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Axial Turbine Blade Tips: Function, Design, and Durability

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An overview of the science and technology involved in today's turbine engines is presented with specific focus on the critical rotational-to-stationary interfaces comprising axial turbine blade tips. The purpose is to provide a concise informative review of turbine blade tip functional, design, and durability issues. Neither a historical account nor a bibliography is presented. Attention is paid primarily to the most challenging blade tips in high-pressure, high-temperature gas turbine systems, although most of the science discussed applies equally well to blade tips in low-pressure turbines, as well as steam turbines. As such, a wide range of both aircraft engine and power generating turbine systems are considered. Basic functional requirements, turbine systems design aspects, and transient operational considerations affecting blade tips and affected by blade tips are discussed in light of the multidisciplinary tradeoffs involved in a successful design. The three dominant design philosophies for blade tips in practice today are presented with detailed examination of the aerodynamics, heat transfer, and cooling benefits and detractors. Finally, the in-service durability aspects of turbine blade tips are noted.

Introduction

TURBINE blade tips have been, and continue to be, not only one of the major causes for loss of efficiency in a turbine engine, but also a primary contributing factor in the operational degradation of turbines leading to periodic removal from service for repairs. As with all components of the turbine hot-gas path, blade tips must perform multiple functions while being subjected to many design and operational constraints. This is true of both the cooled blade tips in the high-pressure turbine stages and also the uncooled blade tips of the low-pressure turbine stages. The uniqueness associated with all turbine blade tips comes in the multifaceted complexity of the rotational–stationary interface between the work extraction fluid and the turbine casing. Turbine blade tips are distinctly different from the compressor and fan blade tips in at least two respects. Turbine blade tips are subject to higher temperature gases, in some cases exceeding 1400°C in the first high-pressure turbine stages, and these tips are also subject to far higher aerothermal loading (pressure ratios). Aircraft engine high-pressure blades may see as much as 25-atm pressure, whereas large power turbine blades may see about 12 atm, and both may experience blade row pressure ratios up to 2. A perfectly functioning blade tip will not allow any leakage of valuable working fluid over the tip, from pressure side to suction side, which would bypass or short circuit the extraction of work by the turbine. A perfect blade tip will also require no cooling, thereby presenting no thermodynamic losses from the use of chargeable flows and no mixing losses from injection of these flows into the main working fluid. A perfect blade tip will in addition generate no secondary flows to reduce stage efficiency, or to contribute to losses in downstream airfoil stages. These are the major goals that all designs seek to approach, but none attain. Instead, the more realistic goal is to minimize the impact of the imperfections while also satisfying several other system operational requirements.

The present paper will review the competing requirements and constraints placed on turbine blade tips, describe several approaches used in actual designs to satisfy these conditions, provide in-depth summaries of the aerodynamics and heat transfer associated with the major blade tip designs, and highlight aspects of durability that must be included in any successful turbine design. A more detailed treatment of these topics may be found in the recent Von Kármán Institute lecture series of Glezer et al.[1]

Function of a Turbine Blade Tip

The most fundamental question to ask about a blade tip concerns its basic function as part of the entire blade. The turbine blade is a work extraction device balancing the requirements of aerodynamics, structural life, material properties, and thermal efficiency. An individual blade should not be considered a singular element. As shown in Fig. 1,[2] the blade must be in concert with the rotor, the upstream and downstream airfoils, the shrouds and secondary systems, and of course the hot-gas path fluid. Because the turbine rotor inlet temperature (TRIT) is raised to improve system thermal efficiency (for a given aerodynamic efficiency), the blade life generally decreases drastically. When blades exceed a safe temperature limit for the

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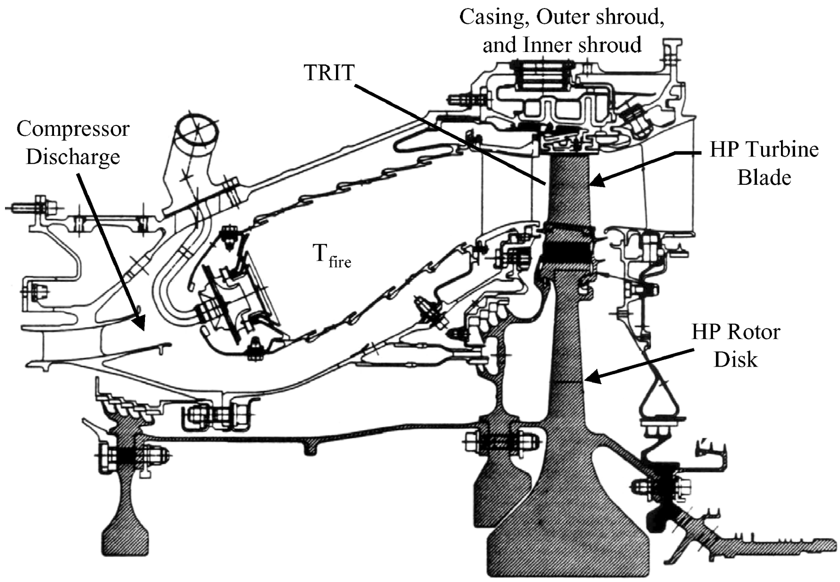

Fig. 1 Gas turbine engine schematic showing combustor and HPT.

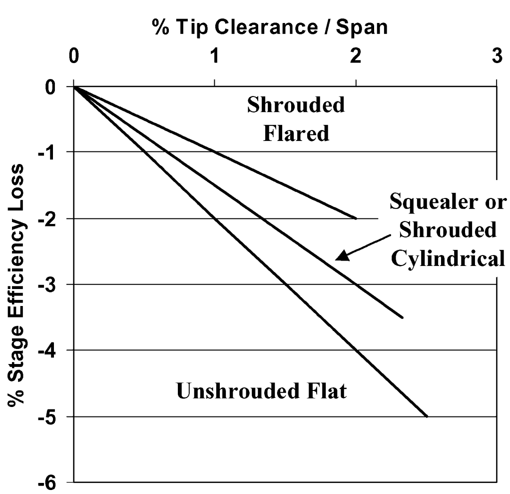

Fig. 2 Effect of blade tip clearance on stage efficiency.

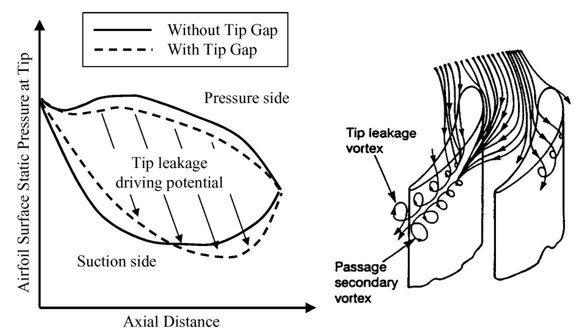

Fig. 3 Generic tip leakage pressure field and flow schematic.

materials, they are cooled, most commonly using the compressor discharge air, to retain adequate material properties and structure. Such cooling represents a short circuiting or bypassing of the ideal engine main flowpath and decreases the cycle efficiency. A portion of the useful work contained in the cooling air (pressure) is used up in the act of cooling the blades, leading to the term chargeable air because this pressure loss is charged as a deficit to the cycle. The remainder of the energy in the cooling air (work and heat) is still recovered in the main flowpath, although with some mixing losses. Up to a certain point, the economical use of blade cooling allows greater overall thermal efficiency via increased TRIT.

These general considerations apply equally well to the specific region of the blade tip, but in addition the tip must also live in harmony with the blade that carries it. The entire blade is designed for maximum work extraction within other system limitations, and the tip is no exception. The blade tip is the required free end of the blade forming the interface between the rotor and the outer diameter flowpath stator (or casing or shroud). The unavoidable physical tip clearance gap is the rotational-to-stationary leakage flowpath that detracts from the aerodynamic efficiency of the blade row. As shown in Fig. 2 for differing types of blade tip designs, stage efficiency is very sensitive to increased relative blade tip clearance with sensitivities from 1:1 to 2:1. This is a far greater issue with smaller turbines in which the effective tip clearance represents a larger relative percentage of the total annular flowpath height. It is also true that this efficiency derivative decreases as one moves from the high-pressure stages to the low-pressure stages of the turbine. An increasing tip leakage, sometimes called over-tip-flow, decreases the amount of work extracted from the hot gases, which in turn increases the engine exhaust gas temperature (EGT) for a given TRIT. The increase in exit temperature of an engine with time is a direct indication of the operational degradation of the engine. At a predetermined EGT setpoint for any engine, also known as the EGT margin relative to the initial new or serviced condition, the engine is either turned down in power or removed from service for maintenance. The losses associated with turbine blade tip degradation, aerodynamic and thermodynamic, typically account for as much as one-third of this EGT margin, hence the high attention paid to blade tips. The increase in aerodynamic loss and decrease in EGT margin is roughly linear with the increase in physical blade tip clearance.

As will be shown in subsequent sections, the aerodynamics surrounding the blade tip region are highly three dimensional, turbulent, and unsteady. Figure 3 shows a generic time-averaged static pressure distribution around an airfoil blade tip section with and without a tip clearance present. The pressure distribution, although similar to that of the airfoil without a tip clearance, is modified by the general tip leakage flows shown here. The overall pressure profile is the main driver for tip leakage flows and will change as the clearance changes. Even in the simplest case of a base-loaded turbine at essentially constant cycle conditions, the aerodynamic loading, and, consequently, the thermal loading, will change as a function of time on all tip surfaces. This change is manifested in the gradual loss of the blade tip material due to oxidation and erosion, in the absence of other more accelerated loss mechanisms such as tip rubbing. The aerothermal characteristics will change as the tip material is lost and the tip clearance opens. In general, increases in blade tip clearance will lead to higher tip region heat loading, which in turn tends to accelerate the tip loss rate with time. This degradation and failure of blade tips constitutes about one-third of total high-pressure turbine (HPT) failures, where failure is defined as the loss of the part from service inventory (unrepairable), or the accelerated degradation of the efficiency/output in service. In low-pressure turbines (LPT) that are usually uncooled, failure may more commonly be manifested in terms of simple oxidation and/or creep deformation without much change in the aerothermal characterisitics. The bottom line here is the cost in terms of maintenance cost per hour or cost of electricity.

The HPT turbine blade is typically also a highly complex heat exchanger. The blade tip is an aerodynamically shaped, structural, and most often internally and externally cooled complex fin heat exchanger. The degree of complexity depends on the turbine design functional requirements (mission, environment, cost, etc.). In highly cooled designs, a mixture of internal convective and/or impingement cooling with external film cooling is used to assure adequate tip life between repairs. The cooling techniques employed are dependent on the aerodynamic shape of the tip, the internal cooling circuit of the blade, and the choice of overall tip design, that is, unshrouded or shrouded. The function of blade tip cooling is to maintain the aerodynamic efficiency of the tip (shape and sealing) with the least possible usage of cooling fluid. Because of the tip profile conditions shown in Fig. 3, blade tip cooling must be distributed in nature, act with the complex flowfield, and if possible not degrade as conditions change. This is a substantial challenge, and as subsequent sections will show, one that has yet to be fully realized.

One further function of the blade tip must be made clear. As the tip provides the rotating-to-stationary interface for the contained working fluid, it must also be able to withstand the conditions associated with a tip rub against the stationary shroud/casing. A tight tip clearance provides higher aerodynamic efficiency, but also leads to a higher probability of rubbing. In the event of a tip rub, blade tip material may be lost or deformed. The blade tip region should be designed to maintain its other functions with minimal efficiency loss after a rub has occurred. The blade tip should also be repairable after some degree of material loss or damage, rather than being scrapped.

