# A review of turbomachinery tip gap effects

## Part 1: Cascades

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This two part review covers experiments examining the effects of blade tip gaps encountered in turbomachines and the methods by which the synthesised data are currently used in turbomachine design and analysis. Data gained since the 1930's are subdivided for convenience into cascade (Part 1) and rotating machinery[1] (Part 2) data, with a further subdivision into diffusing, or compressor type flows and accelerating, or turbine type flows. The overall trend is that an increasing tip gap, whose effect can reach over most or all of the blade height, reduces turbomachine performance. There is some evidence among the compressor and compressor cascade data that an optimum gap exists when the opposing effects of secondary flows and tip leakage with rotor/wall relative movement tend to balance. Turbine data are, in general, more regular than the body of compressor data, possibly because of the enhanced effect of, usually, undefined boundary layers in diffusing flow in the latter. Comment is made in Part 2 on the predictive and design models reported in the literature

**Key words:** *turbines, compressors, turbine blades, tip gap*

To gain high cycle thermal efficiency, gas turbine engines must use high compressor pressure ratios, resulting in relatively small annulus height in the high pressure section of the compressor. There the tip gap, which may be of fixed dimension, becomes large as a percentage of characteristic blade dimension. Since efficiency loss through the effects of blade tip clearance is a recognised feature of compressor operation, increasing generally with non-dimensionalised tip-clearance, it might be anticipated that tip loss would have an increasingly dominant effect. This trend runs, unfortunately, counter to the performance and geometric requirements of new generations of engine and particularly the concept of the energy efficient engine.

Layouts proposed for future engines show the turbomachinery occupying an increasing proportion of the total volume and hence total mass. Relating crudely the cost of manufacture to the total mass, component costs may be apportioned in relation to their proportion of the total engine mass. The turbomachinery thus represents a significant cost item in a gas turbine and, although the turbine demands the costlier materials, the compressor, because of its relative size, is at least as expensive in manufacture. Costs would be reduced, however, if manufacturing tolerances could be relaxed; from the economic point of view, increased tip clearance would be advantageous.

In addition to energy efficiency and minimum manufacturing costs, the designer of the gas turbine engine must ensure high standards of safety in operation. Thus, the equilibrium running line on the compressor characteristic map must be sufficiently separated from the stability limit line to maintain an adequate surge margin, permitting normal transient operation without destabilising the system. The stability limit of a compressor is governed by the onset of rotating stall which, depending on system dynamics, may develop into a surge. Rotating stall is itself usually identified both for part-span and full span stall at the outer sections of the blading, the region of the rotor tip gap, so it may be anticipated that both the initiation of the rotating stall and its characteristics may be affected by interaction with the tip gap aerodynamics; such reasoning has led to the introduction of tip treatments in certain compressor designs.

Clearly then, tip gap aerodynamics play an important part in compressor performance and will be of increasing significance with new engine designs. Historically, however, the record of dealing with the tip gap problem is depressing. Osborne Reynolds, who held what was possibly the first patent on a turbomachine, abandoned his work when he found that only low efficiency could be obtained on a water turbine of small size that he designed. This inefficiency was due, in the main, to tip loss, which would have become less dominant for a gap of the same dimension in larger turbomachinery, a point not pursued by Reynolds. In a discussion on axial flow pumps and impellers held at the Institution of Mechanical Engineers in 1956 Professor A. D. S. Carter suggested a 'rough and ready rule' that clearances up to about 2% of the blade height could be assumed to have practically no effect; the advances from the time of Osborne Reynolds were

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mainly in recognising that a tip gap could have an effect, as yet poorly quantified.

Since every turbomachine must contain some form of gap at the blade tip, it is surprising that so little research has been done in this field. Of the work that has been done, especially on compressors, much has been piece-meal and many features which may now be recognised as important, for instance the annulus wall boundary-layers, have not even been measured. Some data are available for cascades and rotating machinery, both compressive and expansive, and it is these data which have been considered in this two-part review[1].

As well as indicating what has been accomplished, areas demanding further investigation, in the light of new engine design, are suggested. This synthesis of data, measured over a number of years, yet not previously collated in this manner covers both cascades (Part 1) and rotating machinery (Part 2).

## Background

There is evidence that increasing the tip gap of a compressor rotor involves a significant efficiency penalty. Enlarging the gap by 1% of the blade height may yield approximately a 2% drop in compressor efficiency in certain current large scale gas turbines using blades of large dimension. Towards the rear

