Loss and deviation model for a compressor blade element

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A procedure for calculating loss and deviation for a compressor blade element, based on published empirical relationships, is presented. Allowances are made for the position of the blade element along the span and for Mach number effects, so that the method is suitable for use in the design of transonic compressors. The method is based on the work of Swan, Jansen and Moffatt, Davis and Millar, SP-36, Hirsch[4], and Davis. The predictions of the model are compared with experimental results given by Krabacher and Gostelow, SP-36, and Kovach and Sandercock.

Keywords: *compressor blades, pressure loss, deviation angle, transonic compressors*

The basis for most of the work correlating losses and deviations with a diffusion factor is that of Lieblein[3], where he developed a simple analytical relation between the blade-wake momentum thickness and the blade surface-velocity diffusion for conventional cascade blades. He then presented experimental evidence to show that as blade camber or blade angle of attack is increased, large velocity gradients occur on the blade suction surface, but comparatively small changes occur on the pressure surface. Thus, changes in total wake momentum thickness will result primarily from the diffusion contribution of the suction-surface boundary layer.

The magnitude of the velocity diffusion in low speed flow generally depends on the blade geometry of the blade section and its incidence angle. As the Mach number is increased, however[5], compressibility exerts a further influence on the velocity diffusion of a given cascade geometry and orientation. If local supersonic velocities develop at high inlet Mach numbers, the velocity diffusion is altered by the formation of shockwaves and the interaction of these shockwaves with the blade surface boundary layers.

Cascade inlet Mach number also influences the magnitude of the subsonic diffusion for a fixed cascade. This Mach number effect[5] is the conventional effect of compressibility on the blade velocity distributions in subsonic flow. Compressibility causes the maximum local velocity on the blade surface to increase at a faster rate than the inlet and outlet velocities. Accordingly, the magnitude of the surface diffusion from maximum velocity to outlet velocity becomes greater as the inlet Mach number is increased. The principal factors upon which to base an empirical cascade wake momentum-defect thickness correlation may be summarized as follows[5]:

(a) velocity diffusion on the suction surface;
(b) inlet Mach number;
(c) blade-chord Reynolds number;
(d) turbulence level (if possible).

In a compressor blade element the losses and deviations are also influenced, in addition to the above parameters, by three-dimensional effects such as secondary flows, changes in axial velocity across the blade element, radial flows etc, and hence the position of the blade element along the span will influence its performance.

This work presents a loss and deviation model suitable for transonic compressors by bringing together published empirical relationships that are diffusion factor based. The model is then tested against published experimental data and the results reported.

An important advantage of using diffusion correlated loss and deviation over incidence-parameter correlated loss and deviation is that the diffusion factor (using any of the definitions available) is a function, among other parameters, of B_2 (the outlet angle) and hence the deviation angle predicted will exert an influence on the loss, thus the deviation angle has to be calculated first before the loss may be evaluated. In incidence-related models, deviation and loss are calculated independently of each other, which is unrealistic.

Empirical relationships and correlations

These relationships for losses and deviations for the design and off-design performance of compressor blade elements have been obtained from published papers. The system of relationships is built from different publications with minor modifications and simplifying assumptions incorporated where possible.

Evaluation of optimum incidence (i^*)

Before any assessment of losses and deviations can be made the incidence at minimum loss, i^*, has to be evaluated. There are a number of correlations on which to base the evaluation of this parameter. Among the most commonly used are SP-36[5], Mellor's chart and,

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for double circular arc profiles, the geometry of the blades. It is found that the use of SP-36 gives results which are nearer to experimental results than the other two methods, and so this is the method adopted in this work.

SP-36 correlations

The details of this method are given in SP-36[5]. The optimum incidence is given in terms of i_0^* which is the optimum incidence for a zero cambered aerofoil (NACA 65 series) of 10% maximum thickness to chord ratio as follows:

$$i_{2D}^* = K_{sh} K_t i_0^* + n\phi$$

where

$K_{sh} = 0.7$ for DCA

$= 1$ for 65-series

$= 1.1$ for C- and DCA profiles

$K_t = f(t/c)$

$i_0^* = f(\mu_1, S/c)$

$n = f(B_1^*, S/c)$

These functions are given numerically by Davis[7]. The equation for n, however, should be that for $(-n)$.

Another correction incorporated into the program is that due to the way the incidence is varied (ie constant B_1 or constant γ). SP-36 suggests a reduction of 1° to 2° in the value of i_{2D}^* (as calculated above) when the stagger is kept constant and the incidence is varied by changing the gas angle B_1. A correction of 2° is incorporated in this program.

A correction for three-dimensional effects is then made to allow for factors influencing the flow in a real compressor as opposed to linear cascades. These corrections are based on the experimental curves given in SP-36 and given numerically by Davis[7] as follows:

for 65-series and C-series blade profiles

$$i_c = i_{2D}^* + (1.8 - 0.048\bar{R})$$

and for DCA

$$i_c = i_{2D}^* + \begin{pmatrix} -2.5 + 10(M_1') & 20 \leqslant \bar{R} \leqslant 80 \\ -2.5 + 6.6(M_1') & \bar{R} > 80 \\ -2.5 + 14(M_1') & \bar{R} < 20 \end{pmatrix}$$

where $\bar{R}$ is the percentage blade height at which the blade element is situated.

Evaluation of deviation (δ^*) at $i = i^*$

Carter's rule is used to evaluate δ^* as follows:

$$\delta^* = m_c \frac{\phi}{\sqrt{\sigma}} \tag{1}$$

where m_c is a function of blade stagger and its values are given by Davis[7] as follows:

(a) for 65-series blade profile, m_c is given by

$$m_c = 0.126 + (1.823 \times 10^{-3})\gamma + (2.14 \times 10^{-5})\gamma^2$$

(b) for C-series and double circular arc profiles m_c is given by

$$m_c = 0.216 + (9.72 \times 10^{-4})\gamma + (2.38 \times 10^{-5})\gamma^2$$

The value of δ^* given by Eq (1) is corrected for the effect of inlet Mach number as outlined below.

