JOURNAL OF PROPULSION AND POWER
Vol. 16, No. 4, July–August 2000

# Gas Turbine Heat Transfer: Past and Future Challenges

Ralf Schiele* and Sigmar Wittig†
*University of Karlsruhe, 76128 Karlsruhe, Germany*

**The demand for gas turbines with increasing power and efficiency calls for extremely high temperatures in the hot-gas sections of the engines. These temperatures can only be realized by employing sophisticated cooling schemes. Inadequate cooling may result in excessive material temperatures with reduced reliability and a lifetime of those parts subjected to the hot gas. Based on a survey of the different cooling techniques employed in modern gas turbine engines and their application in gas turbine combustors, as well as turbine components, modern aspects and future developments are discussed. Results from laboratory experiments that help to understand the physical phenomena are presented, as well as theoretical analyses. The possible use of ceramic materials is demonstrated by means of tests carried out at the Institut für Thermische Strömungsmaschinen, University of Karlsruhe. Besides describing current techniques, new developments are assessed and goals for future research are discussed.**

## Introduction

FOR a long time, generating electric power with gas turbines has been mainly a matter of peak-load coverage. Since the early 1970s this situation has changed dramatically. The enormous increase in maximum power per engine unit from approximately 60 to 240 MW and the combination of gas and steam turbines has made gas turbine power plants very attractive both economically and ecologically. The independent power producers all over the world who invest in power production by gas turbine engines are primarily interested in running the engines all year and selling electric power to their customers.

In just the last 10 years a 10% increase in efficiency, an order of magnitude reduction in NOx emissions, a 20% increase in power density, a factor of two increase of the pressure ratio, and an approximately 200-K increase in turbine inlet temperature have been achieved. These improvements were partly made possible by revisions in the aerodynamics of compressors and turbines. As the most critical technology for reliability and durability of the engines, however, turbine cooling technology has at all times accounted for a significant portion of the resulting powerplant improvement.

Future developments aim at a further increase in process temperatures and pressures with turbine inlet temperatures beyond 1800 K. In addition, the thermodynamic design of combined-cycle power plants will be improved leading to efficiencies in the electric power generation surpassing 60% within the next few years.

In the past aeroengines have for the most part set the pace for new developments in cooling technologies. The demand for high thrust with the lightest engines possible has always called for extremely high turbine inlet temperatures. In today's engines, turbine inlet temperatures greater than 2000 K, at pressure ratios of more than 40, exceed maximum allowable material temperatures by more than 700 K. At the same time, commercial carriers and military customers are asking for longer maintenance intervals and increased engine reliability. Cooling of the hot-gas parts will, therefore, remain one of the key technologies for the design of new engines and for any future economical success by engine manufacturers.

In this context some aspects of modern gas turbine cooling technologies are highlighted in the present paper. In keeping with the authors' own experiences, the review concentrates on selected areas rather than aiming at a complete review of the topic, which would otherwise have been very superficial within the alloted space.

## Cooling Concepts and Their Application in the Hot-Gas Path of Modern Gas Turbine Engines

Various cooling concepts are currently being used to prevent liners and blades in hot sections of gas turbines from failure. In general, these techniques consist of convective and film cooling technologies or a combination of both.

Convective cooling of the rear surface of the engine parts is the simplest method to reduce wall temperatures. For effectiveness it requires intensive heat transfer on the cold wall. This may be achieved by guiding the coolant through thin slots on the rear side of flametubes or through cooling passages in the blades (Fig. 1a). In this way relatively large air velocities are obtained leading to high heat transfer coefficients. Additionally, the cold surfaces may be equipped with arrays of pins or ribs to increase turbulence and the area for heat exchange.[1–5] Impingement cooling is another method of obtaining high heat transfer rates on the cold side of the engine parts.[6–8] When this concept is applied, the coolant is in the form of high-momentum jets that hit the rear surface perpendicularly (Fig. 1b). After the deflection of the jets, very thin boundary layers are formed inducing high heat transfer coefficients. All concepts described can be of varying effectiveness, depending on design and flow conditions in the coolant passage.

A very effective way to lower wall temperatures is the generation of a coolant film on the hot surface, which, thereby, considerably reduces the heat transfer between the hot gas and combustor liners or turbine blades. After formation, the cooling film flows along the hot surfaces and is continuously diluted by mixing with the main flow. Accordingly, new films must be generated after a certain surface distance to guarantee effective protection. To produce cooling films, the coolant is blown through parallel or inclined slots or through differing numbers of inclined holes (Fig. 2).

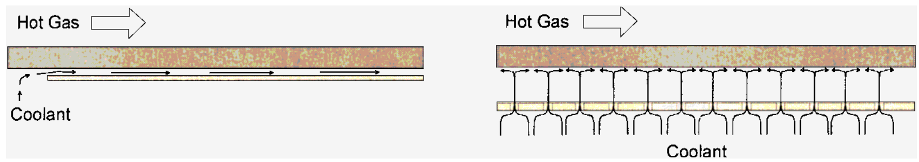
Geometrical parameters such as slot height or distance between cooling holes may vary in a wide range leading to very different film flow patterns. For a given geometry the blowing ratio $M = \rho_c u_c / \rho_m u_m$ and the momentum flux ratio $I = \rho_c u_c^2 / \rho_m u_m^2$, denoting ratios of coolant (subscript $c$) and main (subscript $m$) flow properties, have proved to be good means for correlating performance data on film cooling. Both quantities are, therefore, commonly used in the film cooling literature.

The temperature reduction by film cooling can be considerably higher than by convective cooling. Furthermore, the heat flux transferred to the walls is greatly reduced so that the net heat load on the walls is smaller with film cooling. On the other hand, the performance of film cooling is highly dependent on the main flow. In certain situations this can lead to very low effectivities of film cooling, for example, in the interactive regions of cooling films with mixing

Received 24 January 1999; revision received 24 April 2000; accepted for publication 24 April 2000. 