### Nomenclature

| | |
|---|---|
| $A$ | Blade aspect ratio, $Z/c$ |
| $C_h$ | Stalling static pressure rise coefficient |
| $C_L$ | Lift coefficient |
| $C_{L_r}$ | Lift coefficient retained at the blade tip section |
| $C_m$ | Mass flow coefficient, $\int_0^\lambda v\, dz/[2(P_2-P_1)^{1/2}/\rho]\lambda$ |
| $C_N$ | Normal force coefficient |
| $C_p$ | Static pressure rise coefficient |
| $c_p$ | Specific heat at constant pressure |
| $c$ | Blade chord |
| $D$ | Rotor tip diameter |
| $E_L$ | Blade end coefficient, $2(P_L-P_u)/\rho U^2$ |
| $F_y$ | Tangential force on a blade row per unit span, $2\pi r\rho V_x[(r_{Z+\Delta}V_{xZ+\Delta}-r_Z V_{xZ})/r]$ |
| $f_N$ | Frequency |
| $g$ | Staggered spacing of blades, $S\cos\xi$ |
| $g_0$ | Gravitational constant |
| J | Joules equivalent |
| $K$ | Measured lift coefficient retained at blade tip $=C_{L_r}/C_{L_{2D}}$ |
| $K_{1,2,3}$ | Constants |
| $k$ | Cavitation number |
| $N$ | Rotational speed |
| $n$ | Number of blades in a blade row |
| $P$ | Stagnation pressure |
| $P_L$ | Stagnation pressure at blade pressure surface |
| $P_u$ | Stagnation pressure at blade suction surface |
| $p$ | Static pressure |
| $q$ | Dynamic head |
| $R$ | Tip reaction |
| RBH | Reduced blade height configuration |
| RC | Recessed blade casing configuration |
| $r$ | Radial dimension |
| $S$ | Blade pitch |
| SH | Shrouded blade configuration |
| $T_0$ | Stagnation temperature |
| $T$ | Static temperature |
| $t$ | Profile maximum thickness at the tip section |
| $U$ | Blade speed |
| $V$ | Flow velocity |
| $x$ | Axial co-ordinate |
| $y$ | Pitchwise co-ordinate |
| $Z$ | Blade span |
| $z$ | Spanwise co-ordinate |
| $\alpha_1$ | Flow entry angle to blade row |
| $\alpha_2$ | Flow exit angle from blade row |
| $\alpha_M$ | Mean air angle, $\tan^{-1}[(\tan\alpha_1+\tan\alpha_2)/2]$ |
| $\Gamma_m$ | Mean local value of circulation |
| $\delta^*$ | Boundary layer displacement thickness |
| $\gamma$ | Ratio of specific heats |
| $\eta_{AD}$ | Adiabatic efficiency of the stage |
| $\lambda$ | Tip gap height in turbomachine and cascade or tip gap semi-height in imaged cascade experiments |
| $v$ | Tangential force thickness of boundary-layer |
| $\xi$ | Blade stagger angle |
| $\rho$ | Density of working fluid |
| $\sigma$ | Throttling coefficient, $(V_a/U)^2/[2(P_{out}-P_{in})]/\rho U^2$ |
| $\phi$ | Mass flow coefficient, $V_a/U$ |
| $\psi$ | Work coefficient, $\Delta H/U^2$ |
| $\zeta$ | Incidence angle |

*Subscripts*

| | |
|---|---|
| 2D | Two-dimensional flow |
| x | Axial direction |
| 1 | Entry plane of blade row |
| 2 | Exit plane of blade row |
| a | Axial component |
| out | At compressor outlet |
| in | At compressor inlet |
| t | Tip value |
| h | Hub value |
| max | Maximum value |
| GAP | Inter-blade row gap value |
| R | Rotor |
| S | Stator |

*Superscripts*

| | |
|---|---|
| ~ | Free stream quantities |
| – | Mean value |

of a high pressure compressor, where blades are of small dimension, the effect could be greater. With the need for increased component efficiency in engines, this is a severe penalty.

The mechanics of the loss inducing flows associated with the rotor tip region are barely understood so that attempts to model the flow or reduce its effect are thwarted. It is known, however, that small geometric variations in the region of the rotor tip can have a substantial effect on performance; not only do increased tip clearances generally reduce efficiency but certain types of tip treatment are known to improve the compressor behaviour close to the stability limit line and hence improve the surge margin[2]. At the same time there may be an efficiency penalty in compensating for the increased operational range. As with the case of a tip gap interacting with a rotor blade and a plain annulus wall, the detailed aerodynamics associated with various types of tip treatment, and thus the reasons for changes in compressor overall performance, are not known.

Knowing the effect of tip treatments, it may be presumed that the stability limit of a compressor, marked by the onset of rotating stall, is likely to be governed to some extent by the tip gap geometry and aerodynamics, as is the post-stall recovery which, because of hysteresis effects in the process, is unlikely to occur at the same point on the characteristic. Stability limit operation and post-stall recovery are of vital interest to both manufacturer and user.

Rotating stall is one class of non-axisymmetric flow; although a compressor is designed for uniform inlet flow conditions, a second class of non-uniform flow, circumferential distortion, may occur in practice. The installed gas turbine often has its inlet flow distorted by interactions with upstream bends, struts, shock systems or, for an aircraft undergoing a high 'g' manoeuvre, boundary layer separation at one side of an intake or partial masking of an intake by the aircraft body. Such distortions, although predominantly quasi-steady, and hence planar, usually have some time-unsteady content. There is much evidence to show that the circumferential planar distortions generated by such features have a deleterious effect on performance altering the shape of the running line and precipitating compressor destabilisation by depressing the stability limit line to the compressor operating point[3]. Although this form of distortion is time-steady in the absolute frame of reference, relative to the rotor it is seen as a time-unsteady effect, producing a time-unsteady reaction from the rotor blade[4]. The time dependent features include a complex hysteresis loop in the lift and incidence line[4], but can also involve rotating stall which may be initiated by the distortion. Since it has already been observed that the characteristics of the rotating stall may be dependent upon the rotor tip gap, it may be anticipated that an interaction would occur between the distortion generated and the tip gap, particularly under low flow/high blade load conditions.