First the critical inlet Mach number has to be determined. When the Mach number reaches unity locally in an airfoil cascade, the corresponding inlet Mach number is said to have reached its critical value. The assumption is made by Jansen and Moffatt[2] that, below

Notation

B	Gas angle
c	Blade chord
D_{eq}	Equivalent diffusion factor defined by Eq (6)
D_L	Loading diffusion factor defined by Eq (7)
D	Diffusion factor defined by Eq (8)
i	Incidence
M	Mach number
P	Pressure
ΔP_{tot}	Loss in total pressure relative to the row at a particular radius
R	Radius measured from machine centre line
$\bar{R}$	Percentage blade height along span of blade element, $(1 - \{(R_t - R)/(R_t - R_h)\}) \times 100$
S	Blade pitch
t	Blade thickness
(t/c)	Maximum thickness to chord ratio
V	Velocity relative to blade row
V_z	Axial velocity
γ	Stagger angle or ratio of specific heats
δ	Gas deviation angle
μ	Blade angle
σ	Solidity, c/S
θ/c	Wake momentum-defect thickness to chord ratio
ϕ	Camber angle
$\bar{\omega}$	Mass averaged total pressure loss coefficient, $\Delta P_{tot}/(P_{tot} - P)$ (profile loss only). Relative values are used for rotors

Subscripts

c	Compressor element or critical value
t, h	Tip and hub, respectively
M	Including Mach number effect
2D	Two-dimensional value (linear cascade element)
s	Stator
r	Rotor
1	Inlet to blade element
2	Outlet from blade element, except as defined in Fig 1
max	Maximum
o	Stagnation

Superscripts

*	At $i = i^*$ (incidence at which loss is minimum)
‾	Mass averaged value
′	Relative to blade row

the critical Mach number, the total pressure losses and turning angles are essentially constant. Beyond this value the pressure losses increase and the turning decreases rapidly. Jansen and Moffatt evaluate the critical inlet Mach number as follows. Experimental work on NACA 65 series shows that:

$$\frac{V_{\max}}{V_1}=K_1+K_2\left(\frac{\Delta V_\theta}{\sigma V_1}\right) \tag{2}$$

For compressible flow

$$\frac{\Delta V_\theta}{\sigma V_1}=\frac{\cos B_1}{\sigma}\left(\tan B_1-\frac{V_{a2}}{V_{a1}}\tan B_2\right)$$

and K_1, K_2 are given as function of thickness to chord ratio and given by the above reference as

$$K_1=1.03+0.7(t/c)$$

$$K_2=0.4+(t/c)$$

The relationship between the critical inlet Mach number and $V_{\max}/V_1$ is given by

$$\left(\frac{V_{\max}}{V_1}\right)^2-1=\frac{1-\left(\frac{2}{\gamma+1}+\frac{\gamma-1}{\gamma+1}M_{1c}^2\right)^{\gamma/(\gamma-1)}}{\left(1+\frac{\gamma-1}{2}M_{1c}^2\right)^{\gamma/(\gamma-1)}-1} \tag{3}$$

Jansen and Moffatt then suggest that the above equation may be solved iteratively for M_{1c}.

However, a close examination of the above equation shows that M_{1c} can be explicitly given by

$$M_{1c}=C_3\left(\left[(V_{\max}/V_1)^2\Big/\left(C_2-C_2\left(\frac{V_{\max}}{V_1}\right)^2+C_1\right)\right]^{(\gamma-1)/(\gamma-2)}\right)^{1/2} \tag{4}$$

where

$$C_1=(\gamma+1)^{\gamma/(1-\gamma)}$$

$$C_2=2^{\gamma/(1-\gamma)}$$

$$C_3=(\gamma-1)^{-1/2}$$

A correction is then applied to δ^* as given by Eq (1) if the inlet Mach number is higher than the critical, this is given by Roland and Millar[1] as

$$\delta_M^*-\delta^*=8(M_1'-M_{1c}) \tag{5}$$

when the inlet flow is supersonic, the value of the exit Mach number from the leading edge shock is used for M_1' as suggested by Ref 1.

Evaluation of diffusion factors

These diffusion factors are based on the ratio of the maximum suction surface velocity to the inlet or exit velocity. The three factors most commonly used are those due to Roland and Millar[1], SP-36[5] and Davis[7].

(a) The equivalent diffusion factor, D_{eq}, is defined by the relation:

$$D_{eq}=\frac{V_{\max}}{V_2}=\left(\frac{V_{\max}}{V_1}\right)\left(\frac{V_1}{V_2}\right) \tag{6}$$

(b) Diffusion factor, D_L, is defined by

$$D_L=\frac{V_{\max}-V_2}{V_{\max}}=1-\frac{V_2}{V_{\max}}$$

hence

$$D_L=1-(1/D_{eq}) \tag{7}$$

(c) A third definition may also be used, which may be termed simply diffusion factor, D, defined by

$$D=\frac{V_{\max}-V_2}{V_1}$$

which may be converted in terms of D_{eq} as

$$D=\frac{V_2}{V_1}(D_{eq}-1) \tag{8}$$

The significance of using diffusion factors as loading parameters is that they may be used to give a guide to the onset of separation on the suction surface of the blade.

If D_{eq} is used as a loading parameter, then Swan's work[8] shows that flow separation commences when D_{eq} reaches a value of (2.0 – 2.2), regardless of whether shocks are present or not. This value may also be used to warn that surge is imminent in a compressor. If D_L is used, then separation will commence[5] when D_L becomes greater than 0.5. If D is used, then separation commences at values of D greater than 0.6.

D_{eq}^* may now be evaluated using Eq (2) to give

$$D_{eq}^*=\left[K_1+\frac{K_2\cos B}{\sigma}\left(\tan B_1^*-\frac{V_{a2}}{V_{a1}}\tan B_2^*\right)\right]\times \\ \times\frac{V_{a1}\cos B_2^*}{V_{a2}\cos B_1^*} \tag{9}$$

The above definition of D_{eq}^* does not take into account the change in radius between inlet and outlet, ie it does not include the slope of the stream lines. In an axial machine, however, this effect is generally small and is therefore ignored in the present analysis. For a fuller definition of D_{eq} see Swan[8].

Evaluation of momentum thickness

The value of the wake momentum-defect thickness is given as a function of percentage blade height and D_{eq}, thus

$$\left(\frac{\theta}{c}\right)^*=f(D_{eq},\bar{R}) \tag{10}$$

The explicit form of this relationship is based on the experimental curve fit of data given by Swan[8]. These data are for double circular arc rotors; they include rotor elements which had been operated transonically, ie shocks in the passage. Swan suggested a method whereby he was able to subtract an estimated shock contribution from the total measured loss coefficient, to obtain an estimated loss coefficient represented only by the viscous and wake action. The explicit relation of Eq (10) is given by Davis[7] as

$$\left(\frac{\theta}{c}\right)^*=(-1.0312+0.01722\bar{R})+(1.396-0.0244\bar{R})D_{eq}^* \\ +(-0.467+0.00866\bar{R})D_{eq}^{*2} \tag{11}$$

The relationship between wake momentum thickness and profile losses

Two-dimensional profile cascade losses arise primarily[5] from the growth of the boundary layer on the suction and pressure side of the blade. Theoretical analysis by Lieblein[4] and SP-36[5] shows that the relationship between the wake momentum-defect thickness (θ/c) and the total pressure loss ($\bar{\omega}$) for conventional unstalled blade profiles is given approximately by:

$$\bar{\omega} \simeq 2\left(\frac{\theta}{c}\right)_2 \frac{\sigma}{\cos B_2}\left(\frac{\cos B_1}{\cos B_2}\right)^2 \tag{12}$$

Thus the value of $\bar{\omega}^*$ can be determined. The above equation becomes more doubtful at off-design conditions.