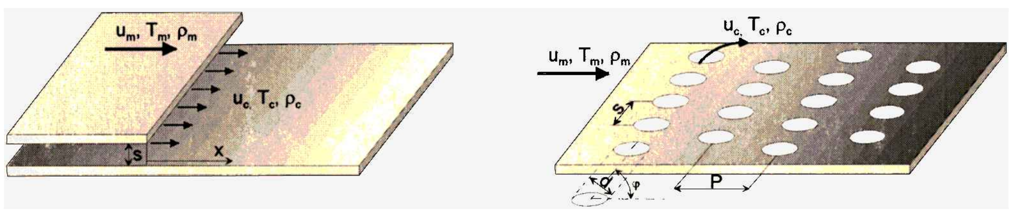
*Senior Research Engineer, Lehrstuhl und Institut für Thermische Strömungsmaschinen, Kaiserstraße 12; Ralf.Schiele@its.mach.uni-karlsruhe.de.

†Professor, Lehrstuhl und Institut für Thermische Strömungsmaschinen, Kaiserstraße 12.



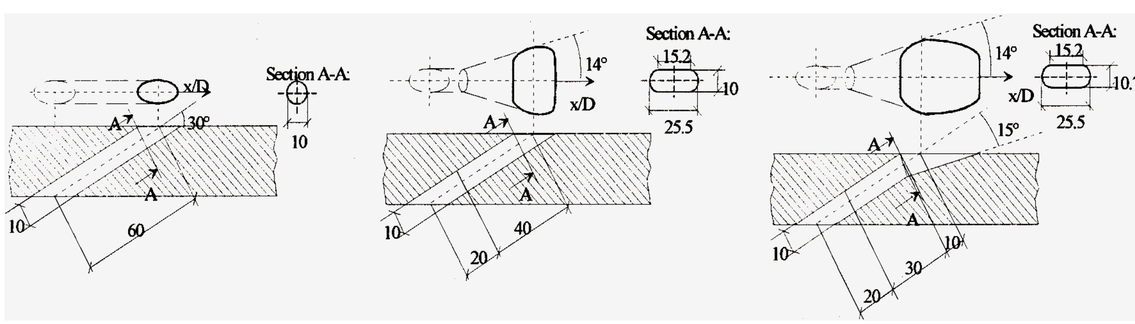

a) b)

**Fig. 1 Convective cooling using a) slot geometry and b) impingement.**




a) b)

**Fig. 2 Film cooling by a) slot injection and b) full coverage film cooling.**




a) b) c)

**Fig. 3 Geometries of film-cooling holes: a) cylindrical, b) fanshaped, and c) laidback fanshaped.[17]**

jets in combustors.[9,10] In contrast to film cooling, convective cooling schemes can be controlled more precisely over the entire range of engine operation. Therefore, combining convective cooling with film cooling is both the most effective as well as the safest way to prevent engine components from failure through excessive heat loads.

Two areas that have attracted attention in recent years and that have come into use in the newest generation of gas turbine engines will be examined in greater detail. One is the utilization of different hole shapes to increase the effectiveness of film cooling[11–23] and the other is the application of effusion cooling[24–31] in combustors.[32,33]

### Film Cooling Holes with Expanded Exits

Figure 3 shows three different film cooling hole geometries.[17] When contouring is applied, the cross-sectional area at the hole exit is increased compared to a standard cylindrical hole, leading to a reduction of the mean velocity and, thus, of the momentum flux of the exiting jet. Therefore, the penetration of the jet into the main flow is reduced, which results in an increased cooling efficiency. Furthermore, lateral expansion of the hole provides an improved lateral spreading of the jet, which leads to better coverage of the airfoil in the spanwise direction and a higher spanwise-averaged film cooling effectiveness.

Recent studies have shown that expanding the exit of the cooling hole improves film-cooling performance relative to a cylindrical hole. Overall improvements in adiabatic effectiveness were found for laterally expanded holes,[11] as well as for forward expanded holes.[12] Sen et al.[13] and Schmidt et al.[14] compared a cylindrical hole to a forward expanded hole, both having a compound angle injection. Although the spatially averaged effectiveness for the cylindrical and forward-expanded holes were the same, a larger lateral spreading of the forward expanded jet was found. Haller et al.[15] performed aerodynamic loss measurements on a transonic cascade. Holes with a spanwise flare angle of 25 deg were found to offer significant improvements in film-cooling effectiveness without any additional loss penalty.

Flowfield measurements performed by Thole et al.[18] showed that jet penetration as well as velocity gradients in the mixing region were significantly reduced for holes with expanded exits as compared to a cylindrical hole, at the same blowing rate. Peak turbulence levels were found downstream of the plane for the expanded holes. Numerical studies performed by Giebert et al.[16] were able to predict the general flow features of coolant ejection through diffuser shaped holes. Discharge coefficient measurements presented by Gritsch[20] for the same hole geometries as shown in Fig. 3 showed that the discharge coefficient for all geometries strongly depends on the flow conditions (crossflows at hole inlet and exit and pressure ratio). The discharge coefficient of both expanded holes was found to be higher than of the cylindrical hole, particularly at low-pressure ratios and with a hole entrance crossflow applied. The effect of the additional layback on the discharge coefficient was negligible.

Figure 4 gives an impression of the possible improvements that might be achieved by contouring of the film cooling holes. The diagrams show the two-dimensional distributions of the adiabatic film cooling effectiveness $\eta_{ad}$ downstream of the cylindrical, the fanshaped, and the laidback fanshaped holes shown in Fig. 3. For this illustration, a typical blowing ratio $M$ of unity is depicted. The calculation of this blowing ratio is based on the inlet cross-sectional area of the holes, which is identical for the three different geometries. Thus, the constant blowing ratio is synonymous with a constant amount of coolant.