In general, a reduction in tip gap size improves the efficiency of the unit but higher costs are involved. The engineer must know the trade-off between compressor performance as a function of rotor tip clearance and cost so that the economics of manufacture may be measured against cycle efficiency.

The effects of the tip gap in a turbine parallel largely those of a compressor. A loss of efficiency with an associated loss in work coefficient is involved and the perturbed aerodynamics may reach over much of the blade height. In turbines, however, destabilisation is not of interest, but the tip gap problem is enhanced because of the sizes of gap necessary to accommodate thermal growth.

The rotor tip is a sensitive area which affects compressor operational efficiency, stability limit, post-stall recovery, and distorted inflow performance and turbine efficiency, flow coefficient and detailed aerodynamics. To make significant advances in these areas resulting in reduced tip losses through geometric variations, casing treatment and tip feathering calls for a realistic flow model gained from an understanding of the physics involved. The flow is, however, highly three-dimensional, viscous, time-dependent and possibly compressible; such interdependent effects cannot yet be modelled. To obtain the data upon which a model, resulting in a designers' and analysts' code, may be based, it is necessary to resort to experiment. Such experiment, however, needs to be comprehensive in its scope and detailed in its measurement.

## Basic flow mechanisms in tip gaps

A substantial body of evidence indicates that while turbomachine manufacturing costs decrease with increasing tip clearance, there is a progressive performance penalty. The mechanics of the flow that result in performance loss are not understood in detail, but it may be assumed that the contributory factors include:

1. The pressure difference between the suction and pressure surfaces of a blade, resulting in a leakage flow through the tip gap from the pressure to the suction surface. In the case of an aircraft wing this results in the tip vortex and the progressive loss in lift of the wing section towards the wing tip. Associated with this in a compressor or turbine is the vortex flow and related radial distribution of loading to which the blading was designed.
2. The presence of the boundary layer on the casing or hub wall of the turbomachine. This shear flow itself affects the tip region performance encouraging secondary distributed vorticity which, locked in the region of the interface between the blade suction surface and the annulus wall, results in a three-dimensional separation contributing to loss of efficiency.
3. The relative movement between the blade and the boundary layer on the casing or hub wall, encouraging a jet flow through the gap. For a compressor blade the jet flow relative to the blade is from the pressure to the suction surface, additive to the flow mentioned in (1) above. For a turbine blade, because the relative movement is in the opposite direction, the jet flow relative to the blade is from the suction to the pressure surface, in opposition to the mechanism in (1).

4. The size of the gap. Clearly, as the gap increases in size, resistance to the flow mechanisms in (1) and (3) is reduced, so their effect is generally increased. For very large gaps, the effect of reduced blade height in reduced work associated with the change of blade height can also become noticeable.

Attempts have been made to understand in detail the individual effects listed above, possibly with a view to superimposition of results to gain an understanding of the complete phenomenon.

Of the three fluid mechanical, as opposed to geometric, features listed, the flow at the tip due to the pressure difference between the surfaces may be considered inviscid, while those due both to the presence of the casing boundary layer and the relative motion between the rotor tip and the casing are viscous. It is possible to separate these effects somewhat so that the inviscid and viscous effects may be examined in isolation and this has been an approach used by various workers.

Although there are obvious geometric similarities between compressor and turbine rotor assemblies in the region of the tip gap, there are likely to be different aerodynamic effects because the flow across a turbine blade is accelerating and therefore:

- The wall boundary layers are not so prone to separation;
- The pressure distribution around the blade profile is rather different to that of a compressor. With a compressor blade profile the maximum pressure difference across the tip occurs towards the leading edge and this probably locates the tip shed vortex observed and used in certain of the models. In general, however, a turbine blade profile maintains a more constant chordwise pressure difference across the tip and it may be anticipated that this will alter the geometry of the flow.

## Compressor cascade experiments

### *Tip gap behaviour with inviscid flow*

A technique to investigate the inviscid flow through a tip gap involves the use of imaging—a blade in cascade split at the mid-height section to produce effectively a gap between two blade ends. Ignoring the viscous growth on the end surface of the blades and that on the blade sections themselves, for a uniform inlet flow such as one would be likely to have at a cascade mid-region, the flow may be assumed to be inviscid. In a two-dimensional cascade, the two tip sections resulting from the geometric split are identical resulting in identical, image flow at each tip. There is thus a line of symmetry at the mid-gap height which becomes the height of the pseudo-inviscid gap to be investigated. This geometry was used by Khabbaz[5], Yokoyama[6] and Lakshminarayana and Horlock[7,8].

The aim of Khabbaz's experiments was to determine the effect of the gap in inviscid flow on the stall characteristics of the blade section. He noted that, at the tip section, stall was delayed to a higher flow incidence in the presence of a gap than without the gap. He also found that increasing gap size increased the blade loading near the tip while, for the remainder of the blade, conditions remained substantially unaltered. Yokoyama[6], using the same test facility observed that, while the lift coefficient increased near the tip region, the axial velocity and the turning angle locally decreased (Fig 1). These observations discredited a momentum theory model proposed by Khabbaz but led to the conclusion that the tip vortex made a strong contribution. Yokoyama also observed that the enhanced lift close to the tip was similar to that measured by Holme[9] on a rectangular wing of unity aspect ratio (Fig 2). Yokoyama's experiments were all conducted at gap semi-height/chord ratio $\lambda/c$ of 0.03 while Khabbaz varied the gap from $0<\lambda/c<0.10154$. Lakshminarayana and Horlock[7,8] using a similar geometry, measured the spanwise lift distribution for $0<\lambda/c<0.426$. At small gap geometry ($\lambda/c<0.03$) the lift results were qualitatively similar to those of Yokoyama (Fig 3). With larger gaps, the blade experienced a reduction in total lift, the spanwise extent and overall lift reduction being progressive with gap size. In terms of normal force coefficient variation with blade span and gap size, their data