If the value of the inlet Mach number is higher than the critical value as calculated by Eq (4), then a correction suggested by Roland and Millar[1] is applied and is given by:

$$\bar{\omega}^* = \bar{\omega}^*[(2M_1' - M_{1c}) + 1] \tag{13}$$

Evaluation of off-design deviation

The first step is to evaluate an off-design D_{eq}. The method suggested by Swan[8] and given by Davis[7] is simply to add another term to the rhs of Eq (2), giving V_{max}/V_1 as follows:

$$\left(\frac{V_{max}}{V_1}\right)_{\text{off design}} = K_1 + K_2\left(\frac{\Delta V_\theta}{\sigma V_1}\right) + a(i - i^*)^{1.43} \tag{14}$$

where

$a = 0.0117$ for 65-series blade profile;

$a = 0.007$ for C-series and DCA profiles; hence D_{eq} is given by

$$D_{eq} = \left[K_1 + a(i-i^*)^{1.43} + \frac{K_2 \cos B_1}{\sigma} \times \left(\tan B_1 - \frac{V_{a2}}{V_{a1}}\tan B_2\right)\right]\frac{V_{a1}\cos B_2}{V_{a2}\cos B_1} \tag{15}$$

Swan[8] gives the equation for the off-design deviation as a function of ($D_{eq} - D_{eq}^*$) and inlet Mach number (at off-design the spanwise location has little effect). This equation is based on his experimental tests of a transonic double circular arc rotor. The equation is as follows:

$$\delta - \delta^* = [6.4 - 9.45(M_1' - 0.6)](D_{eq} - D_{eq}^*) \tag{16}$$

Since D_{eq} is a function of B_2, ie a function of δ, the above two equations have to be solved iteratively for the values of δ and D_{eq}.

Evaluation of off-design losses

Again the correlations used here are based on the work of Swan[8]. His experimental work has shown that, at off-design, the spanwise location of the element has little effect on the deviation or losses. Thus the wake momentum thickness is given as a function of ($D_{eq} - D_{eq}^*$) and the inlet Mach number only as follows:

for $D_{eq} > D_{eq}^*$

$$\left(\frac{\theta}{c}\right) - \left(\frac{\theta}{c}\right)^* = \{0.827M_1' - 2.692M_1'^2 + 2.675M_1'^3\}(D_{eq} - D_{eq}^*)^2$$

and for $D_{eq} < D_{eq}^*$

$$\left(\frac{\theta}{c}\right) - \left(\frac{\theta}{c}\right)^* = \{2.8M_1' - 8.71M_1'^2 + 9.36M_1'^3\}(D_{eq} - D_{eq}^*)^2 \tag{17}$$

The off-design loss ($\bar{\omega}$) may then be evaluated using Eq (12).

Evaluation of shock losses

An estimate of shock losses is necessary if the inlet Mach number is greater than unity. The model used in this work is that due to Swan[8], which is based on experimental observations (see Fig 1). The assumptions of this model are, first, the flow is assumed to approach the leading edge tangent to the upper surface. The blade section (made of two circular arcs) is then assumed to have a bow wave always attached to the leading edge followed by a Prandtl–Meyer expansion until the expansion wave intersects the neighbouring blade. A normal shock is assumed to occur across the channel at the minimum area (assumed at inlet to the blade passage). From geometry[8] we have

$$R_u = r + \frac{\left(\left[\frac{t}{2} - r\right] + \left[\frac{c}{2} - r\right]\left[\tan\left(\frac{\phi}{4}\right)\right]\right)^2 + \left(\frac{c}{2} - r\right)^2}{2\left(\left(\frac{c}{2} - r\right)\tan\left(\frac{\phi}{4}\right) + \left(\frac{t}{2} - r\right)\right)}$$

where R_u is the radius of curvature of the suction surface

r is the leading edge radius

ϕ is the blade chamber angle

$$\theta = \tan^{-1}\left(\frac{S\cos B_1}{S\sin B_1 + R_u}\right)$$

where θ is the strength of the Prandtl–Meyer expansion wave (see Fig 1).

The problem then is to find the maximum Mach number on the suction surface just before the shock which is assumed to occur at the point of minimum area. θ is normally small, thus for isentropic flow between 1 and 2 it may be shown[9] that

$$\Delta M = \frac{M_1\theta\left(1 + \left(\frac{\gamma - 1}{2}\right)M_1^2\right)}{(M_1^2 - 1)^{1/2}}$$

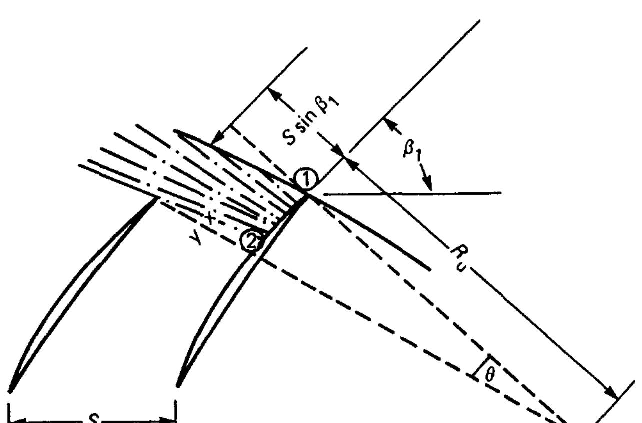

Fig 1 Swan's model[8]

where θ is in radians

$$M_2 = M_1 + \Delta M$$

Now using the assumption made by Swan[8]

$$M_x = (M_1 + M_2)/2$$

and from the one-dimensional normal shock relations we have

$$M_y = \left(\frac{M_x^2 + \dfrac{2}{\gamma - 1}}{\dfrac{2M_x^2}{\gamma - 1} - 1} \right)^{1/2}$$

and

$$\frac{P_{ty}}{P_{tx}} = \frac{\left(\dfrac{\dfrac{(\gamma+1) - M_x^2}{2}}{2 + (\gamma - 1)M_x^2} \right)^{\gamma/(\gamma-1)}}{\left(\dfrac{2\gamma M_x^2}{\gamma+1} - \dfrac{\gamma - 1}{\gamma + 1} \right)^{1/(\gamma-1)}}$$

The isentropic relationship gives

$$\frac{P_1}{P_{t1}} = \left[1 + \frac{(\gamma - 1)M_1^2}{2} \right]^{\gamma/(1-\gamma)}$$

Hence the shock contribution to the total losses is given by

$$\omega_s = \frac{1 - \dfrac{P_{ty}}{P_{tx}}}{1 - \dfrac{P_1}{P_{t1}}}$$

Comparison of the model predictions with experimental results

The aim of this work is to put together a loss and deviation model suitable for transonic compressors. The blade profiles used in such compressors are mainly double circular arc and the experimental data presented are for these profiles.

Three sources of experimental data are chosen for comparison; these are NASA 2D rotor[6], SP-36[5] and Kovach and Sandercock[10]. The findings are as follows.