Because no convective effects are presumed to be present in adiabatic walls, any change in surface temperature is solely determined by the cooling film. The adiabatic film-cooling effectiveness

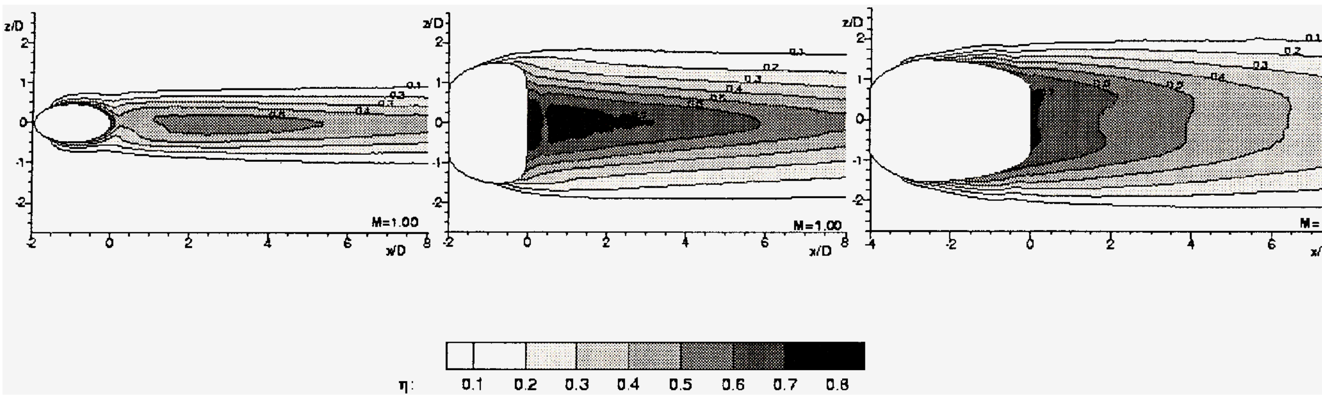



**Fig. 4 Local film-cooling effectiveness distribution for the cylindrical, fanshaped, and laidback fanshaped hole geometry.**

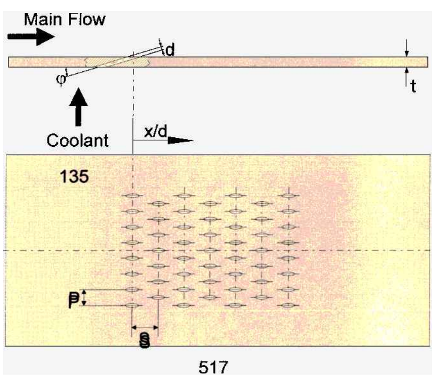



**Fig. 5 Test plate with effusion cooling.[32]**

$\eta_{ad} = T_m - T_{aw} / T_m - T_c$, which is a dimensionless form of the adiabatic wall temperature $T_{aw}$, therefore, is a good measure for the performance of the cooling film. Its value ranges from zero ($T_{aw} = T_m$) to one ($T_{aw} = T_c$).

The superior performance of the shaped film-cooling holes is obvious from these results. For the blowing ratio shown, both the maximum adiabatic film-cooling effectiveness and the surface area covered are greatly increased when contouring is applied. The lateral averaged value of $\eta_{ad}$ is, therefore, much higher for the shaped holes than for the cylindrical hole. Going from the fanshaped hole to the laidback fanshaped contour, the maximum adiabatic effectiveness decreases, but lateral spreading of the jet improves. In general, this leads to a higher value of the surface-averaged film-cooling effectiveness for the laidback fanshaped hole than for the regular fanshaped hole.

Note that cooling effectiveness and discharge coefficients of film-cooling holes are strongly affected by the way the coolant is supplied to the film-cooling hole.[17–21] Therefore, a plenum chamber, such as that widely used for experimental studies, is not a correct means of representing the internal coolant passage flow of an airfoil. Any future investigation on the performance of film-cooling holes should take this consideration into account.

### Effusion Cooling

Reducing pollutant emissions in combustors asks for new combustor concepts such as lean-premixed-prevaporized and rich-quench-lean combustion. Both methods require a larger amount of combustion air than conventional concepts. As a consequence, less air is available for cooling purposes while the heat load to the liner increases. The development of highly efficient cooling techniques is, therefore, crucial for modern combustor design. Effusion cooling, in which the coolant is lead through a large number of small diameter holes and then discharged onto the inner combustor surface, is a promising method to meet the high cooling requirements (Fig. 5).

Effusion cooling acts on three areas: first, on the back surface, where the heat is withdrawn by convection; second, on the inner hole surface, along which heat is convectively exchanged between the coolant and the wall material; and third, on the inner surface of the combustor liner where the coolant forms a protective film that drastically reduces the thermal load to the wall.

The heat transfer on the back surface may be of high importance for realistic conditions as seen in Fig. 6, where the lateral-averaged overall cooling effectiveness $\eta = T_m - T_w / T_m - T_c$ is shown for a metallic test plate for three different blowing ratios with and without impingement cooling on the backside.

Even though cooling effectiveness is already at a high level for effusion cooling without impingement, $\eta$ is considerably increased through the enhanced heat transfer on the cold side of the liner when impingement is applied.

It is obvious that the increase in effectiveness is highest in the area where the cooling film is the least effective, that is, for $x/d < 25$. There, the large temperature difference between wall and coolant enables an intense convective cooling on the backside. At positions downstream of $x/d = 35$, the increase in effectiveness is less pronounced because the cooling film already offers good protection. The impingement area ends at $x/d = 45$. Farther downstream, the effectiveness approaches the same level of effusion cooling without impingement.

Again, these results demonstrate that film cooling and convective cooling support each other in a kind of teamplay, and thus, a combination of both is very promising to meet the current requirements of cooling for combustor liners or turbine components. Regions where only poor film cooling can be achieved may still be well protected if convective cooling is established on the backside of the liner.

## Ceramic Gas Turbine Components

Because of the favorable properties of ceramics at high temperatures, considerable efforts have been made to replace thermally highly loaded metallic gas turbine components with ceramic parts. If ceramics are successfully employed in gas turbine engines, higher operating temperatures may be realized and/or the amount of cooling air required may be diminished.

Thus far, brittleness and the inability to reduce stresses by plastic deformation have for the most part prevented ceramic components from being applied in gas turbines. From a design point of view, it is not possible to replace metallic structures by simply using ceramic structures. Instead tailored design methods have to be used. Only by taking these considerations into account can ceramic materials be employed successfully in modern gas turbine engines.