Blade Tip System Design Aspects

At a first glance, the turbine blade tip regions may appear to be isolated locations in the overall design, residing at the ends of the blades and only requiring local design attention. This is the inside-looking-outward viewpoint as though the viewer were riding on the blade tip. Taken from the outside-looking-inward viewpoint of the viewer riding on the stationary frame, the blade tip is actually a very highly sensitive region that reacts to the entire turbine system design. It is one of two meeting points of the rotating system and the stationary system, the other being the bearings, which should coexist as closely as possible without actually coming into contact. Effective and efficient turbine blade tip design has a distinct and important place in the overall turbine engine operation. High impact and high payoff are placed on blade tip survival because it affects so many key or critical parameters. In designing blade tips, both cooled and uncooled, for proper operation within the larger turbine system, one must consider the following major factors (in no particular order):

1) Stage and turbine aerodynamic efficiency are greatly affected by the blade tip design in terms of the resulting effective leakage clearance. The effective clearance, which may also be thought of as an effective overall tip discharge coefficient, is determined not only by the tip geometry, but also by the tip aerodynamic distribution, injected cooling flows, tip sealing arrangement, rotational speed, shroud surface treatments, and much more. As a first estimate, each stage can be thought of as having an isolated tip region aerodynamically, but the reality of multistage turbines is that all stages must be designed together to obtain maximum benefit. Another important aspect of the aerodynamic efficiency directly tied to blade tips is the mixing loss associated with the tip leakage flows as they combine with the high momentum suction side passage flow.

2) Stage thermal efficiency, and then also overall turbine efficiency, is strongly affected by the amount of chargeable cooling air used to maintain blade tip integrity and life. In highly cooled HPT blades, the tip region alone may account for as much as 20% of the total blade cooling flow.

3) Bulk material temperature limits must be considered for the entire blade structure. Whereas the tip region is generally not subject to the same limitations as the rest of the blade in this respect, the tip design does influence the resulting bulk temperatures of the lower blade sections through the overall cooling design. The tip may also present enough weight to require lower bulk temperatures in the main blade sections to avoid creep rupture issues.

4) Maximum local material temperatures are typically a major concern for blade tips because these regions are the most difficult to cool. Temperature limits will be placed on the metal substrate, the bond coat, and the thermal barrier coating (TBC) to avoid, for example, excessive oxidation, high coating strains, and melt infiltration of surface deposits, respectively. This can be thought of as the material system design.

5) Tip sealing methods vary widely, as will be shown later, but all methods attempt to reduce the effective tip clearance. The type of sealing arrangement is intimately tied to the other system design aspects. In many ways, the sealing design is the result of which system design parameters are given the most emphasis.

6) Casing out-of-roundness, that is, noncylindrical, will be transmitted through the structure response to the hot-gas flowpath roundness bounding the blade tips. This leads to nonuniform tip gaps around the circumference and potential tip rubs.

7) Shroud segment variation, such as bowing, can result from the thermal gradients present in the design, again leading to nonuniform tip gaps either radially and/or axially.

8) Approaching and leaving disturbances in the flow around blade tips can affect both the aerodynamics and the cooling. Approaching disturbances are most notably associated with the wakes and shocks being shed from the upstream vane row, which to some degree must influence the tip flow and heat transfer by the introduction of unsteady effects. Approaching and leaving disturbances may be encountered in tip designs that involve shroud recesses and axial flow gaps between the stationary shrouds and attached tip shrouds.

9) Gas temperature profiles are the result of the particular combustion system design, the operational point, and mixing through the subsequent stages. The radial gas temperature profile may have severe impact on the blade tip, both in respect to the temperature field itself and the pressure distribution. Stronger radial flows may bring hotter gases to the blade tip than desired, whereas gas temperatures may drive strong material thermal gradients and cause lower cooling effectiveness.

10) Aeromechanics must be considered in the overall blade structural design, and the tip region must be included in this response.

11) Stresses, both mechanical and thermal, are key in turbine blade survival. Blade tips must typically deal with very high thermal stresses locally. Higher cooling effectiveness in the tip can alleviate thermal stresses, but must be weighed against the cost to the cycle efficiency. As noted earlier, the blade tip design will influence the weight distribution in the entire blade, which must then be dealt with in the allowable stresses, as well as the low cycle fatigue (LCF) and high cycle fatigue responses. This effect will also be transmitted into stress requirements for the blade shank, dovetail, rotor disk posts, and the rotor disk.

12) Operating conditions must be considered at various limiting points in the engine cycle because these change the gas and coolant flow rates, temperatures, and pressures. A blade tip design focused solely on steady-state takeoff conditions may not be well suited for cruise conditions. A balanced or optimized cycle design must be sought.

13) Transients play a major role in the durability and life of any effective blade tip design. The relative displacements, radial and axial, of the rotor and stator systems during various transients will determine the ultimate steady-state operating clearances, as well as the potential for detrimental interference.

14) Durability is desired for both the blade tip and the opposing shroud as a system. In the long term, durability may be associated with oxidation, whereas in the short term, durability is a matter of

survival in the face of tip rubs (intentional or unintentional), plugged cooling holes, and thermal stresses.

15) Materials and material loss must be planned in blade tip design. Blades and blade tips are not automatically designed with the highest temperature capability material, nor the highest strength material. The tip material may be different from the rest of the blade. The compatibility of the tip and shroud materials must be considered, for example, if a highly abrasive shroud should damage a relatively weak tip material.

16) Cumulative damage of blade tips is typically experienced in certain characteristic locations in each design type. Uniform damage or material loss is not the general rule. The change in tip geometry with characteristic damage and loss will alter the aerodynamics and heat transfer, usually leading to accelerated loss.

17) EGT is directly and strongly affected by blade tip clearance. Any improvement in effective tip sealing will preserve valuable EGT margin.

18) Cost of new parts and cost of repair depend on the complexity of tip design.

19) Blade weight impacts the blade root stresses, LCF life, and blade creep. This is not limited to a simple matter of centrifugal stresses, but can also have severe effects on the overall aerodynamic design, changing the reaction and work of the stage.

20) Thrust bearing location and bearing housing distortion affect axial motion and disk sag, which in turn are transmitted through the rotor to the blade tip potentially creating larger clearances on one side of the turbine and rubs on the other side.

21) Rotor and stator systems should be thermally matched to minimize variations in blade tip clearances during transients. Active clearance control systems can aid in this goal by providing fast thermal response of the shroud radial location.

22) Blade tips are commonly at least partially damaged or worn in the course of operation. The ability or inability to repair blade tips becomes an important factor in lifetime cost. The complexity of a blade tip design impacts the decision to provide more or less cooling to balance the cost of repairs. Unrepairable blade tips result in scrapped blades.

Whereas this summary of system design aspects may appear quite detailed and daunting for such a relatively small region of the turbine, there is one requirement that exceeds all others: The blade tip system design must never cause such severe damage as to liberate blades or pieces of blades in operation. As in the other interacting system relationships within the turbine, prior design and operational experience must guide and temper improved designs.

Transient Operational Requirements

Transient cycle conditions represent the most severe test for turbine blade tips in most respects, structurally, aerodynamically, and thermally. Figure 4 presents an example of a typical aircraft engine HPT transient from a cold start, through takeoff and climb to steady-state condition at cruise. Also shown is the possible transient condition during a trip, or engine shutdown, from steady state. In Fig. 4, the ordinate represents both the turbine speed and also the radial growth ΔR of the rotor and stator portions bounding the blade tip. Because the speed of the turbine spools up very quickly, immediate response is seen due to the centrifugal growth of the rotor disk and the blade. During this short time of initial rotor growth, the stator lags in response because its growth is only due to thermal expansion. The rate of thermal growth of the stator then begins to overtake that of the rotor. The stator is capable of maintaining higher bulk material temperatures, so that it is cooled to a lesser degree than the blade, and hence, thermally grows outward faster. The blade is highly cooled, so that its thermal growth is less than that of the stator. In the last phase of overall radial growth, the slowly responding thermal mass of the rotor disk expands to its steady-state condition. The overall radial growth of the rotor disk and blade is intended to exceed that of the stator slightly, such that the tip clearance gap closes to an efficient tight position. From this transient history, it can easily be seen that off-design conditions, such as overspeed or reduced cooling flow, can quickly result in blade tip rubs. The postulated trip scenario shown in Fig. 4 highlights a key operational danger in engines. When the rotor speed is suddenly dropped, the centrifugal growth factors of the rotor and blade are reversed quickly, whereas the thermal growth of the stator only slowly responds at first. However, it is the large thermal growth of the rotor that lags the most in the system. The stator thermally contracts inward before the rotor and blade can move away, resulting in potentially severe and very damaging tip rubs. This is the critical time period for restart of the engine to avoid such harsh interference. A safety factor can be built into the design to allow for larger clearances at cruise conditions such that rub events are less likely, but this has obvious negative efficiency implications. Another alternative may be the design of stator systems with faster thermal response or controlled response.

From the perspective of the blade tip clearance response during transients, Fig. 5 shows the rotor and stator growths relative to the initial cold clearance of the blade tip. Progressing through a similar takeoff, climb, and cruise transient, the tip clearance is seen initially to close, then open slightly again, and then close up to the hot steady-state magnitude. On a trip condition, the tip clearance first opens somewhat, and then closes as the stator response catches up. If a hot restart is then initiated, the blade thermal growth and rotor centrifugal growth rapidly respond again because the rotor has not had time to cool down. Here again, because the stator cannot respond as quickly, a potential exists for a blade tip rub event. In this case though, the rub is generally slight because the entire system moves back to its steady condition.

The basic blade thermal design is based primarily on the conditions present during the hot cycle point of the engine, for example, hot day takeoff conditions for an aircraft engine. However, for the balance of peak thermal efficiency with minimal usage of valuable compressed air, the blade tip cooling will be designed such that a nominal bulk material (metal or ceramic) temperature is maintained under the most common running condition, while also limiting the maximum local material temperatures experienced under the peak load conditions. For aircraft engines, the nominal operating condition is cruise and the peak condition is takeoff or thrust reverse. For land-based power turbines, the nominal and peak conditions are the same at the base load operating point of 100% output. Blade tip life

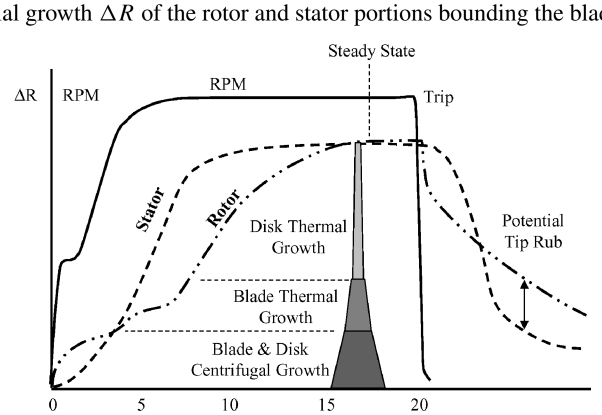

Fig. 4 Transient stator and rotor system radial growth curves.

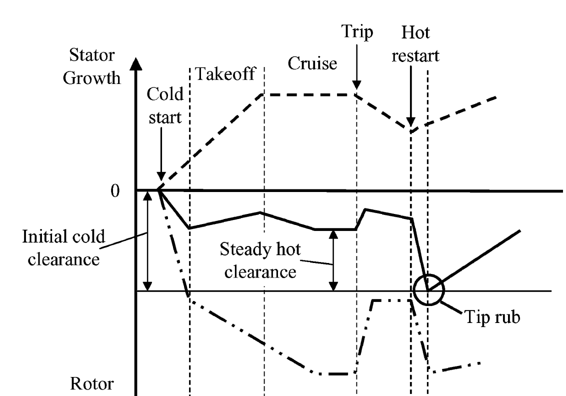

Fig. 5 Blade tip clearance variation with operating condition.