NASA 2D rotor

This is a transonic rotor with double circular arc profile along the entire span. When the comparison for this rotor was made the model for the shock losses was not incorporated into the program and thus only the experimental results for rotor speeds that give inlet Mach numbers less than unity were chosen for the comparison (50% and 70% design). Blade elements at 50% immersion were also excluded from the comparison due to the effect of the part-span shroud. Before a meaningful comparison may be made there are a few points regarding the experimental details which are worthy of mention. The measuring station, obviously, is not exactly at the trailing edge and leading edge of each blade element. Krabacher and Gostelow[6], therefore, calculate the conditions at the trailing and leading edge of each blade element by first applying the condition of constant angular momentum along design stream lines to obtain the tangential velocity at each blade edge. The assumption is made that the shape of each meridional stream tube between the measuring station and its adjacent blade edge remains fixed at the design shape for all data conditions. This may lead to errors, especially at the blade trailing edge[6]. The values of the loss coefficient for the section $\bar{R} = 10$ (Fig 5) are extremely suspect as they are very small, and in fact two of the experimental points show a negative loss coefficient. Nevertheless, the experimental loss coefficients for all the blade elements show the variation of loss coefficient with incidence to be much flatter than the strong bucket type variation for linear cascade elements as reported in SP-36[5].

The experimental points for each section are presented for two different Mach numbers (representing the 50% and 70% design speeds). These Mach numbers are indicated on the figures. These comparisons are given in Figs 2 to 5. The loss coefficient comparisons seem to be good for the 70% section (Fig 3), while with the other sections the experimental loss coefficient seems to be on the low side, but this could very well be due to the way these experimental points were obtained as explained earlier. The prediction of the deviation, on the other hand, is excellent for all the sections considered.

Experimental results from SP-36

The experimental points of loss versus diffusion factor for a number of rotors (15–20) that have DCA profiles, for the hub and mid sections are reproduced in Fig 6. The drawn line shows the predictions from the model. These show

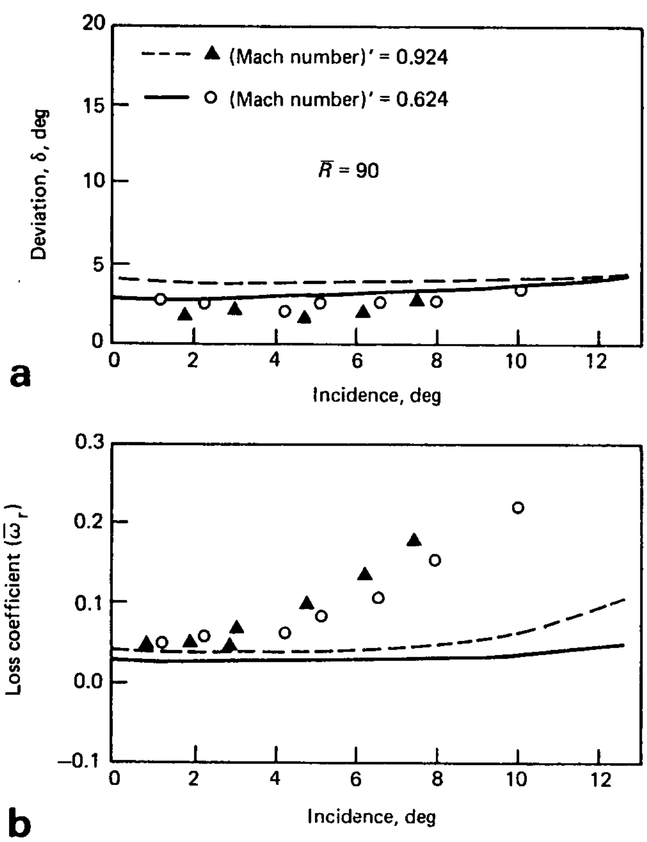

Fig 2 Data for NASA 2D rotor[6] for $\bar{R} = 90$: (a) deviation angle versus incidence angle; (b) loss coefficient versus incidence angle

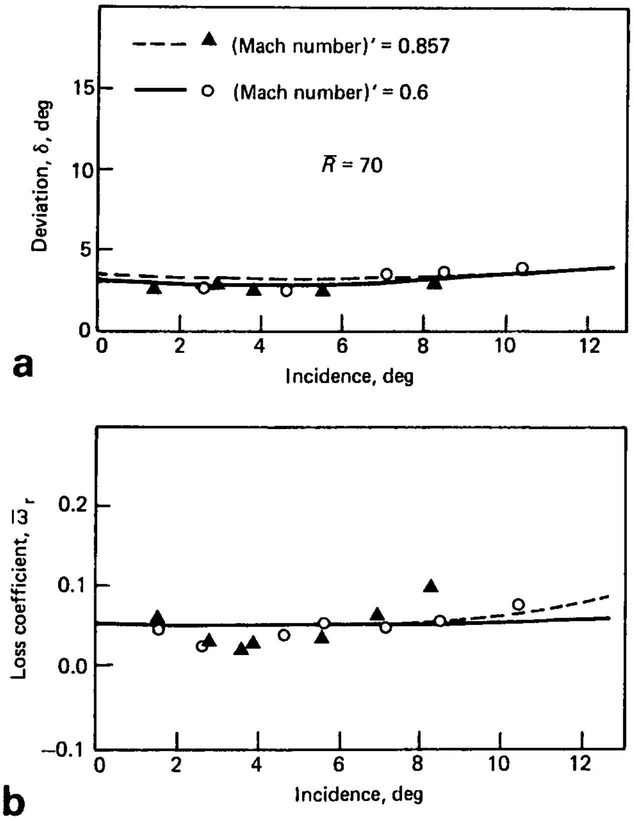

Fig 3 Data for NASA 2D rotor[6] for $\bar{R}=70$: (a) deviation angle versus incidence angle; (b) loss coefficient versus incidence angle

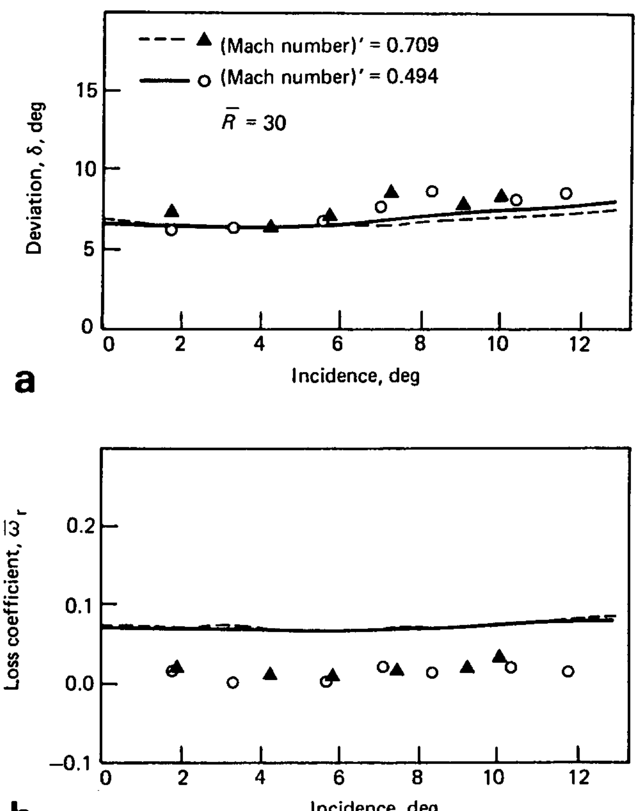

Fig 4 Data for NASA 2D rotor[6] for $\bar{R}=30$: (a) deviation angle versus incidence angle; (b) loss coefficient versus incidence angle

