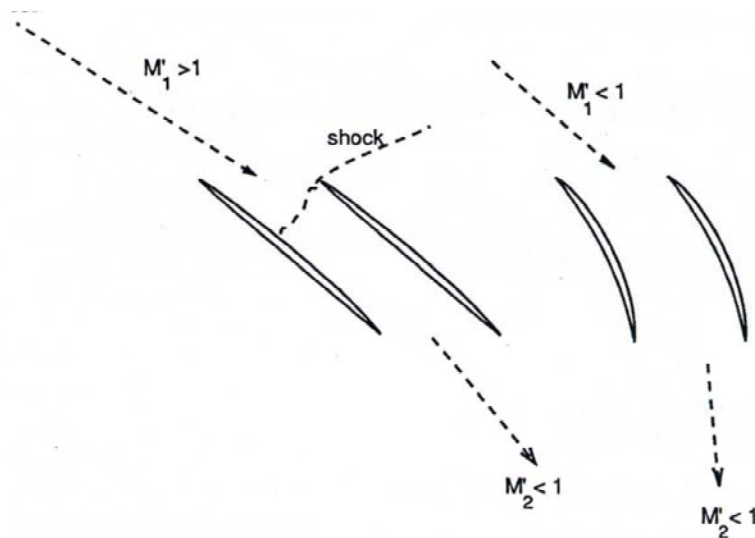
Both of these are acceptable from the viewpoint of losses, but D_H barely so. It is generally true that the hub section of the blade is limiting from the viewpoint of diffusion.

Mach Number Effects:

Typically the first stages of modern aircraft engine compressors operate with axial Mach number ≈ 0.7 and blade tangential Mach number $M_T \approx 1.3$ giving

$$(M_1)_T = \sqrt{M_1^2 + M_T^2} = 1.48$$

so the tips of the blades see supersonic flow. For $r_H/r_T = 0.5$, $(M_1)_H = \sqrt{.49 + .42} = .95$, so the hub is at a very high subsonic speed. Just as for inlets, hub and tip therefore require quite different diffusion techniques. At the hub, blading that is the equivalent of a subsonic inlet is called for



At the tip, the blades must be designed to minimize shock losses.

Multi-staging

Because ρ increases, the flow area must decrease as we go through the compressor. We can choose to effect this area variation by decreasing the tip radius, by increasing the hub radius, or something in between, like keeping a constant mean radius. All these solutions have both advantages and disadvantages.

The constant outer diameter design maintains the blade velocity at the maximum for the entire length of the compressor, hence gives the highest pressure ratio for a given number of stages. But because the length of the blades becomes small at the exit end of the compressor, leakage through the tip clearance is most serious for this design choice.

The constant inner diameter design minimizes the tip clearance problem and also yields lower stresses in the discs of the last stages, but requires more stages for a given pressure ratio.

The Polytropic Efficiency

What is most nearly constant among stages of a multi-stage compressor is the limiting isentropic efficiency for small compression, which is called the Polytropic efficiency:

$$\eta_{poly} \equiv \frac{(dh_t)_s}{dh_t} = \frac{\frac{1}{\rho_t} dp_t}{c_p dT_t} = \frac{RT_t d \ln p_t}{c_p dT_t} = \frac{\gamma - 1}{\gamma} \frac{d \ln p_t}{d \ln T_t}$$

If we assume η_{poly} remains constant along the compression path, this definition can be integrated to

$$\frac{T_{t,out}}{T_{t,in}} = \left(\frac{P_{t,out}}{P_{t,in}} \right)^{\frac{\gamma-1}{\eta_{poly}}}$$

or $\tau_c = \pi_c^{\frac{\gamma-1}{\eta_{poly}}}$. Inserting this into the definition of the normal Isentropic Efficiency allows one to compute it in terms of η_{poly} and the finite pressure or temperature ratio:

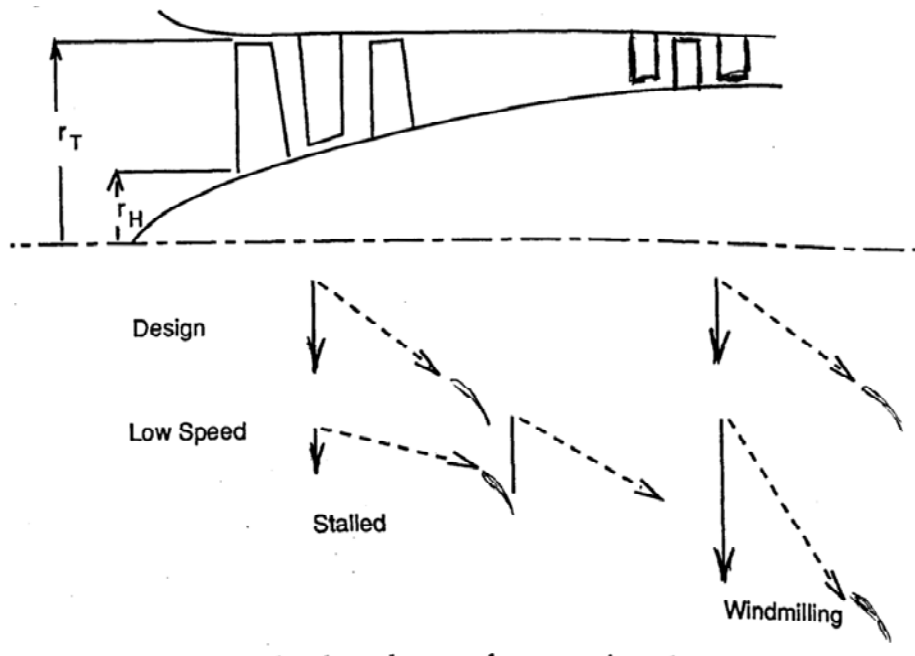
$$\eta_c \equiv \frac{\pi_c^{\frac{\gamma-1}{\gamma}} - 1}{\tau_c - 1} = \frac{\pi_c^{\frac{\gamma-1}{\gamma}} - 1}{\pi_c^{\frac{\gamma-1}{\eta_{poly}}} - 1}$$

As an example, take a multi-stage compressor with an overall pressure ratio $\pi_c=16$ and assume the polytropic efficiency is constant and equal to 0.9 in all stages. Its isentropic efficiency (ideal work required divided by actual work) is then calculated to be $\eta_c=0.856$. This is less than η_{poly} , the small-increment efficiency, because the last stages of the compressor receive gas that is hotter than it should, due to the inefficiencies in the previous stages, and hence more work is needed to compress it. It could be mentioned here that a similar argument can be made for turbines, and in that case one finds $\eta_{poly} > \eta_c$, because the extra thermal energy due to inefficiencies of the early stages is now available for conversion to work by the latter stages.

One warning: the isentropic efficiency of a compressor with N stage, each with a stage efficiency η_s (close to η_{poly}) is not η_s^N .

Starting and Low-Speed Operation

Because the density variation through the compressor is much less at low speed conditions, the compressor develops adverse blade loading situations in both the inlet and the outlet stages at low operating speeds. As shown in the sketch, the axial velocity in the inlet stages tends to be lower, and that in the outlet stages higher, relative to the blade speed, than at design, so the front stages tend to be stalled and the rear ones to “windmill”. This makes it difficult to design a compressor that will operate satisfactorily over a wide range of speeds.



The solution to this problem has taken two forms: one is to divide the compressor into two or even three “spools”, operating at different speeds, the other is to use stator blades of variable angle, to adjust the flow direction at low speeds. Most modern high pressure ratio engines use both.

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