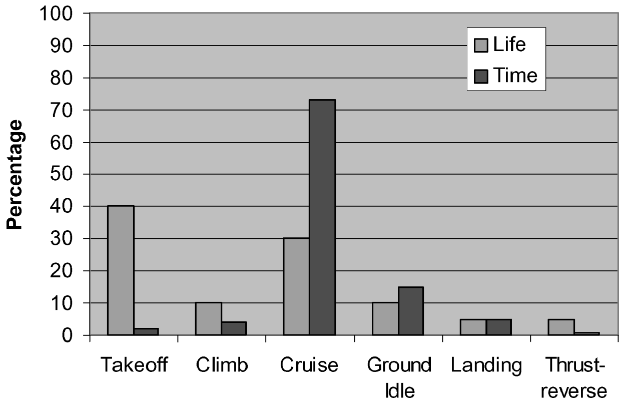

Fig. 6 Example of mission mix effect on blade tip life.

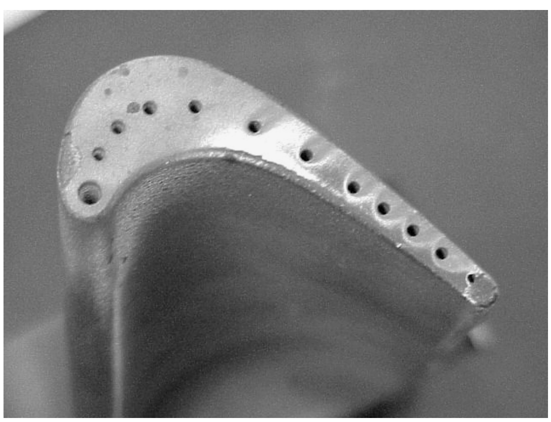

Fig. 7 Power turbine flat blade tip example.

may be treated in design as the cumulative effect of varying thermal loads and cooling from the total time spent under the different transient and steady load conditions. Figure 6 shows this concept in terms of a simple pareto diagram using the major elements of aircraft engine operational time. Keep in mind that the time exposure under takeoff conditions is the smallest; nevertheless, the effects on blade tip heat transfer and life may be the most severe. This is analogous to the use of a cumulative damage model in mechanical fatigue of materials. Figure 6 is an example only because the specific pareto may be very different for various system designs.

Blade Tip Design Philosophies

When the design and operational requirements discussed in the preceding sections are considered, it should come as no surprise that several different forms of blade tips have been devised and used successfully. No single solution satisfies all requirements in an optimized manner, just as there is no single design of turbine engine, nor a single mission definition for all engines. Each blade tip design has its advantages and disadvantages. There are three major design philosophies in practice today: 1) flat cylindrical blade tips, 2) cylindrical tips with squealer sealing rims, and 3) attached tip shrouds used with either cylindrical or flared blade tips.

Flat cylindrical turbine blade tips are no longer very commonplace, but are still in use, even in some modern designs. Figure 7 shows a photograph of an older vintage heavy frame turbine first stage blade tip. The tip is flat and cylindrical (common outer radius along axial direction). This older style blade uses straight cooling passages from hub to tip, discharging the coolant out the tip radially. In one respect, this manner of coolant discharge provides a sort of fluid resistance to hot-gas tip leakage flows, but this was not the intent of the design. By the time the coolant exits the tip in such a design, its thermal potential for cooling purposes has been nearly used up. In modern flat blade tip designs, complex internal cooling passages deliver tip cooling through film holes near the tip on the pressure side of the airfoil, as well as on the tip surface. Note that

Fig. 8 General Electric Aircraft Engines HPT blade with squealer tip.

at least one tip hole is required per cooling circuit within a blade to allow for dust purge by centrifugal action. Such flat blade tips rely on good thermal matching of the rotor and stator systems to keep tip clearances tight. Because no physical leakage resistance sealing mechanisms are employed in flat tips, these designs typically result in the lowest tip aerodynamic efficiency due to relatively high leakages. The higher leakage flows can also lead to higher heat loads on the tip, emphasizing the importance of sufficient film cooling in such designs. Flat blade tips may also be more susceptible to damage if and when they do rub, because there is no sacrificial buffer material present. One positive aspect of flat tips is that there are no extended surfaces to be cooled, and hence, the cooling design becomes very simple. As long as the flat tip integrity can be maintained, its performance at a given cycle point condition remains unchanged with time.

A cylindrical blade tip with a perimeter seal strip, known as the squealer rim, is the most prevalent design in practice today within HPT turbines. Figure 8 shows an example of an advanced, highly cooled HPT turbine blade with squealer tip surrounding the tip cavity. The squealer rim is a natural radial extension of the aerodynamic surface of the airfoil. In most designs, this squealer rim extends as far aft on the tip as possible, until the trailing-edge thickness requires closure of the perimeter. In some designs, the trailing-edge portion of the rim is left open on the pressure side, allowing tip cavity cooling air to exit over the suction side rim as a sort of film cooling. The function of the squealer rim is as a simple two-tooth labyrinth seal. Tip leakage gas is forced to contract between the pressure side rim and the shroud, then expand into the tip cavity defined by the entire squealer rim, and then contract again to pass the suction side rim restriction before expanding into the main flow. This is a very simplistic picture of the tip seal, but captures the intended function. A squealer tip running with a very tight clearance can be a very effective seal aerodynamically. Likewise, a squealer tip running with a very open clearance can be a very poor seal. The rim thickness is minimized to reduce weight and actually has a negligible role with respect to the aerodynamics. The rim material is allowed to rub against the shroud in transient situations because this material is sacrificial. Minimal, moderate, or even heavy tip rubs in such designs will not compromise the integrity of the remainder of the blade. The difficulty in a squealer rim design lies in cooling of the rim to prevent loss by oxidation and erosion. The rim represents an extended fin that sometimes requires many pressure side film cooling holes, and also interior tip cavity cooling holes, to provide adequate cooling coverage, as shown in Fig. 8.

Fig. 9 Rolls–Royce Trent 800 HPT blade after 8299 h (cleaned) (reproduced by permission of Rolls–Royce, plc.).

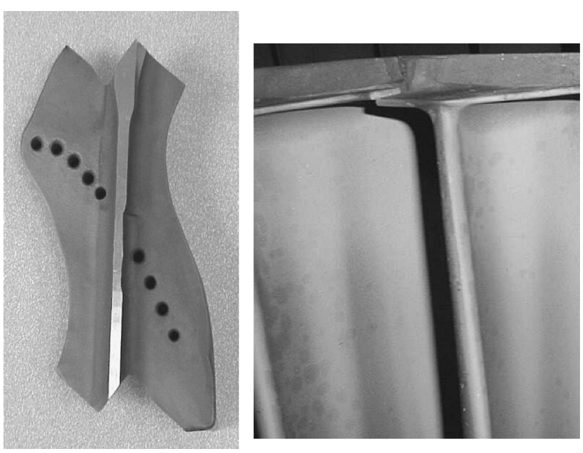

Fig. 10 Scalloped attached tip shroud for power turbine.

Turbine blade tips using attached tip shrouds are found in some HPT blade designs, but are more common in most LPT blades. Figure 9 shows an example of a HPT blade with a highly cooled, attached tip shroud. The tip shroud is in essence a shroud, or bounding outer radius flowpath, that moves with the blade tip. There is a stationary shroud casing outside of this tip shroud. The attached tip shroud allows the hot-gas tip leakage path to be decreased from the full tip profile to only seal gaps between the tip shroud and the casing shroud. The primary gaps are located at the forward and aft circumferential edges of the shroud and radially at each seal tooth on the top of the tip shroud. The blade tip of Fig. 9 employs two circumferential and two axial seal teeth on the upper surface. This design acts as a multitooth labyrinth seal arrangement with much more complex flow restrictions and interactions. Of all of the current blade tip designs in use today, the tip shroud has the lowest aerodynamic loss when properly implemented. Figure 10 shows an example of a simple LPT blade tip shroud. The function of this design is the same as that of the HPT tip shroud, but the complexity is far less, using only a single seal tooth. A few other aspects of these designs are worth noting here. The HPT tip shroud is certainly a heavier blade tip than the squealer or flat tip, so great attention must be paid to stress. This also requires a much more complex cooling system, not only because of the geometry, but also to maintain material temperatures for acceptable stresses. The tip shroud of Fig. 9 is a flared tip, meaning that the mean radius of the tip increases along the axial direction. This requires the casing shroud to do the same, which then allows the axial gaps to be used as sealing faces. This practice has the advantage of greater sealing, but also means that the tip and shroud system are more sensitive to axial movements of the turbine rotor and stator. The tip shroud of Fig. 10 is a cylindrical tip, but still recessed into a casing shroud. Weight can have enough of a penalty on stress (creep) that the tip shroud may require scalloping. Scalloping is the practice of removing tip shroud material to reduce weight, while still attempting to maintain the aerodynamic sealing function. The tip shroud of Fig. 10 is moderately scalloped, as shown by the curved edges. This material removal will decrease aerodynamic sealing capability and reduce the airfoil lift coefficient. A final note concerning the function of attached tip shrouds is that they physically abut one another along the circumferential outer radius. This connection, though not sealed or made permanent, provides some damping of blade vibrational characteristics. This feature of tip shrouds can be an advantage for the aeromechanics of the blade row. The tip shroud of Fig. 9 shows a straight shroud-to-shroud interface, whereas that of Fig. 10 has a so-called Z-lock shape to assist with axial displacements.

Although the three major designs of turbine blade tips have been introduced here, there are many variations on each of these designs, including hybrid designs that attempt to capture desirable characteristics of each. In every specific blade tip design, tradeoffs must be made between the many factors to determine the most feasible approach. Here again, the most important single factor that will determine the direction of design is operational experience. In some missions and engines, the choice will be obvious. For example, blade tips in a short mission expendable engine will be flat and cylindrical for simplicity and cost reasons. Uncooled blade tips in LPTs, and also those in steam turbines, will be designed with tip shrouds for best aerodynamic performance. Turbine engines requiring long life and high firing temperatures involve a proliferation of possible blade tip design philosophies.

Aerodynamics

To gain an appreciation for turbine blade tip aerodynamics, it is first instructive to examine the sources of losses within a turbine stage. Figure 11 shows the breakdown of loss components for several differing experimental turbines per the summary of Booth.[3] Included in the overall losses are the vane (stator) profile, endwall, and secondary flow losses, which account for about 25% of the total stage loss. The remaining 75% of stage losses are associated with the blade (rotor). Of these losses, the most significant is that due to tip clearance, which on average accounts for roughly one-third of the total stage loss. When the turbine stage aerodynamic efficiencies at the top of Fig. 11 are noted, the tip clearance loss represents about 4% of stage efficiency, a very substantial amount. To understand the specifics of what contributes to this tip clearance loss, the detailed tip region flows must be described. Figure 3 showed the general pressure distribution around a flat blade tip with and without clearance, showing the driving pressure potentials for the leakage flow. Using a two-dimensional linear

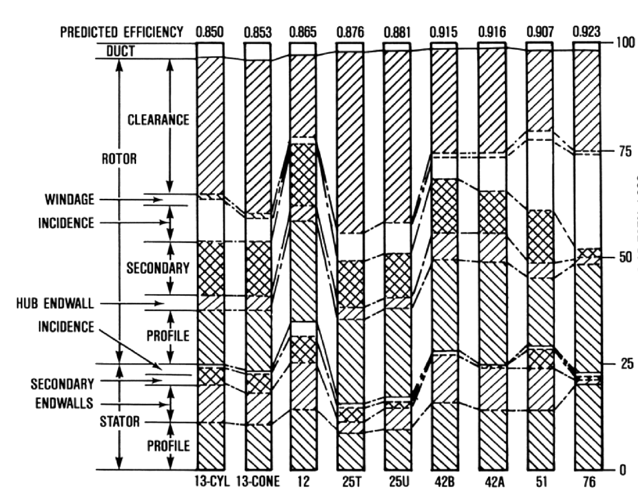

Fig. 11 Efficiency loss contributions for several single-stage turbine rigs (reproduced by permission of Von Kármán Institute).