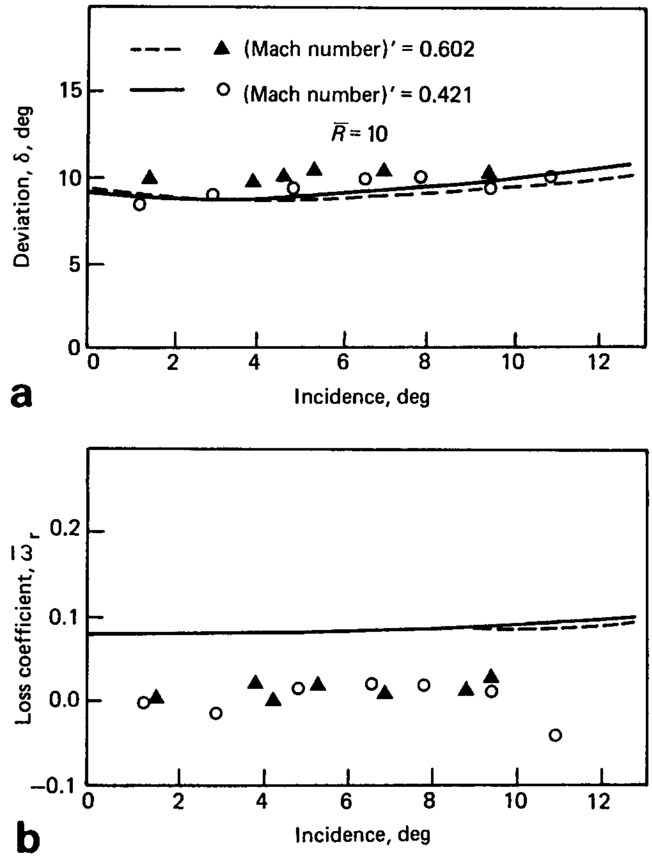

Fig 5 Data for NASA 2D rotor[6] for $\bar{R}=10$: (a) deviation angle versus incidence angle; (b) loss coefficient versus incidence angle

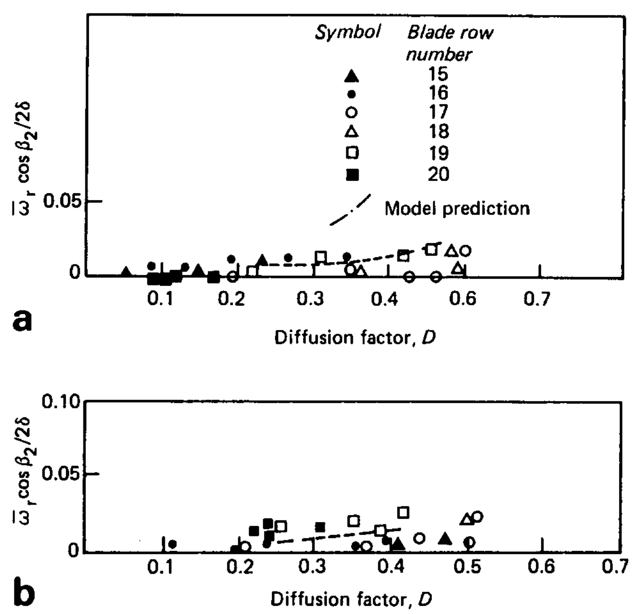

Fig 6 Data from NASA SP-36[5] for DCA blades: (a) hub section; (b) mean section

that the model represents a good mean estimate of the actual losses occurring in a real compressor blade element. (The experimental points shown in Fig 6 are those for inlet Mach numbers less than 1.)

In Fig 7, which represents the tip section for the rotors given in Fig 6, we have the opportunity of testing the shock losses model. The solid line shows the profile losses; it is clearly not a good estimate of what is actually happening. The chain dotted curve includes the shock losses as predicted by the model: clearly an improvement in the prediction of losses.

Karl Kovach and D. Sandercock

The experimental work presented in Ref 10 is for a 5-stage transonic compressor. The blade element data are presented at two different speeds (100% and 90% design) and for a number of sections along the span. All the rotors and the first two stators are made of DCA profiles.

The optimum incidence, i^*, is calculated using SP-36 3-D correlations, which seem to give the best estimate when compared with the experimental points. It is found, moreover, that the effect of i^* values on the estimation of deviation and losses is small.

The graphs for losses and deviations have been replotted as losses against rotor and stator number, and as deviation against rotor and stator number for each section along the blade span, and for the two speeds considered (90% design and 100% design) (Figs 8 to 13). Only two stators (stators 1 and 2) are considered in the comparison as these are the ones with double circular arc profiles. Study of the figures leads to the following observations.

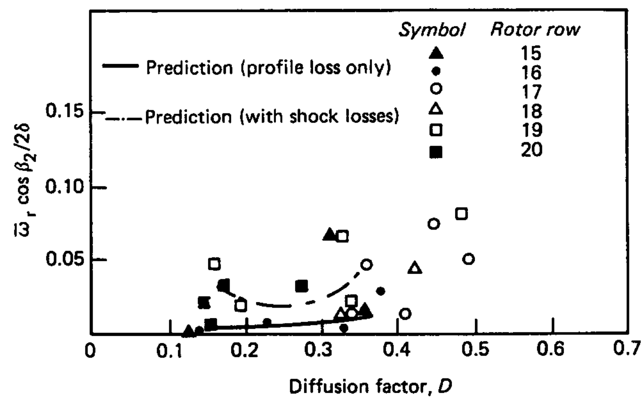

Fig 7 Data from NASA SP-36[5] for tip section

Rotors

(a) For rotor 1, the model predictions of loss seem to be higher than the experimental values for all the sections and both speeds apart from one value corresponding to section 84% from hub at 100% design speed where the theoretical predictions agree exactly.

(b) With rotor 2, the model underestimates the losses near the tip, overestimates near the hub and agrees quite well at mid sections.

(c) For rotor 5, the losses for the section near the tip seem to be underestimated by the model.

(d) For rotor 4, at 100% speed (Fig 8) the model seems to underestimate considerably the losses at all sections.

(e) The prediction of deviation, δ^*, is good for all the rotors but, once again, the predictions for the sections near the tip, for some of the rotors, are not as good.

Stators

The prediction of deviation is not as good as that for the rotors but it still represents quite a good estimate of the experimental deviation. The estimate of losses by the model is no better than that for the rotors, and the tip losses, once again, are underestimated.

