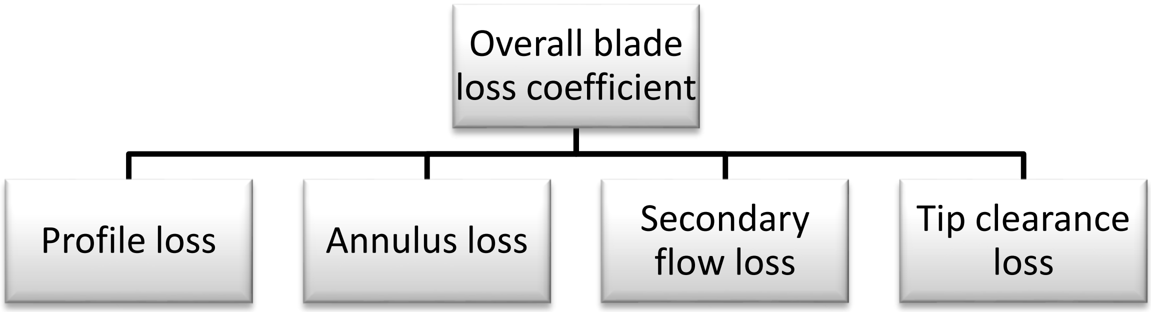
Choice of blade profile, pitch and chord

The choice of stator and rotor blade shapes, which will accept the gas incident upon the leading edge, and deflect the gas through the required angle with the minimum loss. An overall blade loss coefficient Y must account for the following sources of friction loss. We are going to choose the stator and rotor blade angles which will accept the gas incidence upon L.E and deflect the gas through the required angle with the minimum loss.

Losses



$$\lambda_{\text{Total}} = \overset{\text{Measured in cascade}}{\lambda_{\text{Profile}}} + \overbrace{\lambda_{\text{Annulus}} + \lambda_{\text{Secondary flow loss}}}^{\text{Grouped into one term secondary loss } \lambda_s} + \lambda_{\text{Tip clearance Loss}}$$

Profile loss

It is associated with boundary layer growth over the blade profile, which includes the separation loss under adverse conditions of extreme angles of incidence or high inlet Mach number.

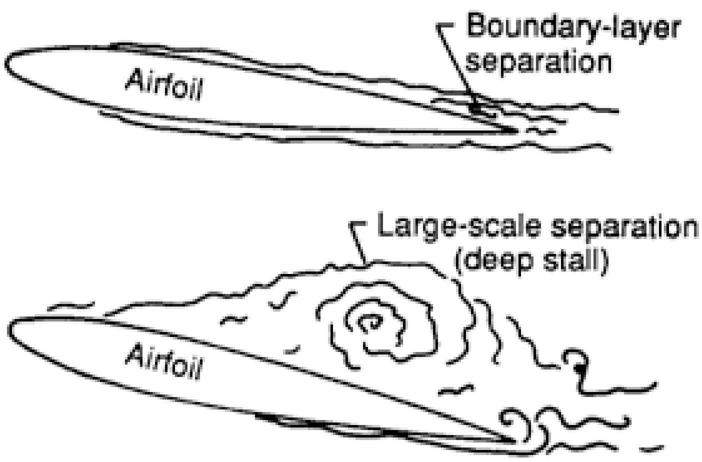


Fig 1.8 Boundary layer growth over blade profile

Annulus loss

It is associated with boundary layer growth on the inner and outer walls of the annulus.

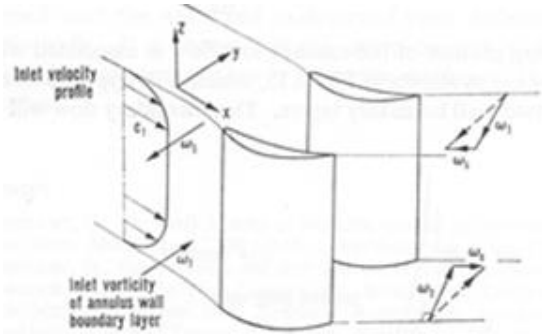


Fig 1.9 Annulus loss

Secondary flow loss

This loss arising from secondary flows which are always present when a wall boundary layer is turned through an angle by an adjacent curved surface.