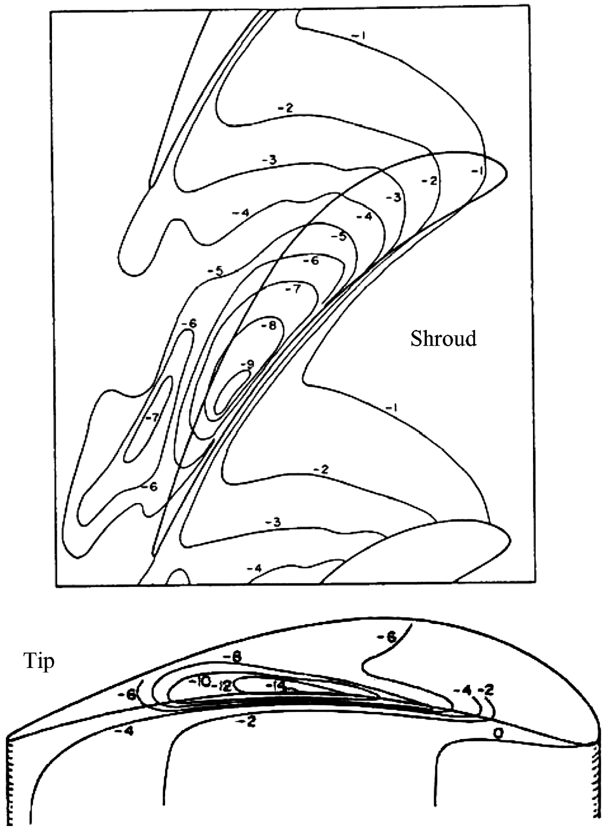

Fig. 12 Flat blade tip and shroud pressure distribution contours (reproduced by permission of the American Society of Mechanical Engineers).

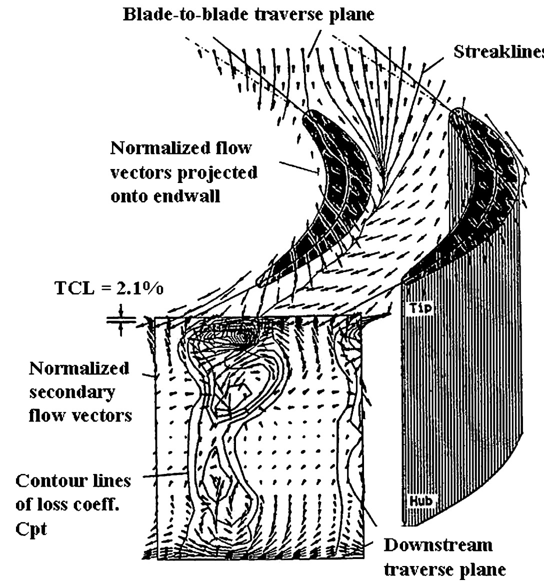

Fig. 13 Blade cascade flow vectors and exit loss coefficient contours (reproduced by permission of the American Society of Mechanical Engineers).

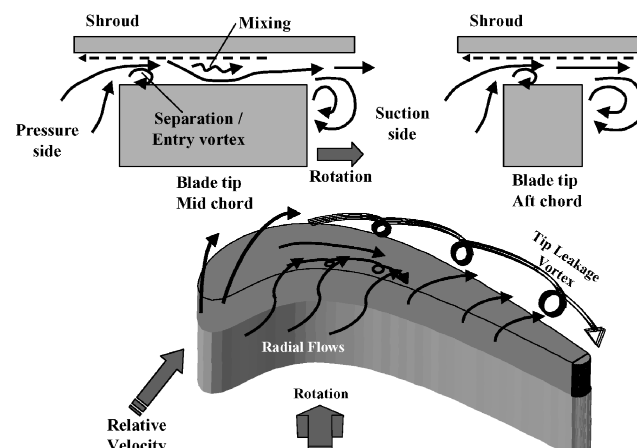

Fig. 14 Schematic of flat blade tip leakage flow characteristics.

blade tip cascade (flat tip), Bindon[4] measured the detailed blade tip and shroud pressure distributions for several tip clearance gap magnitudes.

Figure 12 shows the contours of static pressure coefficient for a tip gap of 2.5% of blade span. These contours show the clear presence of an entry separation region on the tip pressure side in the mid-to-aft chord location. This feature is also reflected in the shroud contours. The zone of highest tip leakage flows emanates around this region, driven by the pressure-to-suction side overall aerodynamic profile. The leading-edge region of the blade tip is relatively calm, but still has lower magnitude leakages.

Yamamoto[5] performed five-hole pitot probe measurements in a linear blade cascade for a plane through the tip clearance and passage region, as well as a plane downstream of the trailing edge. Figure 13 shows the typical flow vectors and streaklines for the nominal incidence angle of 7.2 deg and a tip gap of 2.1% blade span. The blade tip is flat. The flow vectors exhibit low velocities of leakages in the tip leading edge region, with higher magnitudes in the midchord to aft regions. Leakage flows are typically angled from the regions of highest source pressure to lowest sink pressure. The secondary endwall flows in the passage show direction from the pressure side to the suction side. The tip leakage vortex created by the interaction of the gap flows with the mainstream passage flow is initiated in the region of highest airfoil curvature, where leakage flows first enter the mainstream, and grows from there downstream. The downstream radial plane shows both flow vectors and contour lines of loss coefficients. The tip vortex aft of the trailing edge is intense and contained within the upper 10% of the span. The vortex has actually induced a secondary counter-rotating vortex in the mainstream flow, which is larger and penetrates quite far into the passage both radially and circumferentially. Yamamoto also studied the effects of off-incidence angles (negative angles of attack) that simulate conditions at other cycle points. It is common that blade tips may experience very wide variations in flow incidence angles within the cycle, most especially for aircraft engines. The tip leakage vortex is still roughly unchanged for such conditions, but flows over the tip and induced secondary flows in the passage can be greatly altered. Note that the studies[4,5] were set in stationary cascades without relative motion effects of the shroud surface. In a rotating turbine rig test, Morphis and Bindon[6] found that such relative motion effects did not strongly influence the resulting pressure distributions.

With the general flow characteristics surrounding a flat blade tip elucidated, an overall schematic of the basic tip aerodynamics can be established as in Fig. 14. This general view shows the leading edge, midchord, and aft region flows as described from the studies noted earlier. Also shown are two cross-sectional views through differing portions of the blade tip, one in midchord and the other aft. Figure 14 demonstrates that the local blade tip thickness, as well as the local flow conditions, will alter the existence of such effects as the separation vortex, the extent of mixing within the tip gap, and the jetting of leakage flow into the mainstream. The space shuttle main engine computational fluid dynamics (CFD) prediction of Ameri and Steinthorsson,[7] shown in the particle traces of Fig. 15, clearly illustrates both the tip entry separation and the tip leakage flow vortices, the former being composed primarily of radial flows entering the gap (light lines), and the latter of bulk gap flows (dark lines) mixed with the postseparation flows. Here again, the streamwise growth of the leakage vortex is apparent from the high

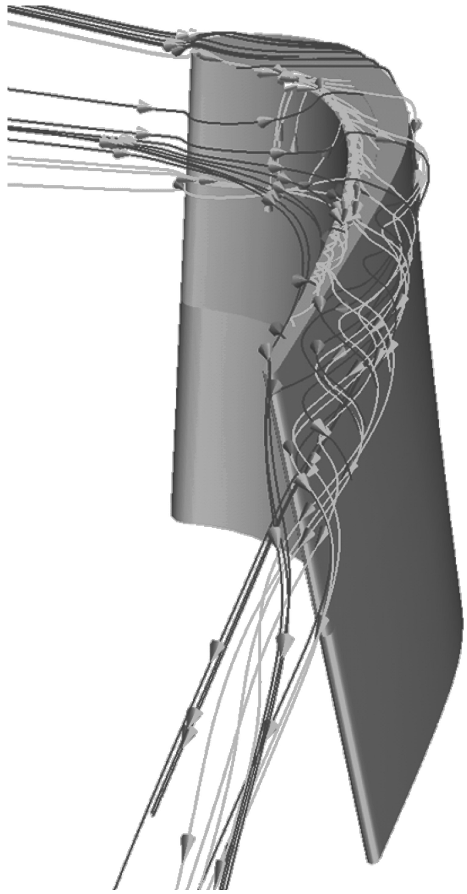

Fig. 15 Computed prediction of flowfield streaklines for flat blade tip (reproduced by permission of the American Society of Mechanical Engineers).

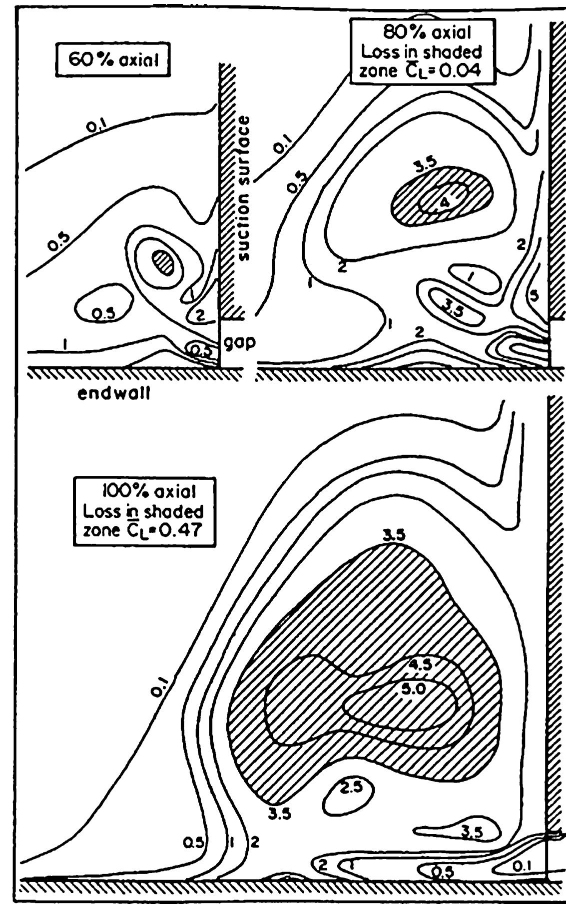

Fig. 16 Growth of suction side tip leakage vortex strength (reproduced by permission of the American Society of Mechanical Engineers).