At the Institut für Thermische Strömungsmaschinen, University of Karlsruhe, a new design concept has been introduced to reduce the mechanically and, especially, the thermally induced stresses in ceramic components.[34] With the aim of an homogeneous temperature distribution in the ceramic structure, the concept relies on a three-layered construction (outer ceramic shell, heat insulating layer, metallic core) and an optimization of the thicknesses of the single layers. During the optimization process, calculations are performed using the finite element method in combination with a failure probability analysis.[35]

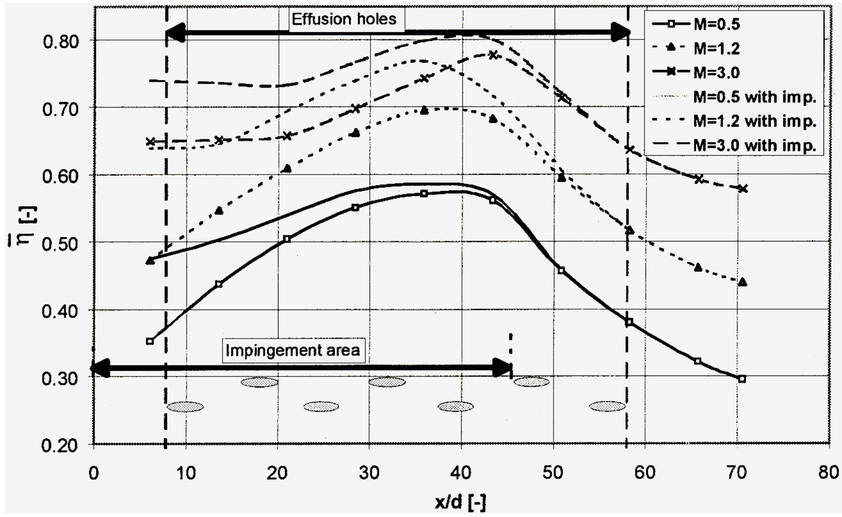



**Fig. 6 Comparison of lateral averaged overall effectiveness of effusion cooling with and without impingement on the backside.**

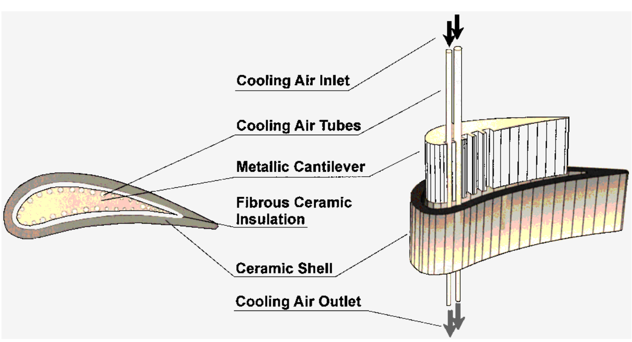



**Fig. 7 Schematic view of an internally cooled ceramic nozzle guide vane.[40]**

In recent years there have been many attempts to introduce ceramic nozzle-guide-vanes in stationary gas turbines.[36–39] Nevertheless, there has been no systematic adjustment of the shape of the ceramic component with respect to thermally and mechanically induced loads. By the application of the new design method to a first-stage ceramic nozzle-guide-vane (SSiC) of a 70-MW, 1400°C class stationary gas turbine, the stresses may be reduced by more than a factor of two, as compared with conventional design concepts (Fig. 7). At the same time, the reliability has been raised substantially, so that even under severe transient thermal boundary conditions (fuel cutoff) a satisfactory probability of failure is attained. An experimental investigation of the ceramic-guide-vane under trip conditions demonstrates the validity of this design technology.[40]

By the combining of the hybrid concept with an ingenious segmentation, even more complex parts, such as a vaneless scroll of a small radial gas turbine, can be designed with significantly reduced stresses and, hence, improved reliability.[41] In combustors, segmentation has proved to be an efficient way to reduce the critical stresses.[42,43] Extensive investigations have shown that from a design standpoint ceramic flametubes can be reliably employed in combustors. Nevertheless, for long-term application in combustors, ceramic materials, which combine high tensile strength and a high resistance to oxidation, have yet to be developed.

## Numerical Simulation Methods

When designing components for modern gas turbines, development time and cost can be reduced significantly by applying numerical methods. To be successful, these tools have to be reliable and inexpensive and, therefore, of high accuracy and efficiency. Besides convective heat transfer phenomena, radiative heat transfer as well as heat conduction through the solid material have to be taken into account.

In gas turbine combustors, the contribution of radiation to the thermal heat load of the liner can be significant. Additionally, radiation also affects the temperature field of the reacting flow and, thus, many other processes such as droplet evaporation, chemical reaction and pollutant formation. Among various methods for predicting radiative heat transfer in participating media, the discrete ordinates method is presently judged as the best compromise between accuracy and computational effort.[44–49] By subdividing the directional space into discrete angles, this method is capable of resolving the non-isotropic character of radiative transfer.

There are many questions related to the correct simulation of the flow and heat transfer in gas turbine engines. It is impossible to address this topic with completeness within the scope of the present paper. Therefore, in the following subsections, two areas are selected that we consider to be of crucial significance.

### Transition

Although the main flow in gas turbines is highly turbulent and unsteady, the flow next to the surfaces may either be laminar or turbulent. By examination of the value of the Reynolds numbers, one can conclude that boundary-layer flows in any gas turbine engine are transitional and, to calculate the losses and heat transfer on various components in the engine, one must be able to predict boundary-layer development through transition.[50]

In the past low Reynolds number $k$–$\varepsilon$ turbulence models were widely considered to be the most promising approach for calculating transition under enginelike conditions. In the literature, various models of this kind have been applied, and it could be demonstrated that transition can be simulated by diffusing freestream turbulence into the laminar boundary layer without further empirical input. Comprehensive reviews on the relative performance of various models in predicting transition are given by Savill[51] and Sieger.[52] Unfortunately, due to difficulties in predicting both onset and length of transition correctly, the experimental heat transfer data are not satisfactorily reproduced in many cases. On turbine blades, the most reasonable results can be obtained with the so-called Production Term Modification (PTM) models,[53] which were developed by incorporating some empirical input.[54]

Recently, two concepts have been presented that have the potential to improve both reliability and accuracy of the transition prediction. By concentrating on the calculation of the turbulentlike fluctuations within pretransitional boundary layers, Mayle and Schulz[55] aim at a physically sound prediction of the onset of transition. These authors developed a new transport equation for the so-called laminar kinetic energy that recognizes the production of laminar fluctuations by the work of the imposed fluctuating pressure forces. Their first results prove the potential of this approach. Schiele et al.[56] have presented a very different concept called the transitional intermittency