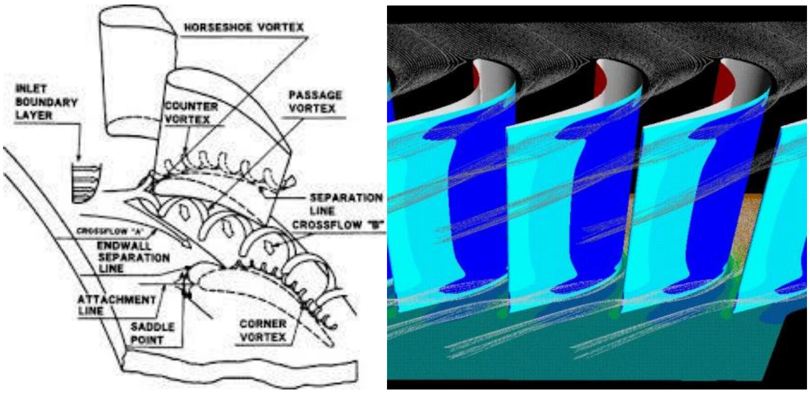


Fig 1.10 Secondary flow loss

Tip clearance loss

Near the rotor blade tip the gas does not follow the intended path, fails to contribute its quota of work output, and interacts with the outer wall boundary layer.

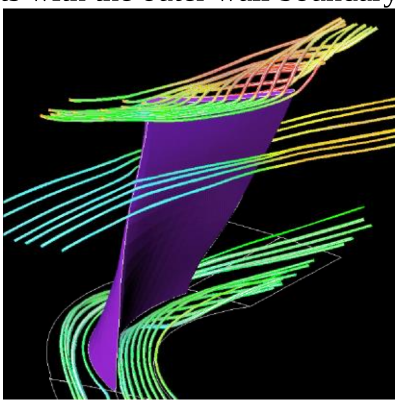


Fig 1.11 Tip clearance loss

The profile loss coefficient Y_p is measured directly in cascade tests. Losses annulus and secondary losses cannot easily be separated, and they are accounted for by a secondary loss coefficient Y_S. The tip clearance loss coefficient, which normally arises only for rotor blades, will be denoted by Y_k. Thus the total loss coefficient Y comprises the accurately measured two-dimensional loss Y_p plus the three-dimensional loss (Y_S + Y_k) which must be deduced from turbine stage test results.

Choice of Blade profile

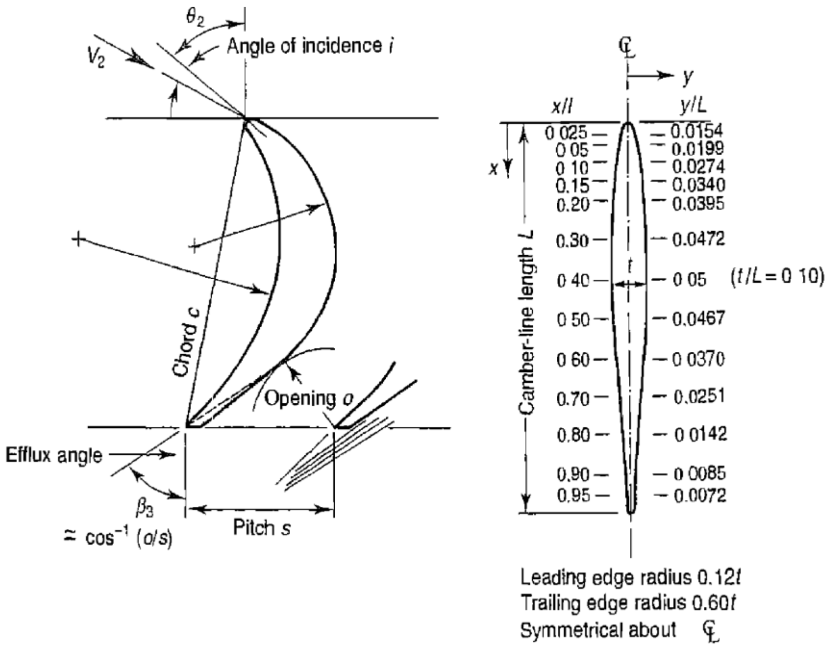


Figure 1.12 conventional steam turbine blade profile

Figure 1.12 shows a conventional steam turbine blade profile constructed from circular arcs and straight lines. Gas turbines have until recently used profiles closely resembling this, although specified by aerofoil terminology. One example is shown as the T6 base profile which is symmetrical about the centre line, It has a thickness/chord ratio (t/c) of 0·1, a leading edge radius of 12 percent t and a trailing edge radius of 6 percent t. When scaled up to a t/c of 0·2 and used in conjunction with a parabolic camber line having the point of maximum camber a distance of about 40 percent c from the leading edge, the T6 profile leads to a blade section similar to that shown but with a radiused trailing edge. In particular, the back of the blade after the throat is virtually straight. Other shapes used in British practice have been RAF 27 and C7 base profiles on both circular and parabolic arc camber lines. All such blading may be referred to as conventional blading.