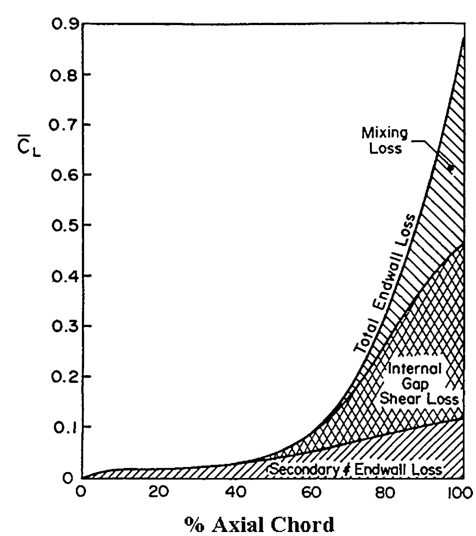

Fig. 17 Contributions to total endwall loss development as function of axial location on tip (reproduced by permission of the American Society of Mechanical Engineers).

curvature portion of the airfoil suction side. Bindon[8] captured this vortex growth quite well in his linear blade cascade tests. Figure 16 shows the initiation of the vortex just past 50% axial chord location, with subsequent growth and increase of loss coefficients at 80% and 100% axial chord. Bindon further used such data to quantify the contributions to loss due to endwall secondary flows, internal tip gap mixing, and leakage vortex mixing. Figure 17 shows the results for his airfoil shape and loading. The endwall and secondary flow losses are those present with no tip gap clearance, contributing along the entire axial chord of the passage. The internal gap losses due to separation, shear flows, and mixing do not begin to contribute until about 40% axial chord, but make up almost one-half of the total losses. The tip vortex (mixing) losses are seen to begin at about 60% axial chord and rise significantly as the internal gap losses diminish (due to the thin trailing edge). This view of losses is quite generic, but applies well to most flat tip cases with some adjustment for specific aerodynamic shapes and loading. The same generic view holds fairly well for squealer and tip shroud designs also. It is immediately apparent that the total blade tip loss for any design may be reduced by attacking its two main contributors, internal gap loss and vortex mixing loss.

Most blade tip designs do not seek to reduce these main aerodynamic losses through detailed manipulations of the interacting flows and structures, but instead take the more straightforward approach of simply reducing the total tip leakage flow. This is accomplished through sealing geometries and tight running clearances. Figure 18 shows the general flow features for three sections of a squealer tip. With the additional flow restrictions present, there are now separated flows (or vena contracta) on both squealer rims. The typical picture of tip cavity flow is one in which the flow expands into the cavity, reattaches on the cavity floor, and then contracts again to exit the cavity. In the process, recirculation regions are formed in the cavity corners. Figure 18 shows that this is not the case at every section because cavity aspect ratio is not constant along the tip. In the trailing edge, the tip leakage flows may stream across the cavity with little or no impediment. The flowfield in a squealer tip cavity is actually much more complex than even this simple view. As the CFD predictions of Ameri et al.[9] show in Fig. 19, the three-dimensional nature of the blade tip and airfoil pressure distribution will cause chordwise tip cavity flows that interact with the main pressure-to-suction side flows. Depending on the specific shape and design, such flows can either help or hinder the function of tip sealing.

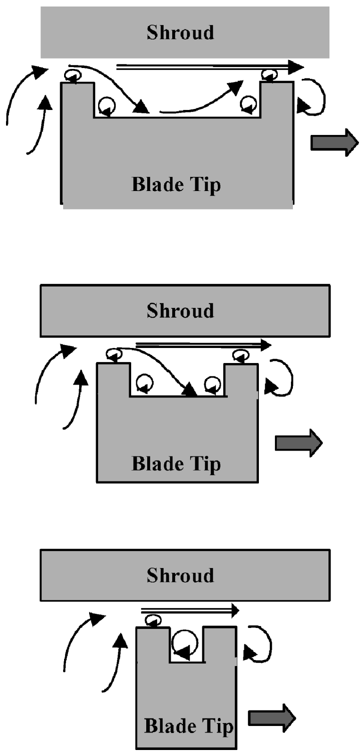

Fig. 18 Flow characteristic changes at varying locations in squealer tip.

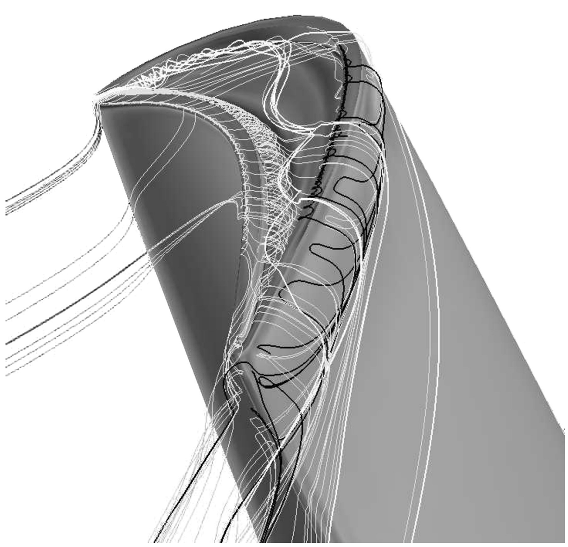

Fig. 19 Computed prediction of flow field streaklines for squealer blade tip (reproduced by permission of the American Society of Mechanical Engineers).

In the case of attached tip shrouds, the number and complexity of flow restrictions increases. The simplest geometry using a single tip seal tooth is shown in Fig. 20. In proper operation, the inboard surface of the tip shroud should line up with the casing shroud surface before and after the casing recess to form a continuous hot-gas flowpath. The flow must then negotiate a forward interface slot to enter the gap region, a seal tooth restriction, and then the aft interface slot to reenter the mainstream. Typically, the casing shroud inside the recess will also be treated to increase flow resistance, for example, through the use of a honeycomb structure. Not shown in Fig. 20 is that the rotating blade tip will cause a significant circumferential flow within the recess gap, especially in the forward section, which will further aid in the reduction of leakage. Figure 21 shows the much more complex design for an HPT tip shroud with axial flaring. Here two circumferential seal teeth (fins) are used in the forward region of the tip shroud, which are engaged with a stepped recess in

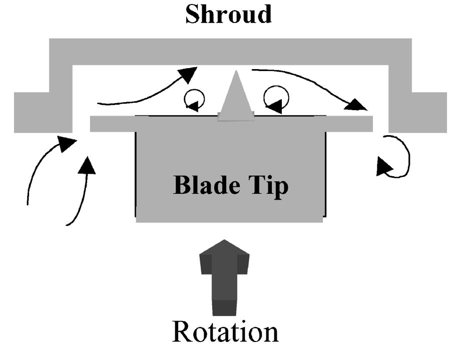

Fig. 20 Recessed tip shroud flow schematic.

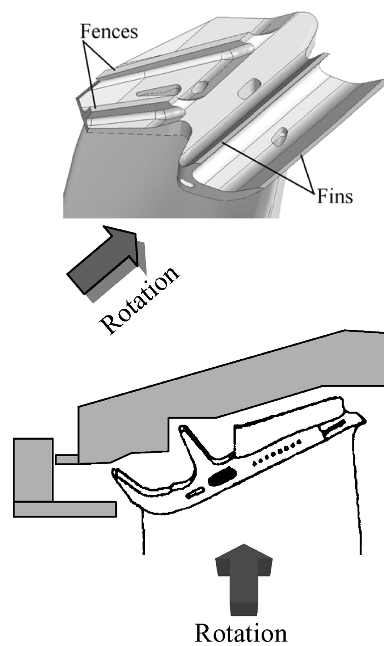

Fig. 21 Complex sealing structure for HPT tip shroud (reproduced by permission of Rolls–Royce, plc.).

the casing. In addition, two chordwise seal teeth (fences) are used in the aft region to help seal the region of highest overall pressure potential across the tip. Together these devices form a formidable tip leakage resistance network. Note that the aft tip fences will also act to extract useful work (blade lift), as will some portion of a squealer tip rim. Tip shrouds have at least one negative aspect with respect to aerodynamics though, in that they are much more sensitive to variations in alignment or positioning. It is easily seen that a tip shroud that protrudes into the hot-gas path will provide a mainstream blockage that not only reduces the overall blade efficiency, but may also exacerbate tip leakage. Likewise, a recessed forward tip shroud may lead to a protruding aft casing surface, which again can reduce stage efficiency.

The noted blade tip designs have common features in aerodynamic efficiency. As in Fig. 2, all designs increase in efficiency as the effective tip clearance is reduced. As the tip clearance is reduced to less than 1% of blade span, the squealer tip and tip shroud designs become roughly equivalent in efficiency. Differences in performance come as blade tips degrade under service conditions and depend on the design's ability to resist such degradation.

Heat Transfer and Cooling

Whereas the function of a turbine blade tip is to reduce as far as possible the aerodynamic losses at the rotational-to-stationary interface, this function cannot be maintained without careful attention to the thermal boundary conditions and the cooling design. This is of course only true for cooled blade tips such as those of

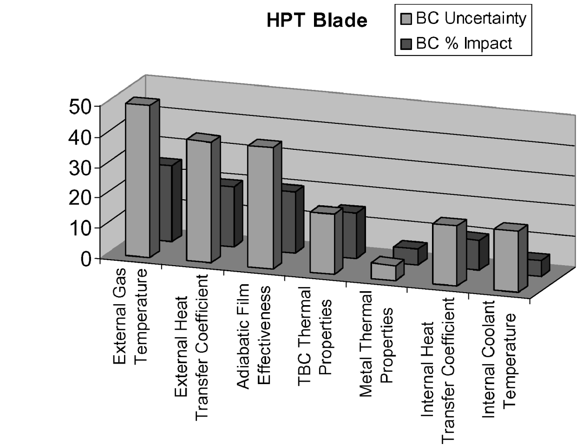

Fig. 22 Pareto of thermal boundary condition impact and uncertainty.

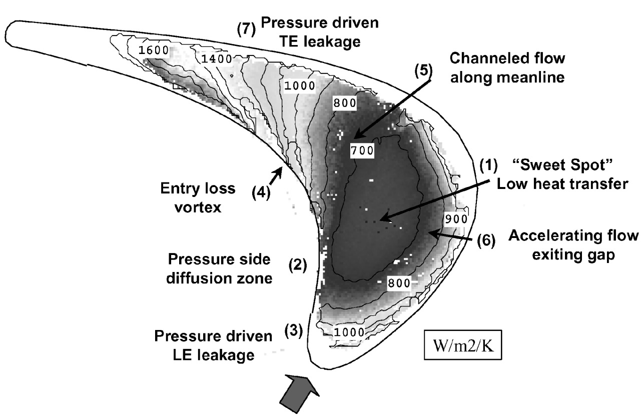

Fig. 23 Flat blade tip heat transfer coefficient distribution (reproduced by permission of the American Society of Mechanical Engineers).

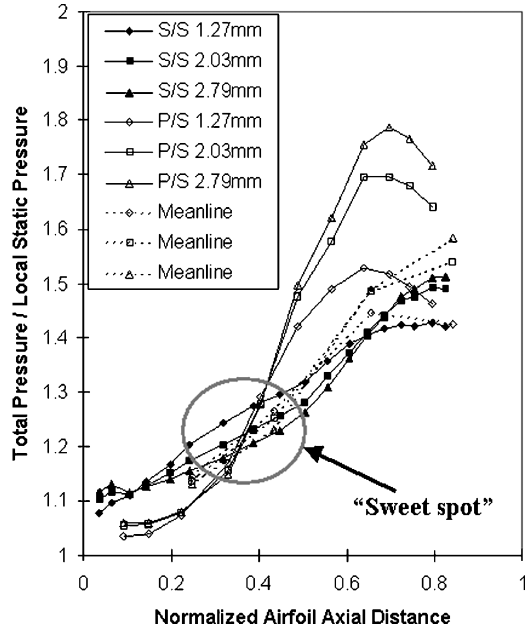

Fig. 24 Flat blade tip pressure field corresponding to Fig. 23 (reproduced by permission of the American Society of Mechanical Engineers).

the HPT blades. Uncooled blades, usually those of the LPT, reside at the gas temperature. Thermal boundary conditions of concern to the blade tip include the following: 1) mainstream hot-gas temperatures, 2) tip surface heat transfer coefficient distributions, 3) tip surface film effectiveness distributions, 4) pressure side sink flow heat transfer coefficients and film effectiveness, 5) suction side source flow heat transfer coefficients and film effectiveness, 6) seal surface heat transfer (rims, seal teeth, etc.), 7) internal blade tip heat transfer coefficients, 8) internal coolant temperatures, and 9) blade tip substrate and protective coating, for example, TBC, thermal conductivity.