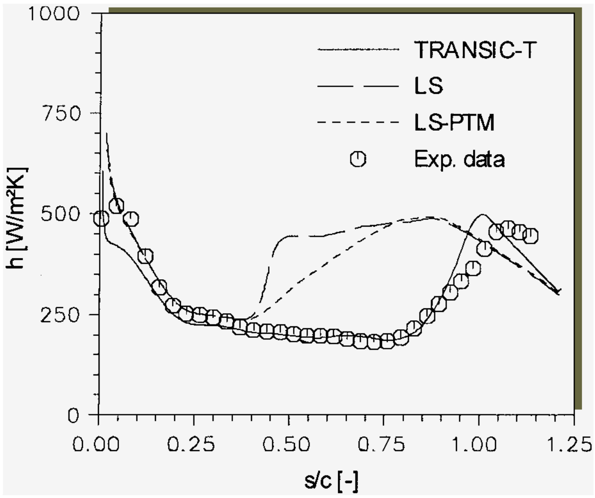



**Fig. 8 Heat transfer along the suction side of a highly loaded turbine nozzle-guide-vane[60]; turbulence models: TRANSIC-T,[56] Launder and Sharma, (LS)[61] and LS-PTM.[53]**

controlled two-layer (TRANSIC-T) model that relies on the incorporation of empirical information in the calculation procedure. Coupling the standard $k$–$\varepsilon$ model[57] with a one-equation turbulence model[58] near the wall, a two-layer turbulence model is used. By the introducing of a new intermittency function into the one-equation model, the simulation of transitional flows becomes possible. Progress in transition is deduced from correlations based on boundary-layer characteristics that are in themselves part of the simulation results. Schiele et al.[56] calculated not only flat-plate test cases[59] but also boundary layers on a first-stage nozzle-guide-vane proving the excellent performance of this method, which has now reached a level of maturity that makes its application in industrial codes possible (Fig. 8[53,56,60,61]).

### Conjugate Methods

When considering the complex cooling schemes in modern gas turbine engines, it becomes obvious that calculating the local temperatures of materials is an increasingly demanding problem. This is especially true in situations in which strong convective effects occur in complex three-dimensional structures for example, effusion-cooled combustor liners, film-cooled turbine blades, or labyrinth seals. To come up with a correct solution, both flow and heat transfer in the fluid as well as heat conduction within the solid have to be calculated. The coupling of these two modes of heat transfer has been identified in the relevant literature by the name conjugate heat transfer. It is obvious that for practical use methods should be developed that enable a solution for both heat transfer modes within the frame of one program. In the long run, this will provide an opportunity to calculate stresses and strains in solid parts and at the same time to consider the changes on the flowfield resulting from the deformation of the solid structures.

Several authors have proposed calculational procedures for conjugate heat transfer. Bohn et al.[62,63] and Heselhaus and Vogel,[64] considered film or internally cooled turbine blades, respectively. Kao and Liou[65] simulated the coupled heat transfer on a convectively cooled turbine cascade and a turbine drum-disk system. Conjugate heat transfer in turbine disk cavities has been investigated by Ho et al.[66] Some authors use finite volume schemes for the fluid and the solid (for example, see Ref. 62) whereas others employ finite volume based codes in the fluid and finite element based programs for the solid (for example, see Ref. 64).

Papanicolaou et al.[67] recently developed a procedure for conjugate heat transfer calculations that is a step forward compared to the previous studies by using body-fitted coordinates, a more sophisticated approach to deal with the coupling between solid and fluid domain heat transfer and the use of a turbulence model that is suited for the recirculating flows that are frequently involved.[68,69]

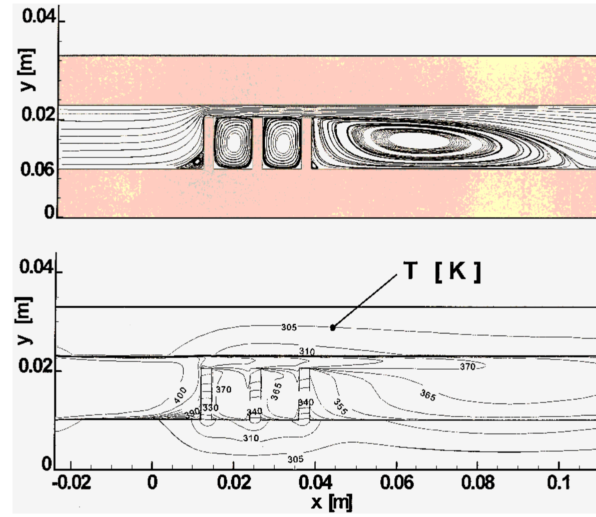



**Fig. 9 Computed flow and temperature fields for a straight through labyrinth seal.[67]**

Papanicolaou et al.[67] successfully applied this method for the calculation of the flow and heat transfer in various labyrinth seals. Small clearances that are subject to large variations due to thermal expansion make temperature control essential for any estimation of the performance of these components.[70,71] Figure 9 shows computed flow and temperature fields for straight-through labyrinth seals as an example. Ongoing work is being done concentrating on the correct simulation of effusion cooled combustor liners, thereby demonstrating the strong effect of heat conduction in the solid structure on the overall cooling performance.

## Conclusion

Cooling in gas turbines is one of the key technologies for any future improvement in engine performance. By developing highly sophisticated concepts, engineers have pushed today's cooling efficiencies to levels that were considered to be out of reach only a decade ago. During this process the interdisciplinary nature of the problem has triggered new developments at the leading edge in various areas, for example, material science and computational fluid dynamics. Based on strong economical and ecological interests, the pace of the development has not settled down yet, making the field of gas turbine heat transfer one of the great engineering challenges in the years to come.

## Acknowledgments

The work at the Institut für Thermische Strömungsmaschinen, University of Karlsruhe, in the area of gas turbine heat transfer was supported by the Deutsche Forschungsgemeinschaft in the context of the Sonderforschungsbereich 167 Hochbelastete Brennräume (Center of Excellence 167) and the Graduiertenkolleg Energie- und Umwelttechnik, by the Bundesministerium für Bildung, Wissenschaft, Forschung und Technologie (bmb+f) within the frame of the AG-Turbo, by the Arbeitsgemeinschaft industrieller Forschungsvereinigungen Otto von Guericke e.V. and by the European community in the context of the Brite Euram program.