It is important to remember that the velocity triangles yield the gas angles, not the blade angles. Typical cascade results showing the effect of incidence on the profile loss coefficient Y_p for impulse ($\Delta = 0\, and\, \beta_2 = \beta_3$) and reaction type blading are given in Fig. 1.13. Evidently, with reaction blading the angle of incidence can vary from -15° to + 15° without increase in Yp. The picture is not very different even when three-dimensional losses are taken into account. This means that a rotor blade could be designed to have an inlet angle θ_2 equal to say ($\beta_{2r} - 5^\circ$) at the root and ($\beta_{2t} + 10^\circ$) at the tip to reduce the twist required by a vortex design. It must be remembered, however, that a substantial margin of safe incidence range must be left to cope with part-load operating conditions of pressure ratio, mass flow and rotational speed.

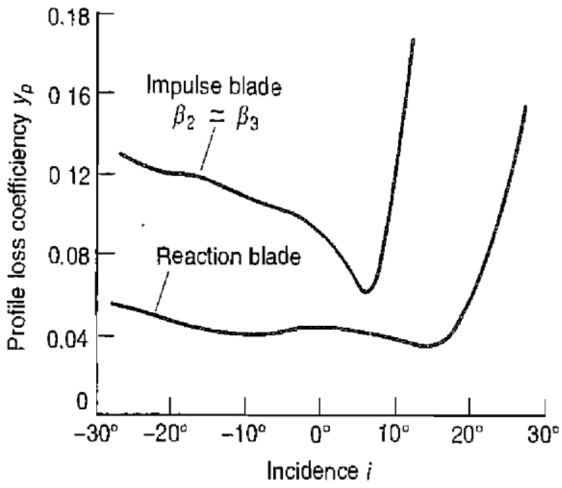


Fig 1.13 effect of incidence upon Y_P

With regard to the outlet angle, it has been common steam turbine practice to take the gas angle as being equal to the blade angle defined by $\cos^{-1}$(opening/pitch). This takes account of the bending of the flow as it fills up the narrow space in the wake of the trailing edge; there is no 'deviation' in the sense of that obtained with decelerating flow in a compressor cascade. Tests on gas turbine cascades have shown, however, that the $\cos^{-1}$(o/s) rule is an overcorrection for blades of small outlet angle operating with low gas velocities, i.e for some rotor blades. Figure 1.14 shows the relation between the relative gas outlet angle, β_3 say, and the blade angle defined by $\cos^{-1}$(o/s). The relation does not seem to be affected by incidence within the working range of ±15 degrees. This curve is applicable to 'straight-backed' conventional blades operating with a relative outlet Mach number below 0.5. With a Mach number of unity the $\cos^{-1}$(o/s) rule is good for all outlet angles, and at Mach numbers intermediate between 0.5 and 1.0 it can be assumed that $[\cos^{-1}(o/s) - \beta_3]$ varies linearly with Mach number.

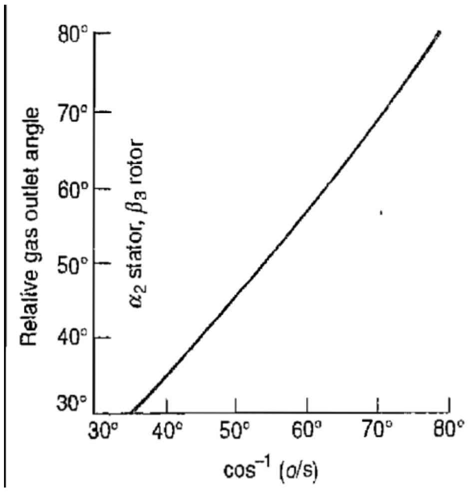


Fig 1.14 Relation between gas and blade outlet angle

Note that until the pitch and chord have been established it is not possible to draw a blade section to scale, determine the 'opening', and proceed by trial and error to make adjustments until the required gas outlet angle α_2 or β_3 is obtained. Furthermore, this process must be carried out at a number of radii from root to tip to specify the shape of the blade as a whole. Now the pitch and chord have to be chosen with due regard to

(a) The effect of the pitch/chord ratio (s/c) on the blade loss coefficient,

(b) The effect of chord upon the aspect ratio (h/c), remembering that h has already been determined,

(c) The effect of rotor blade chord on the blade stresses, and

(d) The effect of rotor blade pitch upon the stresses at the point of attachment of the blades to the turbine disc.