Conclusions

It is very difficult to find a loss and deviation model that is capable of predicting these parameters accurately for any

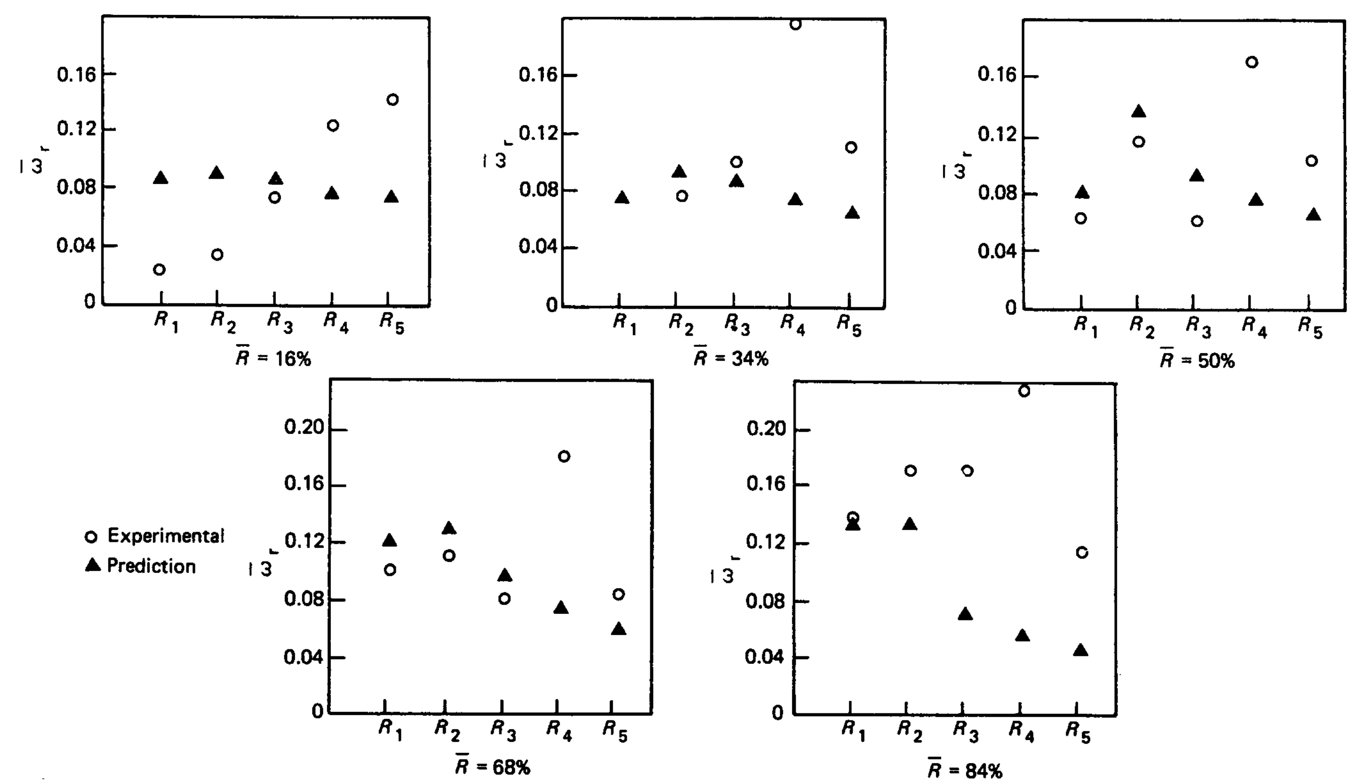

Fig 8 Data from Ref 10 for 100% design speed: loss coefficient versus rotor row number

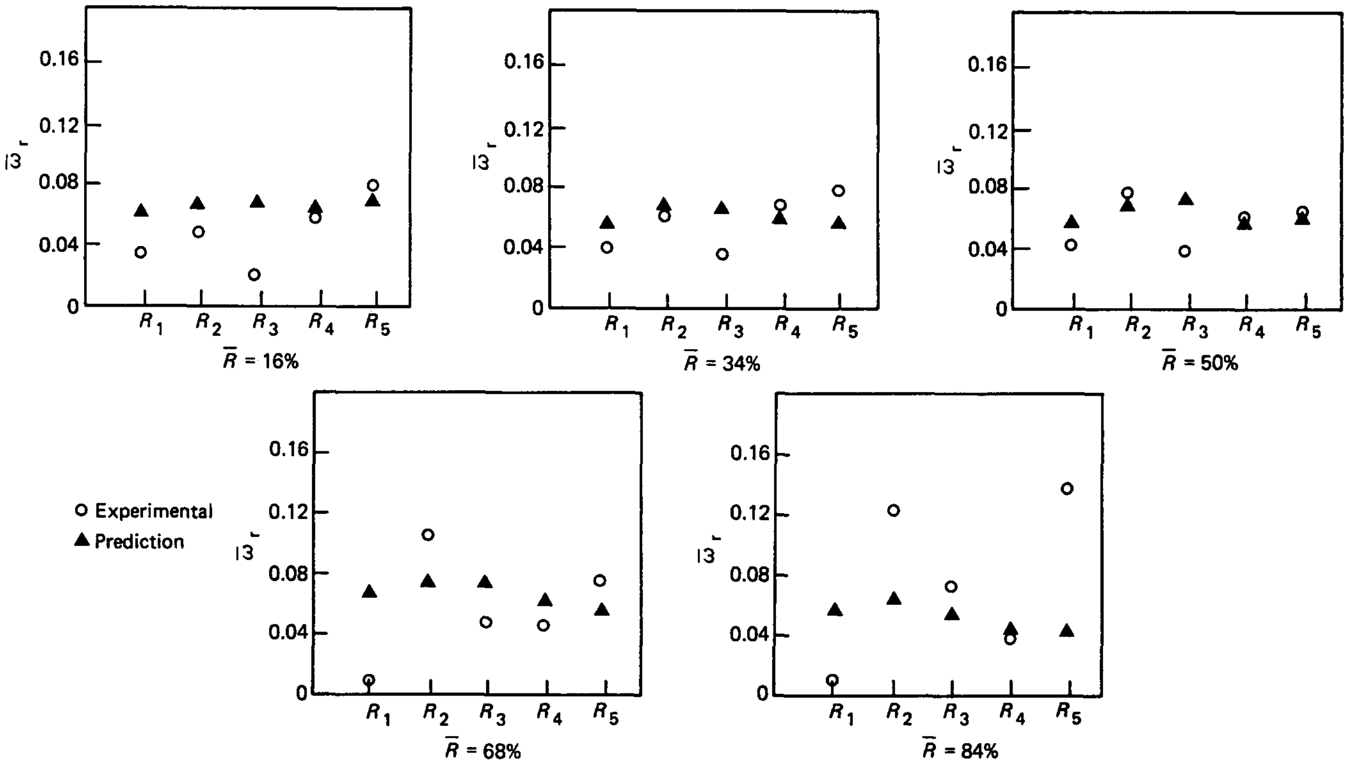

Fig 9 Data from Ref 10 for 90% design speed: loss coefficient versus rotor row number