## References

[1] Arora, S. C., and Messh, W. A., "Pressure Drop and Heat Transfer Characteristics of Circular and Oblong Low Aspect Ratio Pin Fins," CP-390, AGARD, 1985, pp. 4.1–4.14.

[2] Armstrong, J., and Winstanley, D., "A Review of Staggered Array Pin Fin Heat Transfer for Turbine Cooling Applications," *Journal of Turbomachinery*, Vol. 110, No. 1, 1988, pp. 94–103.

[3] Chyu, M. K., "Heat Transfer and Pressure Drop for Short Pin Fin Arrays with Pin-Endwall Fillet," *Journal of Heat Transfer*, Vol. 112, No. 4, 1990, pp. 926–932.

[4] Hwang, J.-J., and Liou T.-M., "Heat Transfer Augmentation in a Rectangular Channel with Slit Rib Turbulators on Two Opposite Walls," *Journal of Turbomachinery*, Vol. 119, No. 3, 1997, pp. 617–623.

[5]Chandra, P. R., Fontenot, M. L., and Han, J.-C., "Effect of Rib Profiles on Turbulent Channel Flow Heat Transfer," *Journal of Thermophysics and Heat Transfer*, Vol. 12, 1998, pp. 116, 117.

[6]Kercher, D. M., and Tabakoff, W., "Heat Transfer by Square Array of Round Air Jets Impinging Perpendicular to a Flat Surface Including the Effect of Spent Air," *Journal of Engineering for Power*, Vol. 92, No. 1, 1970, pp. 73–82.

[7]Metzger, D. E., Florschuetz, Takeuchi, D. I., Behee, R. D., and Berry, R. A., "Heat Transfer Characteristics for Inline and Staggered Arrays of Circular Jets with Crossflow of Spent Air," *Journal of Engineering for Power*, Vol. 101, No. 3, 1979, pp. 526–531.

[8]Landreth, C. C., and Adrian, R. J., "Impingement of a Low Reynolds Number Turbulent Circular Jet onto a Flat Plate at Normal Incidence," *Experiments in Fluids*, Vol. 9, No. 1, 1990, pp. 74–84.

[9]Martiny, M., Schulz, A., Wittig, S., and Dilzer, M., "Influence of a Mixing-Jet on Film Cooling," *42nd Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 97-GT-247, Orlando, FL, June 1997.

[10]Button, B. L., "Effectiveness Measurements for a Cooling Film Disrupted by a Single Jet with Wall Plunging," CP-390, AGARD, 1985, pp. 24.1–24.10.

[11]Goldstein, R. J., Eckert, E. R. G., and Burggraf, F., "Effects of Hole Geometry and Density on Three-Dimensional Film Cooling," *International Journal of Heat and Mass Transfer*, Vol. 17, 1974, pp. 595–607.

[12]Makki, Y., and Jakubowski, G., "An Experimental Study of Film Cooling from Diffused Trapezoidal Shaped Holes," AIAA Paper 86-1326, 1986.

[13]Sen, B., Schmidt, D., and Bogard, D., "Film Cooling with Compound Angle Holes: Heat Transfer," *Journal of Turbomachinery*, Vol. 118, No. 4, 1996, pp. 800–806.

[14]Schmidt, D., Sen, B., and Bogard, D., "Film Cooling with Compound Angle Holes: Adiabatic Effectiveness," *Journal of Turbomachinery*, Vol. 118, No. 4, 1996, pp. 807–813.

[15]Haller, B. R., and Camus, J. J., "Aerodynamic Loss Penalty Produced by Film Cooling Transonic Turbine Blades," *Journal of Engineering for Gas Turbines and Power*, Vol. 106, No. 1, 1984, pp. 198–205.

[16]Giebert, D., Gritsch, M., Schulz, A., and Wittig, S., "Film-Cooling from Holes with Expanded Exits: A Comparison of Computational Results with Experiments," *42nd Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 97-GT-163, Orlando, FL, June 1997.

[17]Wittig, S., Schulz, A., Gritsch, M., and Thole, K. A., "Transonic Film-Cooling Investigations: Effects of Hole Shapes and Orientations," *41st Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 96-GT-222, Birmingham, UK, June 1996.

[18]Thole, K. A., Gritsch, M., Schulz, A., and Wittig, S., "Effect of a Crossflow at the Entrance to a Film-Cooling Hole," *Journal of Fluids Engineering*, Vol. 119, No. 3, 1997, pp. 533–540.

[19]Thole, K. A., Gritsch, M., Schulz, A., and Wittig, S., "Flowfield Measurements for Film-Cooling Holes with Expanded Exits," *Journal of Turbomachinery*, Vol. 120, No. 2, 1998, pp. 327–336.

[20]Gritsch, M., Schulz, A., and Wittig, S., "Discharge Coefficient Measurements of Film-Cooling Holes with Expanded Exits," *Journal of Turbomachinery*, Vol. 120, No. 3, 1998, pp. 549–556.

[21]Gritsch, M., Schulz, A., and Wittig, S., "Adiabatic Wall Effectiveness Measurements of Film-Cooling Holes with Expanded Exits," *Journal of Turbomachinery*, Vol. 120, No. 3, 1998, pp. 557–563.

[22]Hay, N., Lampard, D., and Benmansour, S., "Effect of Crossflows on the Discharge Coefficient of Film Cooling Holes," *Journal of Engineering for Power*, Vol. 105, No. 1, 1983, pp. 243–248.

[23]Hay, N., and Lampard, D., "The Discharge Coefficient of Flared Film Cooling Holes," *40th Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 95-GT-15, Houston, TX, June 1995.

[24]LeBrocq, P. V., Launder, B. E., and Priddin, C. H., "Experiments on Transpiration Cooling: Discrete Hole Injection as a Means of Transpiration Cooling; An Eperimental Study," *Proceedings of the Institution of Mechanical Engineers*, Vol. 187, 1973, pp. 149–157.

[25]Metzger, D. E., Takeuchi, D. I., and Kuenstler, P. A., "Effectiveness and Heat Transfer with Full-Coverage Film Cooling," *Journal of Engineering for Power*, Vol. 95, No. 1, 1973, pp. 180–184.

[26]Mayle, R. E., and Camarata, F. J., "Multihole Cooling Film Effectiveness and Heat Transfer," *Journal of Heat Transfer*, Vol. 97, No. 2, 1975, pp. 534–538.