A pareto of the relative impact and uncertainty of each of these boundary conditions on the resulting blade tip temperatures, that is life, is shown in Fig. 22. This pareto is one example of many, but indicates the typical relative importance of the thermal boundary conditions for most HPT blade tips. A key aspect of blade tip heat loading and cooling is that no two aerodynamic or system designs are completely alike. As will be seen later, there are similarities in all blade tip heat transfer characteristics, but the differences between designs can be subtle or striking. There is, consequently, no single best blade tip cooling design that may be applied to all engines, only design philosophies, which can be broadly grouped by function and form. Also, within any particular philosophy there will be engine-to-engine variations, as well as blade-to-blade variations, which can cause significant deviations from nominal heat loading and cooling conditions. This aspect of blade tip cooling can be thought of as design for reliability applied to the blade or turbine and applies to both the overall cooling design as well as the individual details of local design features.

Beginning with the basic heat transfer characteristics for a flat blade tip, Fig. 23 shows the distribution of heat transfer coefficients measured by Bunker et al.[10] for a first-stage blade tip in an industrial heavy frame turbine design. This distribution was measured in a linear, high-speed cascade with sharp edged tip, an inlet Mach number of 0.4, overall pressure ratio of 1.43 (atmospheric discharge), and clearance gap of about 1% of blade span. For comparison, Fig. 24 provides the local pressure ratio distributions on the tip surface (total pressure to local static pressure) for the same clearance gap, as well as increased and decreased clearances. Both measurements just inside the pressure side edge on the tip (P/S) and measurements just inside the suction side edge (S/S) are shown. Figure 23 is annotated to show the major flow regions and effects for this blade tip. The most striking feature in the heat transfer coefficient distribution is the "sweet spot" of lowest coefficients in the midchord region. This region is also highlighted in the pressure distributions of Fig. 24, where the pressures across the tip all come into approximate agreement, meaning that there is little pressure difference to drive leakage flows here. The leading edge of the airfoil tip ahead of the pressure side diffusion zone indicates leakage flow along the forward S/S region with higher heat transfer coefficients. The pressure side entry loss vortex is clearly observed in the concentration and turning of the heat transfer coefficient contours. From the region of 40–80% axial chord, the pressure differential across the tip is highest, leakage flow is high, and consequently the heat transfer coefficients are also highest in direct correspondence with the pressure distribution. Given the broad nature of this tip profile, there is also some channeling of leakage flow along the mean chordline from the sweet spot aft toward the suction side where the flow exits. To emphasize and contrast such heat transfer characteristics, Fig. 25 shows a second distribution of coefficients measured by Bunker and Bailey[11] for a differing, much narrower tip shape under virtually identical cascade conditions. All of the same features can be identified here, but some are extended and others muted by the differing aerodynamic profile. The sweet spot is relatively enhanced, whereas the entry vortex is more compressed compared to Fig. 23.

With the addition of almost any sealing features to a flat blade tip, the driving pressure distribution around the tip is altered, local leakage flows are redistributed, and so, too, the heat transfer coefficient distribution on the tip can be substantially changed in both magnitude and form. Fig. 26 shows tip heat transfer coefficients for several variants of simple tip seal features as obtained by Kwak et al.[12] In this study, a linear blade tip cascade was again used with a mild overall pressure ratio of 1.2 and a tip clearance of 1% blade span. The flat tip heat transfer coefficients are shown at the bottom of Fig. 26 (note the slightly higher scale for this case). Immediately apparent is that the addition of any tip seal changes the heat transfer coefficients

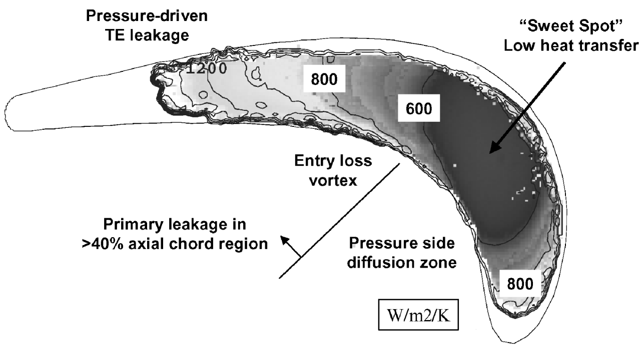

Fig. 25 Blade tip heat transfer coefficient distribution for narrow aeroprofile shape (reproduced by permission of the American Society of Mechanical Engineers).

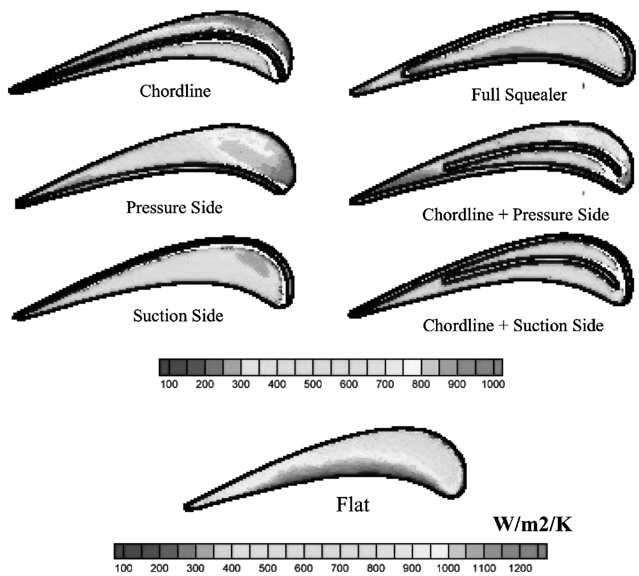

Fig. 26 Effect of various tip seal rim locations on heat transfer distributions (reproduced by permission of the American Society of Mechanical Engineers).

from a distribution peaked on the pressure side entry region to something else. This is the case even when the sealing rim is not on the pressure side. Whereas the sweet spot on the flat tip is located near the leading edge for this airfoil shape and pressure ratio, this is not the case for most seal geometries. In fact, the forced redistribution of flow by the seal rims can actually increase the leading-edge region heat transfer, a characteristic commonly not observed by computational predictions alone. The full perimeter squealer rim tip is seen to lower tip cavity heat transfer coefficients over most of the tip surface, most especially in the midchord and trailing-edge regions. Seal rims placed only on the pressure side, mean chordline, or in combination may tend to act as flow disruptors, or turbulators, leading to locally increased heat transfer coefficients over those of the flat tip. The very simple seal geometry using only a suction side rim provides the lowest overall heat transfer in this example. This positive effect is the combination of two features, effective tip leakage flow reduction by the suction side seal rim, plus the elimination of any tip entry flow disruption from a pressure side seal rim. This tip sealing geometry is not, however, common in practice because the use of a full squealer rim is generally thought to provide additional leakage reduction (better aerodynamic efficiency), especially in cases of material loss due to tip rubs. Whereas the use of blade tip seal rims provides a labyrinth seal effect against hot-gas leakage flow, the resulting heat transfer coefficients only resemble those of labyrinth seals in a limited portion of the tip. Three-dimensional flows dictate that only the midchord to aft region of a tip cavity may be treated approximately as a labyrinth seal cavity oriented transverse to the flow. Fortunately, this is also the region of highest leakage flows. In lieu of actual blade tip experimental measurements, data such as that of Metzger et al.[13] for simulated transverse tip grooves can be used to estimate heat transfer coefficients in these high leakage regions.

The thermal benefit of the squealer tip geometry, noted to be as much as 50% lower heat transfer than the flat tip, is dependent on the survival of the extended seal rims. Depending on the highly three-dimensional flows inside the squealer tip cavity, the heat transfer coefficients on the rim interior surfaces can be either very high, typical of the suction side rim, or very low, typical of the pressure side rim. The higher heat transfer coefficients are very similar to those experienced by the exterior seal rim surfaces, both the upper surface and the outer surfaces in the hot-gas flowpath. These latter surfaces may be characterized by the flow and heat transfer for sink and source-type flows. The pressure side entry region can be modeled as a sink flow from the mainstream passage into the tip gap. The suction side exit region can be modeled as a source flow from the tip gap into the crossing mainstream passage. Metzger and Rued[14] and Rued and Metzger[15] studied such sink and source flow effects on heat transfer for a range of relative flow strengths and gap magnitudes. Sink flow leads to accelerated flow into the tip gap with increasing heat transfer coefficients as the top of the rim seal is approached. Increases of 200–300% over heat transfer coefficients in the mainstream passage below this region can be experienced. Source flow effects are associated directly with the suction side tip vortex noted earlier and may be quite variable with relative gap size and local leakage-to-mainstream momentum ratio. Figure 27 shows the ranges of resulting heat transfer enhancements obtained by Rued and Metzger[15] (shown as Stanton number ratios). The relative gap of the ordinate refers to the magnitude of the tip clearance, where a relative gap size of 7–8 should be thought of as nominal operating clearance. The abscissa shows the ratio of source flow Reynolds number based on gap size to the mainstream passage Reynolds number, which can be interpreted as the ratio of mass velocities locally (or a blowing ratio). Three general behavior characteristics are shown for the source flow heat transfer enhancements in the right-hand side of Fig. 27, where the dashed lines represent the level of the heat transfer due to the mainstream passage flow alone and the abscissa is the radial distance from the tip. In region I, a stratified flow exists, and the tip rollup vortex is absent. This behavior occurs when the relative gap size and the leakage blowing strength are well matched such that mixing momentum transfer is minimized only to that required for turning the leakage flow. Such behavior would be desirable by comparison with the other characteristics, but is not generally obtained. Region II behavior is most common, showing the typical tip vortex effect increasing surface heat transfer as the vortex scrubs the near tip area of the suction side (Figs. 15 and 19). Here the relative gap size is nominal or greater, and the leakage blowing ratio is weak, leading to the formation of the vortex as the flow is mixed and turned. Region III behavior is

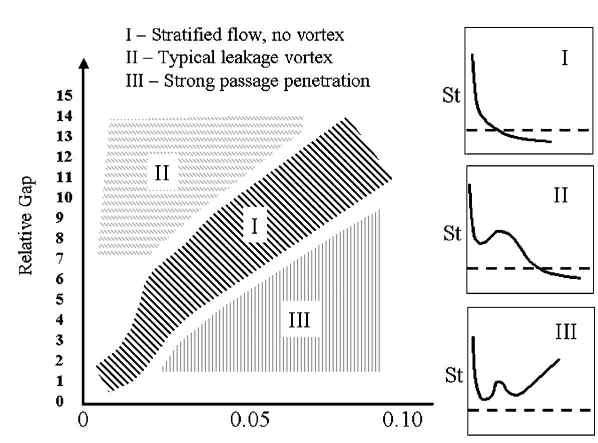

Fig. 27 Heat transfer characteristics for near-tip sink and source flows (reproduced by permission of the American Society of Mechanical Engineers).