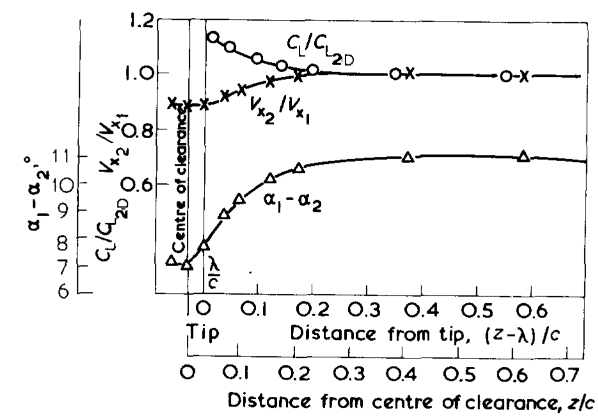


*Fig 1 Distributions of lift coefficient outlet axial velocity and turning angle versus spanwise distance*[6]

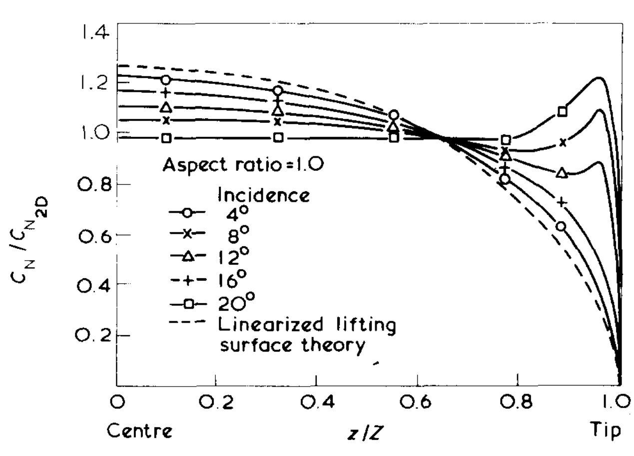


*Fig 2 Normal force distribution along span of rectangular wing*[9]

were closely co-incident with Yokoyama. It was observed, however[8], that in the range of gap size normally employed in turbomachines, there was always a slight increase in average normal force coefficient (Fig 4).

Lakshminarayana and Horlock also measured a progressive increase in induced drag with increased gap size (Fig 5). This change with gap/chord ratio inferred a change in shed vorticity at the tip, leading to the conclusion that, in the experiment with imaged blades separated by a gap, not all of the bound vorticity was shed at the blade tip. The bound vorticity towards the blade leading edge bridged the gap. This contradicted Dean's supposition[10] that, in an inviscid flow field, as soon as the tip clearance becomes finite, no vortex lines could cross the clearance gap. Dean concluded that all of the bound vorticity must then trail off as a vortex sheet of varying strength behind the blade in the manner of a wing. Such a model which supposes that the flow field comprises a series of insulated, two-dimensional lamellae laying in the $x$, $y$ plane, must admit the presence in the flow of infinite values for $\partial v/\partial z$ and $\partial p/\partial z$, the rates of change respectively of velocity and pressure in the spanwise direction. Even in an inviscid fluid, the proximity of an aerofoil section promotes non-infinite pressure gradients around the aerofoil tip and hence, for the geometry under consideration, within the tip gap, resulting in the transverse flow that comprises the tip vortex. It may be concluded then that the resulting pressure field within any particular lamella ($x, y$ plane) embracing the gap region promotes a varying velocity producing circulation and hence vorticity in the $x$, $y$ plane. Extending the argument in an inviscid fluid to a cascade wall, as long as a non-constant pressure field exists at the wall, vorticity will enter the wall.

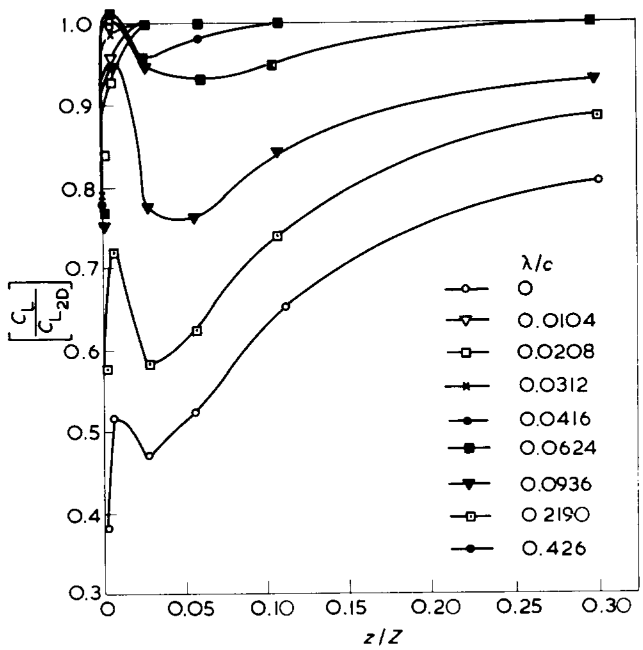


*Fig 3 Spanwise lift distribution at various gap chord ratios*[7]

