PREDICTING THE ULTIMATE PERFORMANCE
OF ADVANCED POWER CYCLES
BASED ON VERY HIGH TEMPERATURE GAS TURBINE ENGINES

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ABSTRACT

It is well known that the history of gas turbine engines has been characterized by a very clear trend toward higher and higher operating temperatures, a growth which in the past 40 years has progressed at the impressive pace of approximately 13°C/year. Expected improvements in blade cooling techniques and advancements in materials indicate that this tendency is going to last for long time, leading to firing temperatures of over 1500°C within the next two decades.

This paper investigates the impact of such temperature increase on optimal cycle arrangements and on ultimate performance improvements achievable by future advanced gas/steam cycles for large-scale power generation.

Performance predictions have been carried out by a modified, improved version of a computer code originally devised and calibrated for "1990 state-of-the-art" gas/steam cycles. The range of performances to be expected in the next decades has been delimited by considering various scenarios of cooling technology and materials, including the extreme situations of adiabatic expansion and stoichiometric combustion.

The results of parametric thermodynamic analyses of several cycle configurations are presented for a number of technological scenarios, including cycles with intercooling and reheat. A specific section discusses how the optimum configuration of the bottoming steam cycle changes to keep up with exhaust gas temperature increases.

Calculations show that, under plausible assumptions on future technology advancements, within two decades the proper selection of plant configuration and operating parameters can yield net efficiencies of over 60%.

NOMENCLATURE

\[ b \] Mass flux ratio \( G_{cl}/G_{g} \)

\[ b_{crit} \] Value of \( b \) yielding zero heat flux

\[ B_{iw} \] Blade wall Biot number

\[ c_{p} \] Constant-pressure specific heat \([\text{J/kg-K}]\)

\[ C_{f} \] Mass flux \([\text{kg/s-m}^2]\)

\[ h \] Heat transfer coefficient \([\text{W/m}^2\text{-K}]\)

\[ m \] Mass flow rate \([\text{kg/s}]\)

\[ p \] Pressure \([\text{Pa}]\)

\[ Pr \] Prandtl number

\[ q \] Heat flux \([\text{W/m}^2]\)

\[ r_{fc} \] Fraction of cooling flow used for film cooling

\[ Re \] Reynolds number

\[ SP \] Turbomachines size parameter, see Ch.4

\[ St \] Stanton number

\[ T \] Temperature \([\text{K}]\)

\[ T_{bmax} \] Maximum allowed blade temperature \([\text{K}]\)

\[ Z \] Convection cooling parameter, see Eq.(1)

\[ \beta \] Pressure ratio

\[ \eta \] Efficiency

\[ \eta_{II} \] Second law efficiency

\[ \eta_{p.o,SP>1} \] Compressor polytropic efficiency, for \( SP > 1 \)

\[ \eta_{p.t,SP>1} \] Turbine polytropic efficiency, for \( SP > 1 \)