Optimum pitch to chord ratio (s/c)

Fig.1.15 shows the optimum pitch to chord ratio for different gas angles and blade angles. These curves suggest, as might be expected, that the greater the gas deflection required [$(\alpha_1 + \alpha_2)$ for a stator blade and $\beta_2 + \beta_3$ for a rotor blade] the smaller must be the 'optimum's/c ratio to control the gas adequately. The adjective 'optimum' is in inverted commas because it is an optimum with respect to Y_P not to the overall loss Y. The true optimum value of s/c could be found only by making a detailed estimate of stage performance for several stage designs differing in s/c but otherwise similar. In fact the sIc value is not very critical.

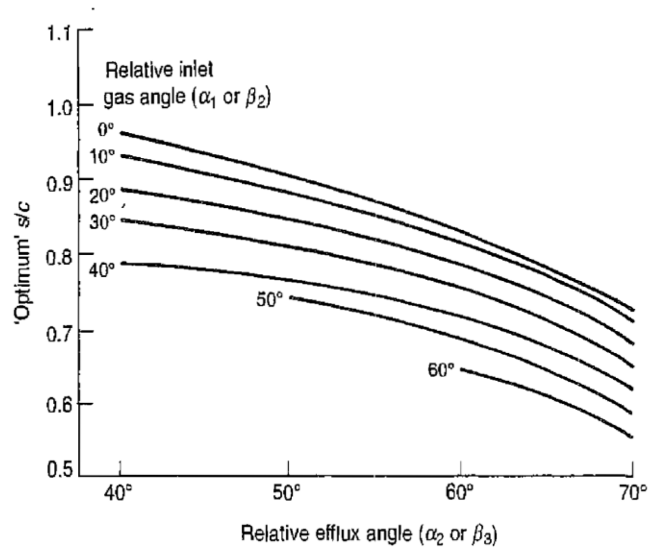


Fig 1.15 Optimum pitch to chord ratio(s/c)

Consider the nozzle gas angles and rotor blade angles as mentioned below

$$\alpha_{1m} = 0^\circ, \alpha_{2m} = 58.38^\circ, \beta_{2m} = 20.49^\circ, \beta_{3m} = 54.96^\circ$$

The optimum pitch to chord ratio for the above considered angles can be found by using fig1.15 by the correlating $\alpha_{1m}\ with\, \alpha_{2m}$ for nozzle and $\beta_{2m}\ with\, \beta_{3m}$ for rotor.

$$(s/c)_N = 0.86$$

$$(s/c)_R = 0.83$$

Choice of Aspect ratio (h/c)

The influence of aspect ratio is open to estimation, but for our purpose it is sufficient to note that, although not critical, too low a value is likely to lead to secondary flow and tip clearance effects occupying an unduly large proportion of the blade height and so increasing Ys for the nozzle row and (Y_S + Y_k) for the rotor row. On the other hand, too high a value of h/c will increase the likelihood of vibration trouble: vibration characteristics are difficult to predict and they depend on the damping 'provided by the method of attaching the blades to the turbine disc. A value of h/c between 3 and 4 would certainly be very satisfactory, and it would be unwise to use a value below 1.

Consider a gas turbine with the mean height of the nozzle and rotor blades are given by

$$h_N = \frac{1}{2}(h_1 + h_2) = \frac{1}{2}(0.046 + 0.0612) = 0.0536\,\text{m}$$

$$h_R = \frac{1}{2}(h_2 + h_3) = \frac{1}{2}(0.0612 + 0.077) = 0.0691\,\text{m}$$

The chord of the nozzle and rotor can be obtained by adopting aspect ratio (h/c) of 3.

$$\left(\frac{h}{C}\right)_N = 3$$

$$C_N = \frac{h_N}{3} = \frac{0.0536}{3} = 0.0175m$$

$$\left(\frac{h}{C}\right)_R = 3$$

$$C_R = \frac{h_R}{3} = \frac{0.0691}{3} = 0.023m$$

Using these values of chord, in conjunction with the chosen s/c values, gives the blade pitches at the mean radius of 0.216 m as

$$(\text{s/ c})_\text{N} = 0.86$$

$$s_N = 0.86 \times C_N = 0.86 \times 0.0175 = 0.01506m$$

$$(\text{s/ c})_\text{R} = 0.83$$

$$s_R = 0.83 \times C_R = 0.83 \times 0.023 = 0.0191m$$

The number of blades can be calculated by using the pitch of the nozzle and rotor blades

$$n_N = \frac{2\pi r_m}{s_N} = \frac{2\pi(0.216)}{0.01506} = 90$$

$$n_R = \frac{2\pi r_m}{s_R} = \frac{2\pi(0.216)}{0.0191} = 71$$

It is usual to avoid numbers with common multiples to reduce the probability of introducing resonant forcing frequencies. A common practice is to use an even number for the nozzle blades and a prime number for the rotor blades. As it happens the foregoing numbers are satisfactory and there is no need to modify them and re-evaluate the pitch s.