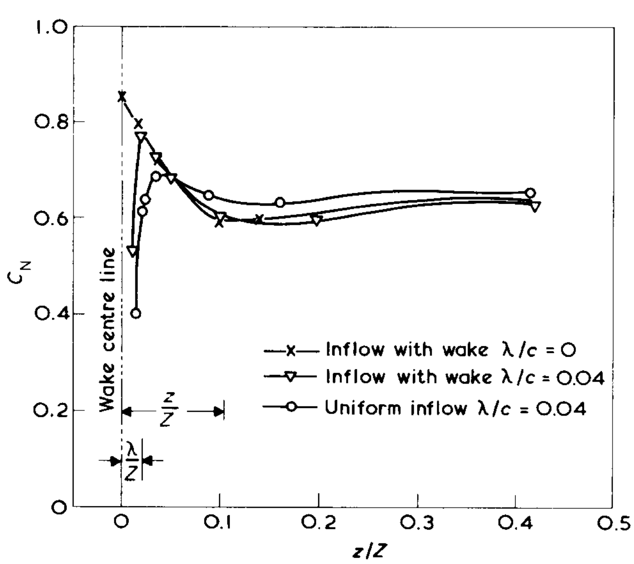


*Fig 4 Spanwise distribution of normal force coefficient for λ/C = 0 and 0.04 (Experiment A and B)*[8]

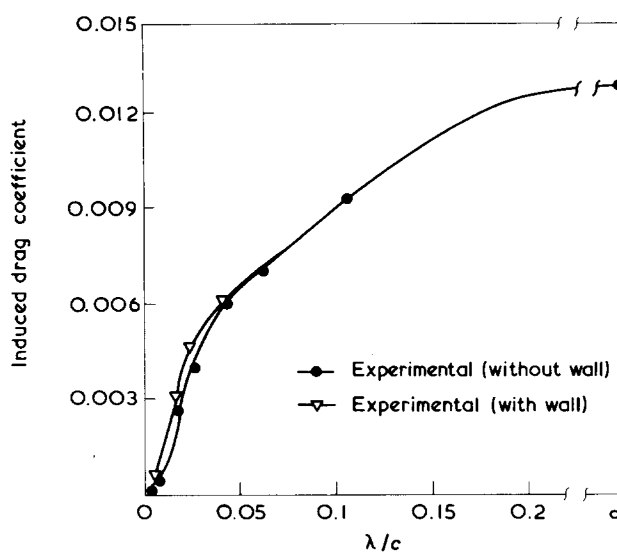


*Fig 5 Experimental and theoretical variation of induced-drag coefficient with λ*[7]

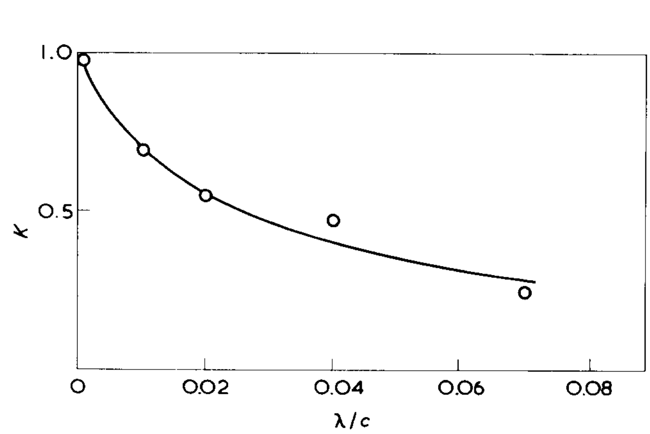


*Fig 6 Values of lift retained (as a fraction of two dimensional value) at the tip of the cascade blade*[8]

This reasoning supports the observations of Lakshminarayana and Horlock[7] for an inviscid flow field. They developed the concept of vortex lines, which originally passed as bound vortices from a blade to its image at zero tip spacing, being progressively shed as the gap increased. From the above reasoning it may be concluded that all of the bound vorticity would be shed at the moment when the mid-gap plane $(x, y)$ representing the pseudo-wall no longer contained any static pressure variations.

For low gap/chord ratios, the experiments led to a model based upon the observation that the shed vorticity from the blade tip was $(1-K)\Gamma_m$ where $K$, the measured lift coefficient retained at the tip as a proportion of its two-dimensional value, was derived from experiment (Fig 6) and $\Gamma_m$ was the mean local value of circulation.

With very large values of gap/chord ratio, the tip aerodynamics tended, with reducing interaction between the two blades, to those of an isolated aerofoil where all the tip region-bound vorticity was shed in the tip region ($C_{L_r} \rightarrow 0$). The shed vorticity thus tended to $\Gamma_m$, the mean local value of wing circulation.

### *Simulation of an inlet shear flow*

In a further series of experiments Lakshminarayana and Horlock[7,8] allowed for the presence of a wall and shear flow in three ways, by placing:

(a) in the gap mid-span a splitter wall extending from the blade leading to trailing edge[7].
(b) upstream of the gap, a perforated splitter plate whose wake was convected through the gap[8].
(c) a downstream extension of the perforated splitter plate to continue it through the gap[8].