\[ \rho \] Density

Subscripts

\( bg \) Blade surface, gas side

\( cl \) Coolant

\( g \) Gas

\( in \) Inlet

\( nz \) First nozzle

\( gr \) Gas recovery conditions

Acronyms

CC Combined Cycles

HRSG Heat Recovery Steam Generator

MST Maximum Steam Temperature

RFH Regenerative Feedwater Heater

TIT Turbine Inlet Temperature

TOT Turbine Outlet Temperature

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1. INTRODUCTION

Increases in turbine inlet temperature, now closely approaching 1300°C, along with improved aerodynamics, advances in cooling technology and larger pressure ratio, have more than doubled the efficiency of heavy-duty large scale gas turbines in less than 50 years. These improvements have also boosted spectacularly the efficiency of combined gas/steam plants, which in recent years has progressed at a pace of one percentage point per year and has now reached net efficiencies well above 50% (Macchi, 1990). These outstanding accomplishments — along with other well known advantages of gas-turbine-based plants like low investment cost, short construction time, high availability and reliability, low maintenance cost and little dependence on cooling media — have greatly increased the use of gas turbines for base-load electricity generation. Given that the selection of powerplant technology will be more and more affected by environmental issues this trend is likely to last for many years to come because, due both to the intrinsic environmental compatibility of natural gas and the dramatic advances in ultra-low-NOx-burners accomplished in the last decade, combined cycles are much more environmentally benign than conventional steam plants. The superior efficiency of combined cycles plays a decisive role also from the standpoint of the reduction of greenhouse gas emissions: CO₂ specific emissions of a modern natural-gas-fueled combined-cycle are about 60% lower than those of a coal-fired steam plant.

With few exceptions, in the last decade the booming development of combined-cycles has been entirely sustained by natural gas availability. Given the worldwide abundance of natural gas reserves, there are no reasons to believe that the synergy between natural gas and combined cycles is going to end shortly. Moreover, integrated fuel gasification/combined cycle technology is now ready to confirm the superiority of gas-turbine-based power cycles over conventional steam cycles also for scenarios characterized by a large differential between the cost of "premium" (natural gas, distillates, clean crude oils) and "dirty" fuels (coal, heavy oil residuals, biomass, etc).

While conventional steam cycle technology has reached its maturity in the late 50's, with only minor improvements in steam conditions and/or plant arrangements in sight for the future, gas-turbine technology is still in full evolution. Aero-engine long-range projections anticipate core thermal efficiencies around 60%, while by the year "2000+" the advent of advanced cooling techniques and single crystal turbine blades should raise the rotor inlet temperature to 2000°C (Webb, 1991). Even more dramatic developments are anticipated by Sosounov and Boguslaev (1991), who envisage "non-metallic" aircraft engines with uncooled blades, stoichiometric combustion and pressure ratios above 80. Given the incessant technology transfer from the aeronautical field to industrial heavy-duty applications, these projections are definitely relevant also to stationary power cycles. Hence, it is worthwhile to investigate the potential impact of advancements in aero-engine technology on the ultimate performance and cycle arrangements of future gas/steam cycles for large-scale electricity generation.

2. EVOLUTION OF GAS TURBINE TECHNOLOGY

Since its introduction into the market after World War II, gas turbine technology has undergone tremendous developments which, after making it the undisputable leader of aircraft propulsion, are now imposing it as the technology of choice also for stationary power generation. Although this progress is the result of advancements in many areas — turbomachinery, combustion, controls, reliability, economics, etc. — the single most important element has been the spectacular escalation of Turbine Inlet Temperature (TIT)¹ shown in Fig.1. In the last 30 years, TIT of stationary engines has steadily increased at a pace of about 13°C per year, while aircraft engines have experienced even faster progress.

![Graph showing Evolution of Turbine Inlet Temperature of commercial engines.](http://vibrationacoustics.asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1993/78903/V03AT15A074/2403296/v03at15a074-93-gt-223.pdf)

Fig.1: Evolution of Turbine Inlet Temperature of commercial engines as provided by Maslak (1992), Day (1992), Roberts (1992) and Joyce (1992). The lines for ABB and MHI turbines are an estimate based on values of TIT ISO provided by Vogel (1992) and of combustor outlet temperature reported by Kano (1991). Engines used for aircraft propulsion typically operate at TIT higher than those adopted in heavy-duties, being designed with the latest technology for shorter lives. Stationary versions of aircraft turbines are always derated by decreasing TIT, to warrant a duration economically competitive with heavy-duties.