[27]Crawford, M. E., Kays, W. M., and Moffat, R. J., "Full-Coverage Film Cooling Part I: Comparison of Heat Transfer Data for Three Injection Angles," *Journal of Engineering for Power*, Vol. 102, No. 4, 1980, pp. 1000–1012.

[28]Kasagi, N., Hirata, M., and Kumada, M., "Studies of Full-Coverage Film Cooling Part 1: Cooling Effectiveness of Thermally Conductive Wall," *26th Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 81-GT-37, Houston, TX, March 1981.

[29]Kumada, M., Hirata, M., and Kasagi, N., "Studies of Full-Coverage Film Cooling Part 2: Measurements of Local Heat Transfer Coefficient," *26th Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 81-GT-38, Houston, TX, March 1981.

[30]Hempel, H., and Friedrich, R., "Untersuchungen zur Effusionskühlung von Gasturbinenschaufeln," *Forschung in der Kraftwerkstechnik*, 1977, pp. 67–73.

[31]Hempel, H., Friedrich, R., and Wittig, S., "Full Coverage Film-Cooled Blading in High Temperature Gas Turbines: Cooling Effectiveness, Profile Loss and Thermal Efficiency," *Journal of Engineering for Power*, Vol. 102, No. 4, 1980, pp. 957–963.

[32]Martiny, M., Schulz, A., and Wittig, S., "Full Coverage Film Cooling Investigations: Adiabatic Wall Temperatures and Flow Visualization," *ASME Winter Annual Meeting, Heat Transfer in Gas Turbines*, American Society of Mechanical Engineers, Paper 95-WA/HT-4, San Francisco, CA, Nov. 1995.

[33]Martiny, M., Schulz, A., and Wittig, S., "Mathematical Model Describing the Coupled Heat Transfer in Effusion Cooled Combustor Walls," *42nd Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 97-GT-329, Orlando, FL, June 1997.

[34]Gutmann, C., Schulz, A., and Wittig, S., "A New Approach for a Low-Cooled Ceramic Nozzle Vane," *41st Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 96-GT-232, Birmingham, UK, June 1996.

[35]Stürmer, G., Schulz, A., and Wittig, S., "Life Time Prediction for Ceramic Gas Turbine Components," *26th Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 91-GT-96, Orlando, FL, June 1991.

[36]Nakakado, K., Machida, T., Miyata, H., Hisamatsu, T., Mori, N., and Yuri, I., "Strength Design and Reliability Evaluation of a Hybrid Ceramic Stator Vane for Industrial Gas Turbines," *Journal of Engineering for Gas Turbines and Power*, Vol. 117, No. 1, 1995, pp. 245–250.

[37]Tsuchiya, T., Furuse, Y., Yoshino, S., Chikami, R., Tsukagoshi, K., and Moro, M., "Development of Air-Cooled Ceramic Nozzles for a Power Generating Gas Turbine," *40th Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 95-GT-105, Houston, TX, June 1995.

[38]Machida, T., Nakayama, M., Wada, K., Hisamatsu, T., Yuri, I., and Watanabe, K., "Development of Ceramic Stator Vane for 1500°C Class Gas Turbine," *41st Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 96-GT-459, Birmingham, UK, June 1996.

[39]Wenglarz, R., Calcuttawala, S., and Pope, E., "Ceramic Vane Demonstration in an Industrial Gas Turbine," *42nd Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 97-GT-155, Orlando, FL, June 1997.

[40]Dilzer, M., Gutmann, C., Schulz, A., and Wittig, S., "Testing of a Low Cooled Ceramic Nozzle Vane Under Transient Conditions," *43rd Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 98-GT-116, Stockholm, June 1998.

[41]Filsinger, D., Schulz, A., and Wittig, S., "Reliable Design of Static Components for Small Gas Turbines," *International Ceramics*, Vol. 1, No. 1, 1996, pp. 91–96.

[42]Pfeiffer, A., Schulz, A., and Wittig, S., "Principles of the ITS Ceramic Research Combustor Design: Segmented Flame Tube and Staged Combustion," *26th Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 91-GT-63, Orlando, FL, June 1991.

[43]Münz, S., Schulz, A., and Wittig, S., "Evaluation of a New Design Concept for a Ceramic Flame Tube Under Engine Conditions," *41st Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 96-GT-498, Birmingham, UK, June 1996.

[44]Chandrasekhar, S., *Radiative Transfer*, Dover, New York, 1960.

[45]Koch, R., Krebs, W., Wittig, S., and Viskanta, R., "The Discrete Ordinates Quadrature Schemes for Multidimensional Radiative Heat Transfer," *Journal of Quantitative Spectroscopy and Radiative Transfer*, Vol. 53, No. 4, 1995, pp. 353–372.

[46]Krebs, W., Koch, R., Bauer, H.-J., and Wittig, S., "Three-Dimensional Spectral Radiative Transfer Calculation in a Cylindrical Model Combustor

Using the Discrete Ordinates Method," *Radiative Transfer*, Vol. 1, edited by M. P. Mengüc, Begell, New York, 1995, pp. 355–371.

[47]Krebs, W., Koch, R., Ganz, B., Eigenmann, L., and Wittig, S., "Effect of Temperature and Concentration Fluctuations on Radiative Heat Transfer in Turbulent Flames," *26th International Symposium on Combustion*, 1996, pp. 2763–2770.

[48]Ganz, B., Schmittel, P., Koch, R., and Wittig, S., "Validation of Numerical Methods at a Confined Turbulent Natural Gas Diffusion Flame Considering Detailed Radiative Transfer," American Society of Mechanical Engineers, Paper 98-GT-228, 1998.

[49]Ganz, B., Koch, R., Krebs, W., and Wittig, S., "Spectral Emissivity Measurements of Thermal Barrier Coatings," *7th AIAA/ASME Joint Thermophysics and Heat Transfer Conference*, HDT 357-1, 1998, pp. 291–296.

[50]Mayle, R. E., "The Role of Laminar-Turbulent Transition in Gas Turbine Engines," *Journal of Turbomachinery*, Vol. 113, No. 3, 1991, pp. 509–537.