Gap size was always variable from zero.

All of these geometries led to the generation of secondary flow as a consequence of the attendant shear flows. In case (a), no shear was present at the cascade entry plane and only built up as a thin boundary layer through the cascade channel. The results indicated a reduction in total blade lift and increase in total induced drag coefficient, commensurate with breaking the bound vorticity previously bridging the gap between the blade and its image.

Convecting through the test section an upstream generated shear flow whose semi-span was about five times the gap semi-height generated a secondary flow within the channel. This was opposite in direction to the leakage flow, observed in the test with uniform inlet flow, which it met to form a core of heavy loss at the exit plane in the mid-passage. With no gap, this loss was locked into the corner created by the interface between the blade suction surface and the sidewall as measured by Peacock[11]. The spanwise distribution of loss coefficient indicated that the effect of the secondary flow was to spread the loss coefficient over a larger part of the blade span and the magnitude of loss increased with gap size. Over much of the blade span the normal force coefficient based upon local inlet dynamic head was reduced somewhat, except in the region of the tip where it was enhanced. The effect of closing the gap was to eliminate the reduction in normal force coefficient immediately at the tip section where the normal force coefficient reached a maximum. The effect of the secondary flow upon the air leaving angle was to reduce the underturning created by the leakage flow through the gap.

Experiments with geometry (c) confirmed that the effect of a shear flow was to create a secondary flow structure operating in opposition to the leakage flow. It was noted that the two flows could be to some extent balanced by tuning the tip gap. The leakage flow through the gap as a result of the pressure difference across the blade from the pressure to the suction surface, which reduces $\alpha_2$ locally, might be balanced by the effect of the secondary flow in the region of the sidewall, which generally increases $\alpha_2$.

At $\lambda/c = 0.04$ Lakshminarayana and Horlock saw that the leakage flow tended to displace the corner separation zone at the blade and end wall surfaces: also the secondary flow, in opposing the leakage flow, prevented the leakage flow from moving down the suction surface along the blade passage. The effects of leakage were confined to a region near the suction surface while the flow elsewhere was greatly influenced by secondary flow and separation. The effect of leakage was therefore to increase the underturning at the suction surface near the wall and decrease it in most other regions, improving the lift distribution along the blade.

### *Relative movement between a blade and wall*

In balancing the effects of the pressure difference across the blade surfaces (inviscid) with the secondary flows (viscous) no account was taken of relative movement between the blade and the sidewall or viscous layer. In a normal rotating machine this effect results in changes in the leakage flow rate at the gap because of convection of the viscous flow through the gap. This viscous effect is in opposition to the secondary flow viscous effect and in sympathy with the inviscid flow direction across the gap. It may then be concluded that the effect of relative movement could be significant.

To investigate this Gearhart[12] constructed a facility in which air passed from a plenum chamber through a bellmouth and entry region to a test section containing a blade with tip clearance. Along the entry region and across the blade tip an endless belt was designed to move in the direction of the airflow. Tests investigated the effects both of relative velocity and blade end geometry. Variables investigated in the programme included gap height, blade end geometry and a blade end coefficient ($E_L$) defined by:

$$E_L = \frac{P_L - P_U}{\rho U^2/2}$$

relating the difference of the average stagnation pressure on suction and pressure surfaces of the blade to the dynamic pressure associated with the belt velocity. For zero belt speed, the value of $E_L$ was $\infty$. For non-zero belt speed (kept constant in the programme) $E_L$ was varied by altering the pressure difference across the blade and by varying plenum chamber pressure.

Although this work was aimed primarily at investigating cavitation problems in water pumps, the experiments reported were in air and certain data of use to the aerodynamicist emerged:

- The effect of decreasing the tip gap was to increase the local maximum depression in the gap, a consequence of decreasing the effective throat in the tip region.
- The effect of increasing the blade end coefficient, by increasing the pressure drop ($P_L-P_U$) across the gap was to reduce the absolute level of non-dimensional velocity (Fig 7) and hence non-dimensional mass flow through the gap (Fig 8). One test with zero belt speed ($E_L=\infty$) produced the minimum non-dimensionalised leakage flow. It may be concluded that the result of blade/wall relative velocity was to increase the tip leakage flow, the effect of viscous flow transported at the wall being in sympathy with the leakage flow normally present in a stationary experiment.
- Tests in which the blade end geometries varied yielded sharply differing results.

The experimental programme, while accounting for relative movement between blade and sidewall did

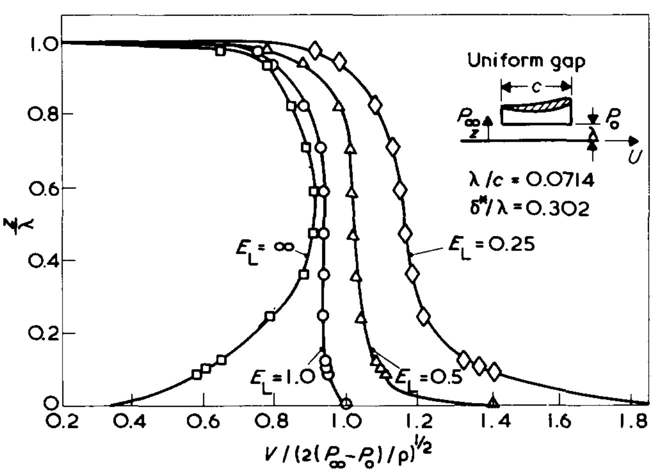


*Fig 7 Non-dimensional velocity profiles at exit of uniform gap*[12]