¹ Following a convention widely used among manufacturers and in the technical literature, here TIT is defined as the total temperature at the first rotor inlet. The difference between TIT and the Compressor Outlet Temperature (COT) is due to gas-coolant mixing in the first nozzle. An alternate definition often used by European manufacturers — hereby indicated as TITISO — is the temperature ideally obtained after mixing the gas at the combustor exit with the whole coolant flow.
The reason why TIT augmentation has been pursued so forcefully lies in its very high pay-off. Differently from the steam Rankine cycle, in a gas Brayton cycle turbine expansion work is typically 2-3 times the net work; thus, given all things equal, any increase of turbine work brought about by higher TIT translates to much larger increases of net work. The climb of TIT documented in Fig. 1 has been accomplished by acting along three directions:

- Increase of cooling flows
- Enhancement of material temperature resistance
- Improvement of blade cooling technology

Material improvements are responsible for almost one third of the temperature increase experienced since 1950 (160–180°C; the rest has been accomplished by the combined effects of better cooling technology and larger cooling flows.

### 2.1 Cooling flows

It is important to emphasize that cooling flow augmentation and cooling technology (or materials) enhancement produce different effects on cycle performances. The situation is exemplified in Fig. 2. Given the materials and the cooling technology, there exists a value of TIT which maximizes efficiency (a similar trend is encountered also for specific work): beyond such value the benefits brought about by higher TIT are outweighed by the penalties due to cooling (cooling air by-passes the combustor realizing a cycle with zero heat input and negative net work). For better cooling technology and/or materials the maximum of $\eta$ shifts rightward and upward. The trade-off between $m_{\text{cl}}$ and TIT which produces the maximum of $\eta$ is missing only in adiabatic (uncooled) engines; in such case the growth of $\eta$ is limited only by thermodynamic considerations discussed later.

Fig. 2 explains why, despite impressive advancements in materials and cooling technology, cooling flow requirements have also steadily increased with time. Starting from an initial optimal point E0/F0, an advancement of cooling technology or materials opens the following possibilities:

- maintain the same TIT, thereby reducing cooling flow (points E1/F1)
- increase TIT and maintain the same cooling flow (points E2/F2)
- increase both TIT and cooling flow to maximize efficiency (points E3/F3)

To reduce the risks inherent to new technologies, introductory models will presumably operate at point E1 or E2; however, mature versions will most likely operate at point E3. These observations clearly show that the evolution of cooling technology and materials is intimately related to the evolution of TIT and cooling flow fractions, and that the "histories" of these crucial elements cannot be separated from each other.

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2 Tracing the historical evolution of cooling flow fractions is extremely difficult because manufacturers regard such information as highly confidential. Nonetheless, the steady increase of the cooling flow fraction is well-known: the next generation of heavy-duty engines is expected to operate with cooling flows of over 20% of compressor inlet flow (Kano et al., 1991).

For combined cycles: the qualitative behavior of $\eta$ vs. TIT is still the same, with the difference that the maximum of $\eta$ is shifted toward much higher values of TIT.

![Fig. 2: Qualitative behaviour of TIT and simple cycle efficiency vs. cooling flow for two different cooling technologies and/or materials. The maximum of $\eta$ is due to the trade-off between the advantages of higher TIT and the penalties of cooling. The qualitative behaviour shown in the figure would not be affected by increases of pressure ratio to take advantage of higher TIT; yet, higher pressure ratios would imply higher coolant temperatures and larger cooling flows, and thus a steeper descent of $\eta$ at high TIT.](image_url)
For these reasons, it would be inappropriate to use the values of Fig.3 for cycles analyses. Considering that: (i) this work is concerned with stationary applications and (ii) peak temperatures can exceed average surface temperatures by as much as 100°C, the values of the average surface temperature \( T_{\text{bmx}} \) used to produce the results presented here are:
- 800°C (turbine) and 830°C (nozzle) for today's technology
- 900/930°C for the "2010 scenario" discussed in Ch.5.

The corresponding average increase of 5°C per year — slightly optimistic compared to the past 40 years — is meant to embody potential improvements afforded by Thermal Barrier Coatings (TBC), by novel manufacturing technologies (e.g., single crystal) and by faster technology transfer from aircraft applications.