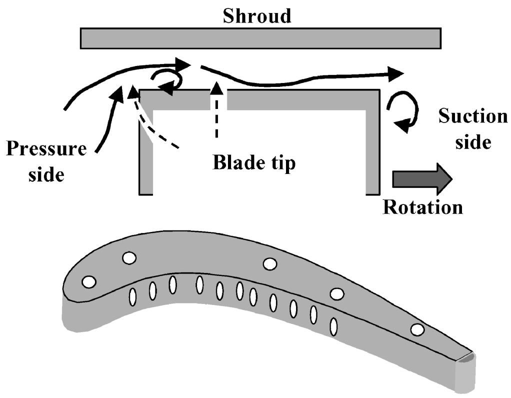

Fig. 28 Schematic of film-cooled blade tip.

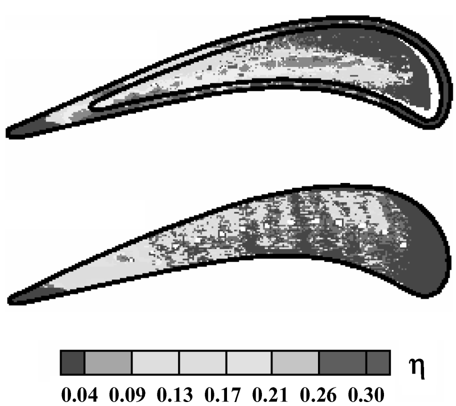

Fig. 29 Film cooling effectiveness distributions for flat and squealer tip blade cascade (reproduced by permission of the American Society of Mechanical Engineers).

associated with blade tips that have lost sealing effectiveness, leading to a high leakage momentum that penetrates the mainstream flow deeply, causes major secondary flow losses, and increases near tip heat transfer.

When these high heat transfer coefficients in the hot-gas leakage stream are considered, film cooling of HPT blade tips is common. Figure 28 shows the general application of film cooling holes to a flat blade tip, with many holes aligned in the radial direction near the tip entry on the pressure side and several more holes on the tip surface. Holes on the tip surface are not always film cooling holes, but may be required dust purge holes for the internal cooling circuits of the blade. The pressure side film holes are angled steeply for best film adherence to the surface, as well as to allow drilling into the internal cooling passages. Similar film hole placement strategies are used on squealer tips as shown in Fig. 8; however, the holes in the tip cavity must be located with a good knowledge of the more three-dimensional flows here. As shown in Fig. 28, tip film cooling flows do not present major alterations to the overall tip leakage flow patterns. Efficient tip film cooling using minimal coolant amounts is intended to reduce heat flux to the surfaces to enhance survival, not to block fluidly leakage flow from entering the tip gap, though this latter effect may be present to a small degree. Most blade tip film cooling designs are achieved through experience, not through pretest analysis because there is little or no comparable database on film cooling of tips to that existing for airfoils and endwalls. One indication of tip film effectiveness is shown in Fig. 29 from the cascade measurements of Kwak and Han.[16] With the same blade tip cascade[12] as that of Fig. 26 used, the local adiabatic film effectiveness is shown with 13 radial pressure side film holes and 13 normal tip surface holes, the group having an average blowing ratio of 2. Effectiveness η is here defined as $(T_{\text{gas inlet}} - T_{\text{adiabatic surface}})/(T_{\text{gas inlet}} - T_{\text{coolant}})$ and is shown for both a flat tip and a squealer tip. With a tip clearance

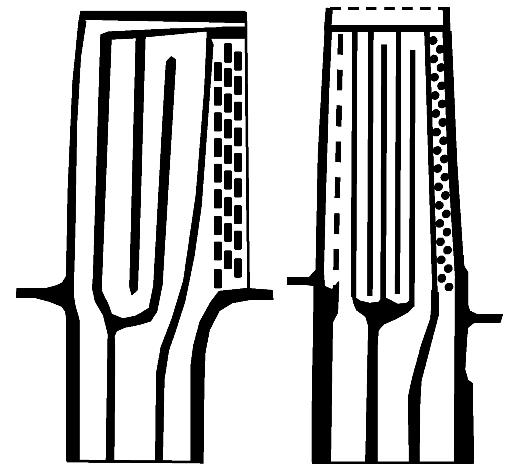

Fig. 30 Exemplary blade internal cooling circuits for flat and squealer tip designs.

of 1.5% blade span, the film cooling effectiveness magnitudes are fairly low. A majority of the pressure side film does not survive the entry into the tip. Film injected on the tip follows the leakage flow directly and exhibits a high degree of mixing. Film inside the tip cavity tends to be disrupted by reattaching flows, but can collect in the more protected regions to provide higher cooling effectiveness as seen in the apparent film cooling directed along the chordline. It is also extremely difficult to cool the suction side seal rim of a blade tip unless dedicated film holes are placed nearby. This simple example illustrates the high degree of complexity in cooling any blade tip region.

In reality, each tip film cooling hole will have a different blowing rate and unique mixing with the leakage flow. This fact is driven not only by the external aerodynamics but also by the internal cooling of the blade tip, which in turn is directly linked to the overall cooling design of the entire blade. Yet this internal cooling plays an important role in maintaining the tip integrity, and so the internal heat transfer coefficients must also be well known. Figure 30 shows two internal blade cooling designs, one with a flat tip and the other with a squealer tip. These are but two designs of an almost infinite variety, but show common cooling circuit features. The flat blade tip uses a dedicated cooling passage under the tip surface that contracts from forward to aft to compensate for the extraction of coolant out tip film and purge holes. Such dedicated channel cooling provides a well-characterized internal heat transfer mechanism. The squealer tip is employed in this example with serpentine cooling channels. The internal tip cooling here is performed by the serpentine turning regions, which typically involve high heat transfer coefficient enhancements due to impingement and secondary flows.

Heat transfer and cooling for the various forms of attached tip shrouds differ substantially from that of flat tips and squealer tips. Some similarities are present, such as the sink and source flow effects at the upstream and downstream extensions of the tip shrouds adjacent to the stationary casing, as well as the converging and expanding flows associated with seal teeth. The heat transfer characteristics of these features are similar to that of the other blade tip designs, but modified by the existence of the casing recess volume. For example, a tip shroud will still generate a suction side vortex as the leakage flow interacts with the passage flow, but any increased heat transfer will affect the tip shroud surfaces more than the airfoil. The immediate hot-gas side of the tip shroud acts as an endwall to the airfoil, and so heat transfer will be affected by a leading-edge horseshoe vortex, passage vortex, and other secondary flows much in the same way as the blade hub region. For the most part, similar heat transfer coefficient characteristics can be applied on hub and tip endwalls with a tip shroud present. Cooling of the region between the tip shroud and casing is more complex. Figure 31 shows two predictions of tip shroud heat flux distributions from the study of Nirmalan et al.,[17] one with rotational effects and one stationary,

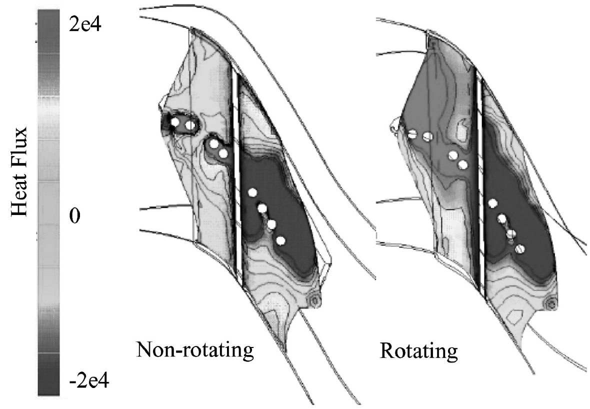

Fig. 31 Predicted tip shroud heat flux distributions (reproduced by permission of the American Society of Mechanical Engineers).

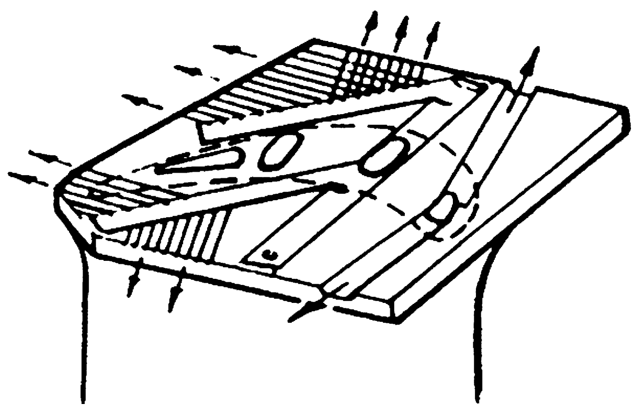

Fig. 32 Distributed internal cooling for a complex tip shroud (reproduced by permission of Rolls–Royce, plc.).

both including coolant injection out the tip holes. The geometry is essentially that of the tip shroud in Fig. 10, but the periodic computational domain is selected based on the underlying airfoil shape. The stationary prediction agreed fairly well with stationary tip shroud cascade data. In the nonrotating case, heat flux is neutral forward of the seal tooth, neither into nor out of the tip shroud surface, except the high heat transfer (coefficients) at the cooling hole exits. When rotation is added, a circumferential flow is set up in the forward tip shroud recess volume that drastically enhances heat flux to the tip shroud. The coolant here does not attach to the tip shroud, but instead mixes into the leakage flow and carries over the seal tooth. Both rotating and non-rotating cases exhibit similar effects aft of the seal tooth. The aft region is flooded by cooling flow to the point that heat flux is actually out of the tip shroud surface. Such disparities in tip shroud heat flux are undesirable, but not necessarily detrimental. More distributed cooling is required to maintain greater uniformity. The cooling scheme for a higher temperature tip shroud is shown in Fig. 32 per Ref. 18. In this schematic, the upper seal strips (Figs. 9 and 21) have been removed to expose the internal cooling passages. Cooling channels spread the air to the extended portions of the tip shroud. Coolant is then discharged out perimeter holes to provide purge sealing against further hot-gas leakage, as well as a degree of film cooling on the mainstream side of the tip shroud. This very complex cooling network is required to maintain the tip shroud integrity against the effects of oxidation and creep. Accompanying this complexity is a higher uncertainty in external heat transfer coefficients and film effectiveness, which are again best determined through operational experience.