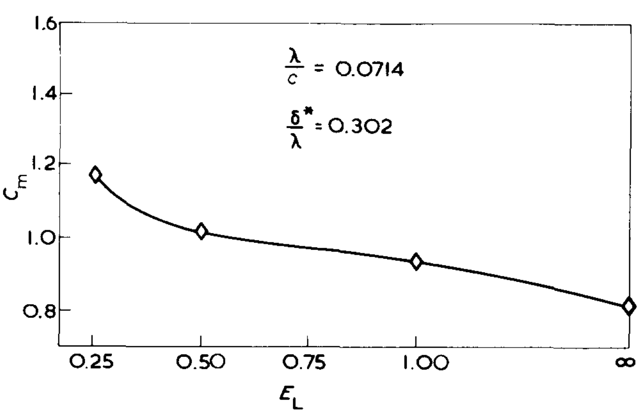


*Fig 8 Mass flow coefficient versus blade end coefficient for various gap configurations*[12]

so with a belt moving in the flow direction. The boundary layer at the sidewall remained two-dimensional with respect to the blade, which does not represent the situation encountered in a turbomachine where the boundary layer is highly skewed with respect to a rotor blade.

Dean[10] also executed a moving belt experiment using a rectilinear cascade in which one wall contained a moving belt adjacent to a gap at the end of the cascade aerofoils. The belt covered an axial distance from 0.4 blade chords upstream of the cascade inlet to the cascade exit plane. In Dean's experiment the blades, inlet flow and belt movement were in the correct relative relationship: the boundary layer was not convected simply in the direction of the stream flow, but had a component in the pitchwise direction of the blading. Further, this pitchwise component being superimposed because of its viscous properties and being most evident at the belt surface, imparted a skew to the boundary layer as it would be seen by the rotor blade of a compressor.

Initial experiments with the belt stationary and with no tip gap confirmed the observations of Peacock[11] and Lakshminarayana and Horlock[8] that a region of low energy fluid was present at the blade suction surface in the tip region. This phenomenon is also present in a compressor and detailed measurements of it have been made by, among others, Dring, Joslyn and Hardin[13,14]. Upon creating a finite gap, Dean noted that this region was moved by tip leakage away from the suction surface to a position approaching mid-pitch close to the cascade sidewall. Again, this was in line with the measurements of Lakshminarayana and Horlock. It was further observed that this effect was enhanced by operating the belt when the viscous effects of the flow were additive to the tip leakage flow. It was however noted that a tip gap ($\lambda/c$) greater than 3.4% did not materially affect the position of the low energy core in the cascade channel and a wall speed greater than 127% of the flow axial velocity also had only a small effect. Qualitatively the effect of wall speed was dissimilar to that of Gearhart[12] who found that the velocity and mass flow through the gap increased with belt velocity (reduced $E_L$), the effect becoming more dominant with higher belt speeds. It would be anticipated that the low energy core measured by Dean would have been moved progressively farther across the passage.

## Turbine cascade experiments

Using a water-table and a series of simple geometries Booth, Dodge and Hepworth[15] examined the flow across a tip gap without relative wall movement. They identified three regions in which different terms of the momentum equation were dominant.

1. At low clearance levels, where shear forces directly balanced pressure forces;
2. At normal clearance levels where convection became dominant;
3. At high clearance levels where pressure gradients became small because of tip unloading.

Their series of tests showed the flow to be primarily inviscid with a high discharge coefficient insensitive to flow on the blade suction surface.

In further tests in which various geometric alterations were made to the tip section, a reduced thickness tip section produced a noticeable performance degradation and while at a particular thickness the degradation reduced with reduced tip gap the effect of the tip thickness became greater.

Wadia and Booth[16] noted that, for a turbine, discharge coefficients were reduced due to wall motion because the relative movement of the wall was against that preferred by the leakage flow (in a compressor the opposite argument holds). They also presented computed streamline patterns in the tip region showing circulation above the suction surface adjacent to the tip region. While these data are rather similar to measurements on compressor cascades, eg Dean[10] and Peacock[11], they do not coincide with turbine cascade data of Peacock[17] who found that the vortex migrated an appreciable distance over the blade span.

## Applicability of cascade results and models

The review of the literature on cascade measurements covers cascade blade tip gap measurements for a variety of conditions:

- With inviscid uniform flow;
- With inviscid non-uniform flow;
- With sidewall boundary layers of varying strength;
- With relative movement between the blade tip and the sidewall.

It is as well to examine the manner in which these experiments simulate conditions in a real turbomachine.

The experiments of Khabbaz[5], Yokoyama[6] and some of the work of Lakshminarayana and Horlock[7,8] were with an imaged blade in uniform inviscid flow. Although Dean[10] indicated that in such a situation none of the bound vorticity could bridge the gap to enter the annulus sidewall, this concept was countered by Lakshminarayana and Horlock[7] who introduced the idea that as the gap grew, vortex lines from the blade tip to its image were progressively cut and turned into the tip-locked trailing vortex. Their statement has already been supported by the earlier empirical argument for the inviscid flow case. The applicability of this concept to the viscous flow situation universally encountered in turbomachinery must however be investigated.