[51]Savill, A. M., "Further Progress in the Turbulence Modelling of the By-Pass Transition," *Engineering Turbulence Modelling and Measurements*, Vol. 2, edited by W. Rodi and F. Martelli, Elsevier Science, New York, 1993, pp. 583–592.

[52]Sieger, K., Schulz, A., Crawford, M. E., and Wittig, S., "An Evaluation of Low-Reynolds-Number $k$–$\varepsilon$ Turbulence Models for Predicting Transition Under the Influence of Free-Stream Turbulence and Pressure Gradient," *Engineering Turbulence Modelling and Measurements*, Vol. 2, edited by W. Rodi and F. Martelli, Elsevier Science, New York, 1993, pp. 593–602.

[53]Schmidt, R. C., and Patankar, S. V., "Two-Equation Low-Reynolds-Number Turbulence Modeling of Transitional Boundary Layer Flow Characteristic of Gas Turbine Blades," NASA CR 4145, 1988.

[54]Sieger, K., Schiele, R., Schulz, A., and Wittig, S., "Gas Turbine Blade Heat Transfer Calculations: An Assessment of Different Low-Reynolds-Number $k$–$\varepsilon$ Turbulence Models," *5th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery*, Elsevier Science, New York, 1994, pp. 454–471.

[55]Mayle, R. E., and Schulz, A., "The Path to Predicting Bypass Transition," *Journal of Turbomachinery*, Vol. 119, No. 3, 1997, pp. 405–411.

[56]Schiele, R., Schulz, A., Kaufmann, K., and Wittig, S., "Effiziente Berechnung des laminar-turbulenten Umschlags in Gasturbinen," *DGLR Jahrbuch 1998*, Deutsche Gesellschaft für Luft- und Raumfahrt, DGLR, Bonn, Germany, Paper JT98-052, 1998.

[57]Launder, B. E., and Spalding, D. B., "The Numerical Computation of Turbulent Flows," *Computer Methods in Applied Mechanics and Engineering*, Vol. 3, 1974, pp. 269–289.

[58]Rodi, W., Mansour, N. N., and Michelassi, V., "One-Equation Near-Wall Turbulence Modeling with the Aid of Direct Simulation Data," *Journal of Fluids Engineering*, Vol. 115, No. 1, 1993, pp. 196–205.

[59]Sieger, K., Schiele, R., Kaufmann, K., and Wittig S., "A Two-Layer Turbulence Model for the Calculation of Transitional Boundary-Layers," *ERCOFTAC Bulletin*, Vol. 24, 1995, pp. 21–25.

[60]Schiele, R., Sieger, K., Schulz, A., and Wittig, S., "Heat Transfer Investigations on a Highly Loaded, Aerothermally Designed Turbine Cascade," *Proceedings of the 12th International Symposium on Air Breathing Engines*, AIAA Publ., edited by F. S. Billig, 1995, pp. 1091–1101.

[61]Launder, B. E., and Sharma, N., "Application of the Energy-Dissipation Model of Turbulence to the Calculation of Flow Near a Spinning Disc," *Letters in Heat and Mass Transfer*, Vol. 1, 1974, pp. 131–138.

[62]Bohn, D., Schönenborn, H., Bohnhoff, B., and Wilhelmi, H., "Prediction of the Film-Cooling Effectiveness in Gas Turbine Blades Using a Numerical Model for the Coupled Simulation of Fluid Flow and Diabatic Walls," *Proceedings of the 12th International Symposium on Air Breathing Engines*, AIAA Publ., edited by F. S. Billig, 1995, pp. 1150–1159.

[63]Bohn, D., Kusterer, K., and Schönenborn, H., "3-D Numerical Simulation of the Flow Through a Turbine Blade Cascade with Cooling Injection at the Leading Edge," *42nd Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 97-GT-247, Orlando, FL, June 1997.

[64]Heselhaus, A., and Vogel, D. T., "Numerical Simulation of Turbine Blade Cooling with Respect to Blade Heat Conduction and Inlet Temperature Profiles," AIAA Paper 95-3041, 1995.

[65]Kao, K.-H., and Liou, M. S., "On the Application of Chimera/Unstructured Hybrid Grids for Conjugate Heat Transfer," *41st Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 96-GT-156, Birmingham, UK, June 1996.

[66]Ho, Y.-H., Athavale, M. M., Forry, J. M., and Hendricks R. C., "Numerical Simulation of Secondary Flow in Gas Turbine Disc Cavities, Including Conjugate Heat Transfer," American Society of Mechanical Engineers, Paper 96-GT-67, 1996.

[67]Papanicolaou, E. L., Giebert, D., Koch, R., and Schulz, A., "A Discretization Approach for Conjugate Heat Transfer and Application to Turbomachinery Flows," *Proceedings of the ASME*, Heat Transfer Div., Vol. 3, HTD Vol. 361-3/PID-Vol.3, edited by R. A. Nelson, Jun, L. W. Swanson, M. V. A. Bianchi, and C. Camci, American Society of Mechanical Engineers, Fairfield, NJ, 1998, pp. 349–362.

[68]Giebert, D., Bauer, H.-J., and Wittig, S., "A Comparative Study of Two Low-Reynolds Number $k$–$\varepsilon$ Models for Recirculating Flows with and Without Heat Transfer," *Engineering Turbulence Modelling and Experiments*, Elsevier Science, New York, Vol. 3, edited by W. Rodi and G. Bergeles, 1996, pp. 665–674.

[69]Abe, K., Kondoh, T., and Nagano, Y., "A New Turbulence Model for Predicting Fluid Flow and Heat Transfer in Seperating and Reattaching Flows," *International Journal of Heat and Mass Transfer*, Vol. 37, 1994, pp. 139–151.

[70]Wittig, S., Jacobsen, K., Schelling, U., and Kim, S., "Heat Transfer in Stepped Labyrinth Seals," *32nd Annual International Gas Turbine and Aeroengine Congress and Exposition*, American Society of Mechanical Engineers, Paper 87-GT-92, Anaheim, CA, May–June 1987.

[71]Waschka, W., Wittig, S., Scherer, T., and Kim, S., "Leakage Loss and Heat Transfer in High-Speed Rotating Labyrinth Seals: An Experimental Verification of Numerical Codes," *Proceedings of the Yokohama International Gas Turbine Congress*, Gas Turbine Society of Japan, Vol. 2, 1991, pp. 239–247.