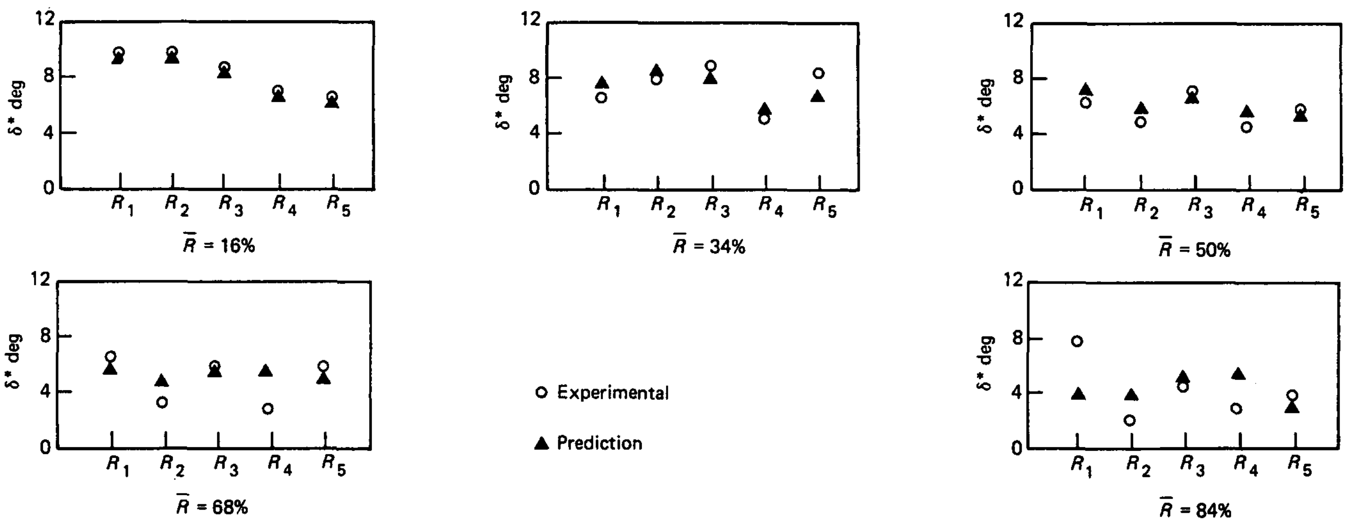

Fig 10 Data from Ref 10 for 100% design speed: deviation angle at minimum loss rotor row number

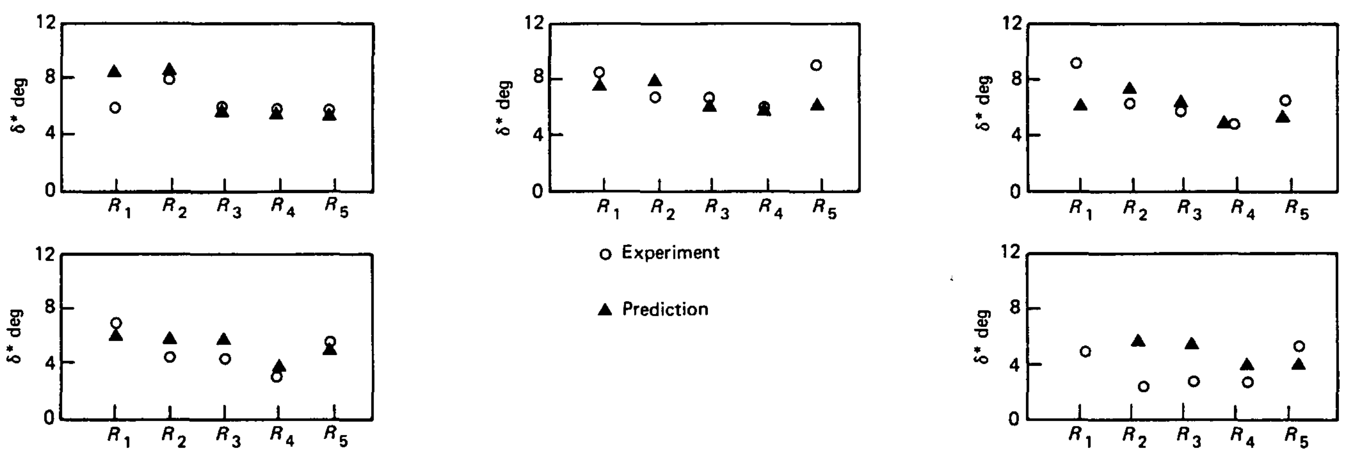

Fig 11 Data from Ref 10 for 90% design speed: deviation angle at minimum loss versus rotor row number

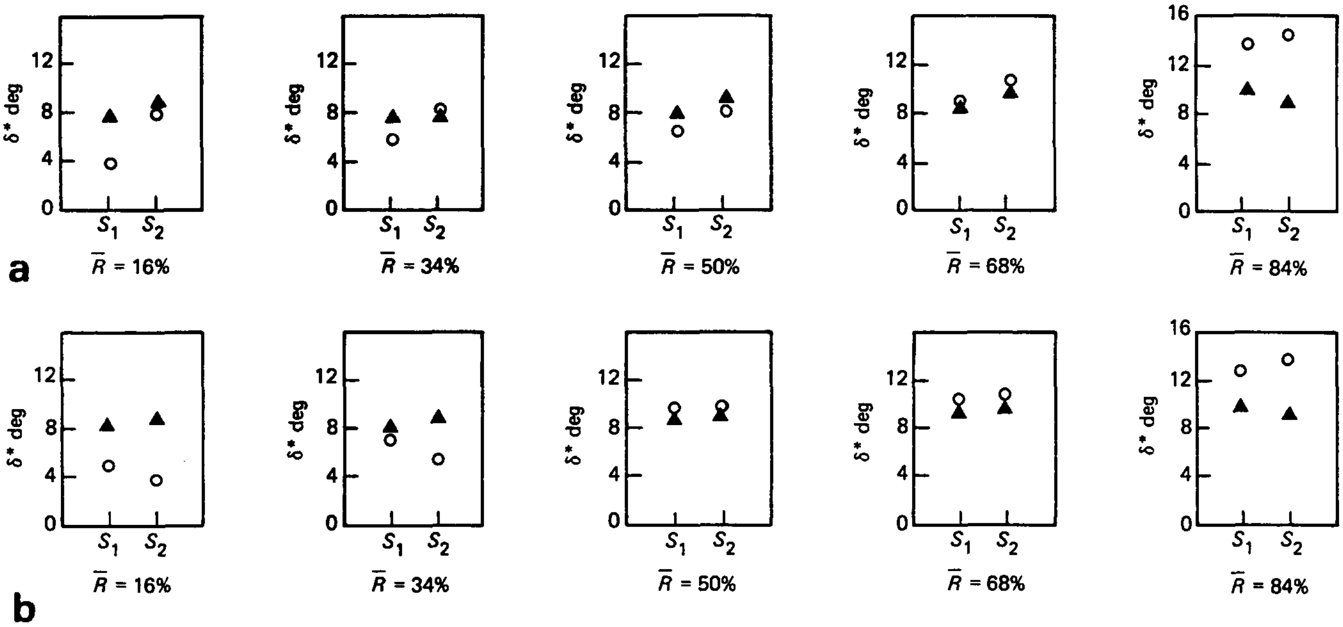

Fig 12 Data from Ref 10, deviation angle at minimum loss versus stator row number: (a) for 90% design speed; (b) for 100% design speed