Appealing to Helmholtz's second and third laws of vortex motion which are applicable in an inviscid, incompressible flow, a vortex filament must form a closed loop or terminate at a boundary. In the instance of a finite wing in an infinite flow field, the vortex filament bound to the wing forms a closed loop with the downstream convected starting vortex by the shed wing-tip vortices. For a two-dimensional cascaded aerofoil in an inviscid uniform flow field there is a similar vortex loop, the shed vortices being contained conceptually within the sidewalls. In the case of the experiments described above in which a blade was terminated at one end at the sidewall and at the other end by a tip gap which reached to an image blade, and assuming that the sidewall flow was inviscid there was then a closed loop of vorticity locked to each aerofoil individually, one streamwise leg conceptually within the wall and the other actually shed at the blade tip. A further loop of vorticity, part of which bridged the gap, embraced both aerofoils and was shed streamwise within the sidewalls to connect with the starting vortex. The sum of the individual aerofoil vortex loop and that bridging both aerofoils yielded, at an aerofoil, the lifting line vortex and hence the section lift.

On introducing a viscous flow to a plain cascade at its sidewalls, a new situation exists. At the wall section there is a zero velocity for the usual no-slip condition. Because of the zero flow velocity in that plane, there can be no circulation at the aerofoil section as it intercepts the wall. In the viscous flow situation however, it is possible to have retained lift at the sidewall section as a consequence of the static pressure field generated by the blade in the moving fluid and transmitted through the boundary layer. This represents a condition of lift without circulation. With no circulation, no vortex line can pass into the wall from the section. The shed vorticity is thus constrained to leave the blade section within the flow field but maintaining the overall picture of a vortex loop as previously described. This shed vorticity is described by Hawthorne[18] as trailing shed vorticity.

As with a solid end cascaded aerofoil, so too with an aerofoil with a tip gap separating it from an end wall; no vortex line intercepts the wall and all must be shed within the flow field even though, in the viscous flow case, there is retained lift at the wall due to the superimposed static pressure distribution.

This conclusion is however in contradiction to that of Dean[10] in considering a tip gap with the presence of a boundary layer. Because of the frictional effects present he postulated that by the vehicle of viscosity a pressure difference could be maintained, permitting the bound vorticity to cross the gap. Because of the viscosity, however, the no-slip condition maintained at the wall, while admitting the pressure field present, precludes the velocity field at the wall necessary for vorticity to exist there.

The results of Lakshminarayana and Horlock[7,8] were interpreted, however, to indicate vorticity bridging the gap in their experiment with a centrally split blade and since, in bridging the gap, the line of symmetry of the experiment was crossed, vorticity conceptually entered the pseudo-wall represented by the line of symmetry. Such an experiment is not then representative of the physics of a real fluid passing through an assembled compressor with clearance between the blade tip and casing. The only way in which the correct vortex shedding condition could be obtained at the gap would be if, at a plane orthogonal to the blade section, the streamwise flow was zero, a condition that can exist only if a splitter plate is fitted. This though introduces a boundary layer at the gap.

The cascade geometry used by Lakshminarayana and Horlock in which a splitter plate was fitted, transporting a boundary layer within the gap was more realistic and in fact would be representative of a cantilivered and shrouded stator assembly or inlet guide vane row in a compressor.

The effect of blade/wall relative motion was not covered in Refs 5–8, but was investigated by Gearhart[12] and Dean[10] and although their results have certain qualitative differences, possibly due to geometric variations, the convection effect due to the wall movement, though not as powerful as the effect of the presence of the gap alone, was seen to be important.

No cascade experiment, however, is able to investigate radial effects, centrifugal force on the blade boundary layer and the consequence of radial variations in the blade flow due to design considerations, so it remains to examine data from real turbomachines to evaluate these.

## Conclusions

Part 1 of this review has covered about 30 years of research into tip gap phenomena in cascades. A good body of detailed, but uncoordinated work has resulted for diffusing type cascades, while that for accelerating type cascades is both sparse and lacking in the same detail. A good physical understanding of the flow in a tip gap region and the flow effect on conditions across the cascade channel have resulted, but the models proposed, as well as having a degree of mutual contradiction, all fail to deal adequately with the distribution of the tip vortex within the gap. In addition, while some workers have used ingenious methods to simulate rotor tip/wall relative motion, they have not been able to represent all the effects of rotation and may not have given a realistic representation of the casing boundary layer in rotor relative coordinates. Nevertheless, the detail with which it has been possible to make measurements in the comparatively easy conditions of a cascade has led to an understanding of the flow structure.

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## Corrigendum

In the Technical Note by R. Peretz titled 'Relation between evaporator and condenser lengths of a finless heat pipe to achieve a maximum heat flow per unit weight' published in the September issue (Volume 3, No 3), Eq. (14) contained $\frac{1}{2}\lambda$ rather than $(1/2\lambda)$ in both numberator and denominator. Thus the equation should have read:

$$(L_c/L_c)^2 = \frac{h_{o,c}d_o)^{-1}+(1/2\lambda)\ln(d_o/d_i)+(h_{i,c}d_i)^{-1}}{h_{o,e}d_o)^{-1}+(1/2\lambda)\ln(d_o/d_i)+(h_{i,e}d_i)^{-1}} \quad (14)$$