In-Service Conditions and Changes

The main characteristics of blade tip aerodynamics, heat transfer, and cooling have been discussed in the foregoing sections. There are still other factors influencing blade tips that have impact on operation

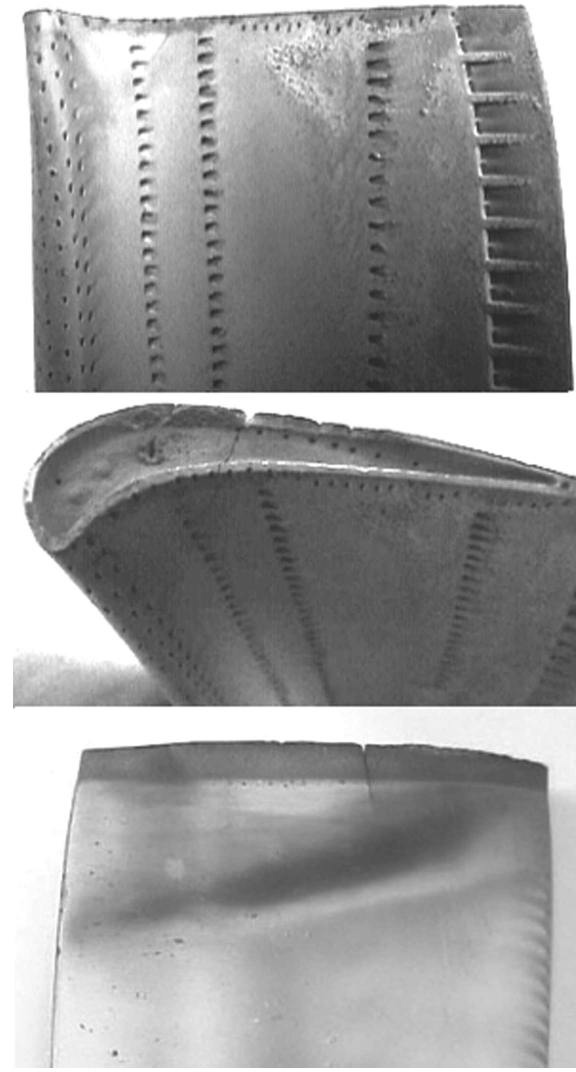

Fig. 33 Sample squealer blade tip after unspecified hot exposure time in service.

and durability beyond those considered in the basic design. Nominal design conditions do not remain constant, either because the blade tips themselves degrade with time and/or because the combustor and surrounding turbine systems change or degrade with time. The degree and rate of modified in-service conditions depends on the mission, the operator, and the local environment of the installation. At least two aspects of blade tip service durability will be present in virtually all designs: Blade tips will lose material due to oxidation, and blade tips will rub against the shrouds.

Tip material loss due primarily to oxidation effects is shown in the squealer tip of Fig. 33. This example is for an HPT blade tip with unspecified exposure time (cycles) and temperature history, but shows the real life distress that such blade tips may experience. The depth of the tip cavity is less than that of the new part, though both the leading-edge region and the aftmost part of the tip cavity are fairly intact. The brunt of the material loss has occurred in the midchord section, especially as noted on the suction side rim. The pressure side rim has been oxidized down very close to the exits of the film holes. In this particular example, the blade tip section has not been coated with TBC, as noted in the airfoil suction side view. One or two deep cracks can also be seen, which are usually associated with local regions of high thermal gradients or stress concentrations due to cooling holes or plugged core support holes. Note, however, that even with such distress, this blade tip has not failed. The squealer tip has performed its function. The progression of conditions from the new to the degraded involves a complex mix of aerodynamics, heat transfer, and film cooling changes leading

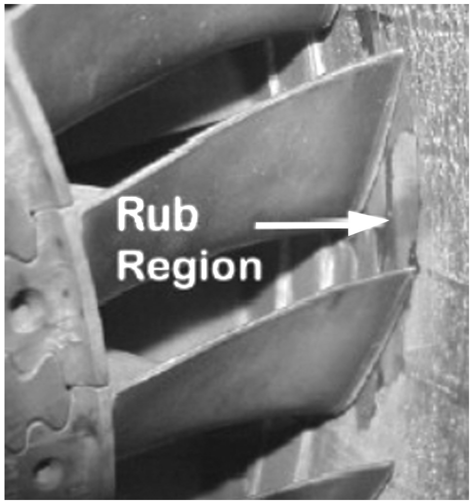

Fig. 34 Example of blade tip rub on stationary shroud.

to altered material temperatures locally, all of which can snowball from some starting nucleation region of initial distress.

Tip changes due to rub events with the stationary shroud may involve the loss of material, or in some cases the redistribution of material. Depending on the materials involved, the temperature levels, and the degree of transient severity of the rub event, blade tip and/or shroud material may be worn away and lost, or may be redeposited on the tip or shroud. Figure 34 shows an example of a blade tip rub against a ceramic–matrix–composite (CMC) shroud from Corman et al.,[19] as evidenced by the smeared local surface region. If the material is simply borne away by the flow, the tip loss is manifested as a uniform increase in clearance. If the material is removed from tip or shroud and deposited on the other component, then a buildup of material locally can occur, which in turn leads to more progressive rubbing and damage. This situation is possible when the casing and shroud ring are not perfectly round, such that each blade tip in the row rubs in one or more repeated locations. In fact, circumferentially uniform rubs are not found in practice, but instead rubs are localized by the eccentricities in the casing. This mode of tip material loss can become quite severe, leading to excessive vibration of the airfoils, and in extreme cases to rotor "freeze."

In both types of tip service changes noted here, the aerothermal boundary conditions become moving targets with time. The local or uniform loss of material will alter the tip flowfield and the heat transfer coefficients. Work extraction and efficiency will change due to clearance increases and redistribution of pressures. Film-cooling flow rates will be modified, and in some worst cases, cooling holes may even be blocked by oxidation, debris, or rub material. As an example of the degree of change possible, Fig. 35 shows the regional averages of blade tip heat transfer coefficients obtained by Bunker and Bailey[20] as modified by the progressive uniform loss of a squealer tip rim. The blade tip configuration of Fig. 25 was used in this study with a constant tip clearance gap. The unaffected squealer rim has an average tip cavity heat transfer coefficient level some 50% lower than that of the flat tip. As the rim material is removed to model a tip rub, average tip cavity heat transfer is increased. When the rim is one-third of its original height, the heat transfer is only 10% below that of the flat tip. With use of other data from the same study, this loss of seal rim material is equivalent to about a 40% increase in tip clearance gap. Thus, tip seal material loss leads not only to greater clearance gaps, but also to less effective seal mechanisms. These two effects combine to accelerate tip degradation in service.

Service conditions for blade tips should also include consideration of repair. Because blade tips are typically degraded in service, repair methods have been devised to remove and rebuild all or portions of the tips, thereby avoiding the scrapping of valuable hardware. Repair operations may take place three to five times on a single blade tip before that part is permanently removed from service. Flat blade

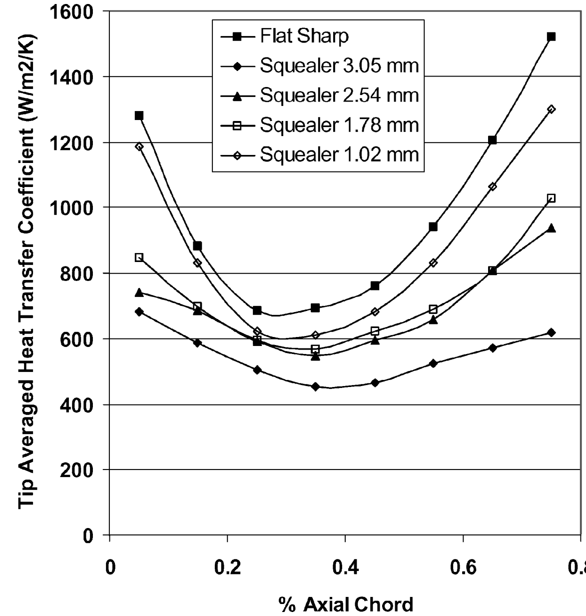

Fig. 35 Effect of squealer tip cavity depth on heat transfer coefficient distribution (reproduced by permission of the American Society of Mechanical Engineers).

tips, although simple in design, are perhaps the most difficult for repair. Any time material must be removed and rebuilt to repair cracks or recover wall thickness, it is virtually impossible to obtain the original new part material properties. Because flat tip repairs occur on wall sections that directly affect the integrity of the cooled airfoil, these are particularly sensitive to changes in properties, for example, less than single crystal strength. Squealer tips have an advantage in this respect in that the rim material may be removed and rebuilt without intrusion into the structural portion of the cooled airfoil. In fact, a squealer rim might be repaired using a differing material from that of the original, a material more suitable to weld operations, for example. Attached tip shrouds have aspects common to both flat and squealer tips. With reference to Fig. 9, the seal teeth might be easily rebuilt, but any repair required to the cooled sections of the tip shroud will be more involved.

Stationary Shroud/Casing Design

A few words are in order concerning the function and design of the stationary shroud or casing opposite the blade tip. In many ways, the shroud and the tip are considered an integral system. This is primarily true for two aspects of their function, first, during transient operation and, second, when they touch. Shrouds are subject to many of the same design constraints and requirements as blade tips, excepting of course the rotational effects. Shrouds in HPTs are usually cooled structures, whereas those in LPTs are uncooled. Shroud designs differ in their circumferential extent, axial recessing, and materials. Shroud systems may be designed to allow an equal number of shroud segments to blades, one shroud segment for every two blades, and sometimes a complete 360-deg shroud ring for the blade row. The advantage of reducing the number of shroud segments comes in decreasing the leakage flows that must be provided at the interfaces. The disadvantage of a single shroud ring is that it does not allow for ease of repair and replacement. Individual shroud segments, sometimes called shoes, are easier to manufacture and can be refaced or rebuilt easily. Continuous shroud rings are more amenable to roundness control and consistency of transient response. Because of the shear size of such rings though, these types of systems are generally only found on some aircraft engines or small turbines. The prevalent shroud systems in practice comprise segmented shrouds with fairly elaborate perimeter sealing.

Cooling of shrouds can usually be accomplished through a combination of impingement and convective methods, though some film cooling is sometimes required. Materials are usually similar to those of the partner blade, including the use of TBC. Some advanced materials are now finding their way into service for shrouds, such as the CMC shown in Fig. 34. These materials have higher temperature capability, allowing less cooling, but also have challenges with respect to stresses and system integration. In most casing systems, there are actually two sets of shrouds for each blade row. An outer shroud or casing, which holds the inner shrouds that form the hot-gas flowpath. This inner–outer system then isolates the hot shroud elements from the colder structure and allows for the introduction of air for both cooling and transient operational control. This latter aspect of shroud systems is referred to as active clearance control (ACC). In concept very simple, ACC uses cool air or heated air at the appropriate times during transients to control the radial thermal displacement of the inner shroud to match the radial movement of the blade tip better thereby maintaining an efficient tip clearance over a greater range of operating conditions. In practice, ACC is more easily implemented on power turbine installations than in aircraft engines.

Shroud designs can be found with a great variety of recessing geometries opposite the blade tip. Flat tips and squealer tips are generally opposed by a shroud of constant cylindrical arc sector or radius (except at the shroud-to-shroud interfaces). Sometimes these shrouds are located at slightly larger radius than the upstream vane endwall to avoid the possibility of undesired flowpath disturbances. Tip shrouds will always be recessed into the stationary shroud as discussed earlier. Such recessed casings may be cylindrical, flared, or even stepped radially outward. Frequently these recessed shroud surfaces will employ additional sealing methods such as honeycomb materials, or labyrinth seal teeth acting in concert with the tip shroud seal teeth. In every case, the stationary shroud surfaces must bear the brunt of any tip rubs. It is a requirement that shrouds be sacrificial in this respect, sparing the life and operability of the rotating blade.

Conclusions

The foregoing discussion has highlighted the basic functional and operational requirements associated with axial turbine blade tips. These requirements, in conjunction with the many other competing system aspects of turbine engines, lead to several design solutions found in practice today. Conceptually, the hot-gas path interface between the rotor and casing is quite simple; however, the extreme environmental conditions plus the cascaded systems issues make this interface sensitive to small changes. In modern turbines, the blade tip regions are responsible for about one-third of the total aerodynamic losses, require a significant amount of cooling, and, thus, pose one of the major remaining challenges to efficiency gains. These same blade tips are also one of the major causes for limiting the service times of turbine engines, impacting blade life, operational costs, and maintenance costs. The proposed and applied detailed design solutions for efficient and durable blade tips are many. All designs have both positive and negative aspects with respect to the blade tips and the turbine system. In the end, each design must balance aerodynamic performance, consequent thermal loading and cooling requirements, blade and rotor stresses, transient operational characteristics, and durability issues to produce a successful outcome.

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