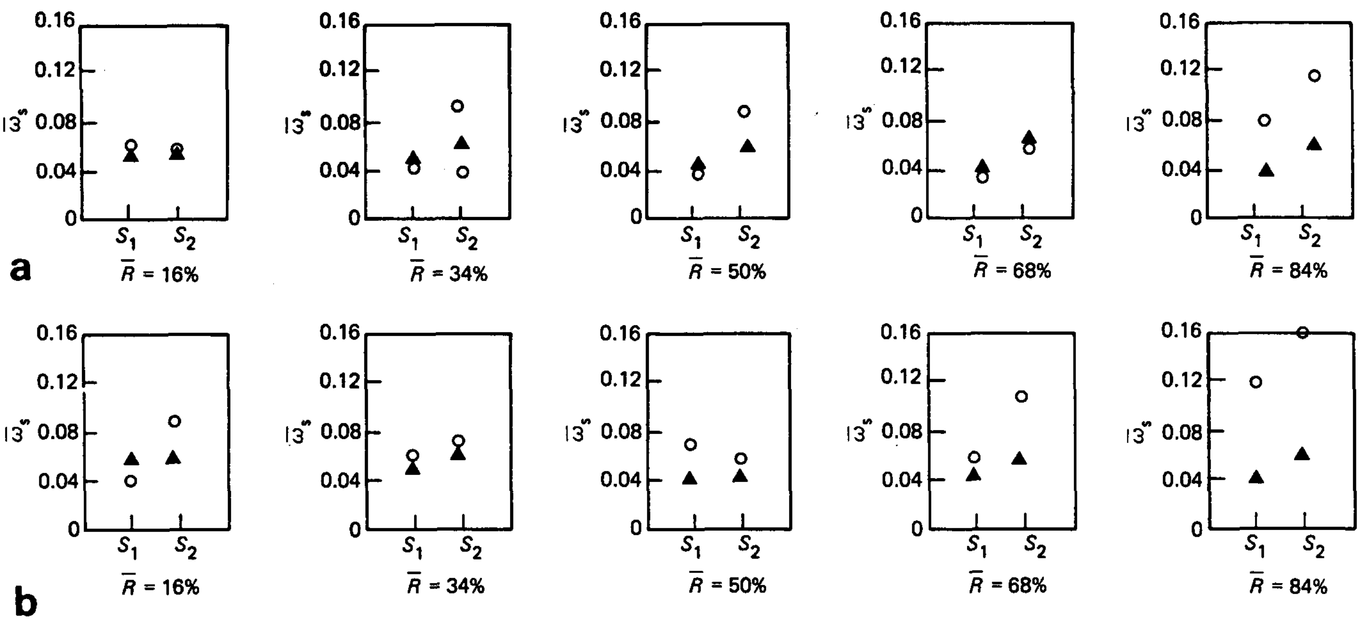

Fig 13 Data from Ref 10, loss coefficient versus stator row number: (a) for 90% design speed; (b) for 100% design speed

compressor, when one considers the complicated nature of the flow within a multistage axial compressor. There are many factors which would influence losses and deviations that various correlations have not even attempted to quantify among these are blade surface roughness, turbulence level, instrumentation errors, unavoidable observation errors when one is confronted with a reading that shows some fluctuations, and tip clearance effects which would have an influence on the entire flow (although the major effect is restricted to sections close to the tip).

The differences between experimental values of blade element performance and predicted values could be due to (a) normal experimental scatter, (b) experimental errors or (c) inadequacy of the prediction procedure. The method presented needs to be incorporated into a throughflow analysis prediction method such as Roland and Millar[1]. The blade element method would produce accurate predictions of overall performance if the differences were due to normal experimental scatter. Nevertheless, examination of the model presented leads to the following general conclusions:

(a) The estimation of deviation for all compressors considered in the comparison is good.
(b) The estimation of loss is poor in two respects. First, close to the tip, the model underestimates these losses considerably. More experimental data are needed to be able to modify the relationships involving blade section position along the blade. Second, the loss and deviation data of Ref 10 are replotted against row number (Figs 8–13). These show that the position of the row within the compressor greatly influences the loss, while its effect on the deviation is minor. A realistic model, therefore, should include this effect as a parameter in the loss empirical relationships.

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CALENDAR

AIAA Joint Fluid Dynamics and Heat Transfer Conference	12–14 May 1986 Atlanta, GA, USA	American Institute of Aeronautics and Astronautics, 1290 Avenue of the America, New York, NY 10019, USA
Water for Energy	14–16 May 1986 Brighton, UK	BHRA Fluid Engineering Centre, Cranfield, Bedford MK43 0AJ, UK
31st International Gas Turbine Conference and Exhibit	8–12 June 1986 Dusseldorf, FRG	ASME, 345 E. 47th Street, New York, NY 10017, USA
Flow Measurement in the Mid-80s	9–12 June 1986 Glasgow, UK	Conference Section, National Engineering Laboratory, East Kilbride, Glasgow G75 0QU, UK
6th International Symposium: Finite Element Methods in Flow Problems	16–20 June 1986 Antibes, France	INRIA, Service des Relations Extèrieures Rocquencourt, BP 105, 78153 Le Chesnay Cedex, France
8th International Heat Transfer Conference	17–22 August 1986 San Francisco, CA, USA	Chang-Lin Tien, Department of Mechanical Engineering, University of California, Berkeley, CA 94720, USA
7th International Fluid Power Symposium	16–18 September 1986 Bath, UK	BHRA, The Fluid Engineering Centre, Cranfield, Bedford MK43 OAL, UK
5th International Conference on Pressure Surges	22–24 September 1986 Hannover, FRG	BHRA Fluid Engineering Centre, Cranfield, Bedford MK43 0AJ, UK
ASME Winter Annual Meeting: International Symposium on Pressure and Temperature Measurement	30 November–5 December 1986 San Francisco, CA, USA	Dr J. H. Kim, Electric Power Research Institute, 3412 Hillview Avenue, PO Box 10412, Palo Alto, CA 94303, USA
Australian Fluid Mechanics Conference	8–12 December 1986 Auckland, New Zealand	AFMC Conference Committee, c/o Dr P. S. Jackson, Dept of Mechanical Engineering, Auckland University, Private Bag, Auckland, New Zealand
3rd International Conference on Multi-Phase Flow	19–20 May 1987 The Hague, The Netherlands	Conference Dept, BHRA, The Fluid Engineering Centre, Cranfield Bedford MK43 0AJ