![Figure 3: Evolution of temperature capability of gas turbine materials according to Maslak (1992) and Day (1992). “DS” stands for directionally solidified, “SC” for single crystal. Many of the super-alloys indicated in the figure have been developed by gas turbine manufacturers themselves. Notice that Udimet 500 (U-500) operates at different temperatures depending on the application (heavy-duty or aircraft engine), a situation shared by many other materials.](image)

**2.3 Cooling Technology**

As already mentioned above, much of the increase in turbine inlet temperatures experienced in the past 40 years must be ascribed to improved cooling technology. Blade cooling has evolved through a variety of different techniques: from simple convection schemes consisting of straight circular channels running from the root to the tip of the blade, to the elaborate schemes comprising enhanced convection, impingement and film cooling adopted in state-of-the-art engines.

Unlike for TIT and materials, predicting the pace of future technology improvements and defining a scenario for a specific date is extremely difficult. One possible course of action could be to extrapolate the historical evolution of the blade cooling parameters \( Z, B_{\text{bmx}} \) and \( r_{\text{fc}} \) (see further); however, the major uncertainties hindering the reconstruction of such historical evolution would make the results highly questionable. For this reason, the future situation will be represented by a range comprised between the current state-of-the-art and the conditions corresponding to ultimate, "ideal" cooling technology.

**2.3.1 Setting the boundaries of future developments**

Despite the intimate connection with the progress of materials and TIT, the evolution to be expected from cooling technology can be appraised on its own because — no matter which TIT and materials are used, and provided that all constraints related to thermal fatigue, thermal stress, reliability, etc. are adequately met — the cooling technology of choice will always be the one that requires the lowest cooling flow.

To identify the limit corresponding to the ultimate cooling technology let's consider the mechanisms controlling blade cooling. The situation can be exemplified by considering the blade as a flat plate with hot gas flowing on one side and cooling air on the other side. The temperature of the gas-side surface of the plate must be kept below a certain maximum value \( T_{\text{bmx}} \): The cooling flow can be reduced by:

1. Enhancing coolant-side heat transfer, thus increasing the heat which can be picked up by each kg of coolant.  
2. Reducing the plate heat resistance, thus increasing the temperature of the coolant-side surface and — like in 1) — the heat which can be picked up by a given mass of coolant.  
3. Ejecting the coolant onto the gas-side surface, thus shielding it from the gas and reducing the incoming heat flux.

Another possibility consists of applying protective TBC onto the gas-side surface; however, this is conceptually equivalent to increasing \( T_{\text{bmx}} \) and can be disregarded.

The three mechanisms above can be brought to extreme conditions by what is called "transpiration cooling", whereby the plate consists of a porous matrix, with the coolant uniformly transpiring across it. In this way:

- Coolant-side heat pick up is maximized by ejecting the coolant exactly at \( T_{\text{bmx}} \) (see Appendix A).  
- There is no wall heat resistance.  
- The heat flux coming from the gas is uniformly and effectively reduced across the whole surface.  

It is important to emphasize that this limiting situation still requires cooling; in other words, as long as \( T_{\text{bmx}} \leq T_{\text{gr}} \) the gas turbine will always require some cooling, even with "ideal" cooling technology.

**2.3.2 Appraisal of convection/film cooling technology**

The gas turbine cooling flows required for the cycle configurations and conditions discussed in Ch.5 have been estimated by a model first developed by Consonni (1992) for coupled convection/film cooling, with extensions to transpiration as illustrated in Appendix A. Impingement is not considered both because it is generally used only for limited portions of the blade surface and because, compared to convection, it would not produce substantial reductions of cooling flow (Consonni, forthcoming).

The cooling flow is calculated by assuming that the
section to be cooled behaves like the tube of a cross-flow heat exchanger subject to the constraint that its gas-side temperature does not exceed \( T_{b, max} \). Although this schematization neglects 2- and 3-D effects — as well as constraints imposed by fatigue and thermal stresses — it does embody the most important physical aspects of the heat transfer process and it appears fully adequate for the purposes of cycle analysis. Let us recall the three parameters used to quantify the "quality" or the "extent", of mechanisms 1, 2) and 3) quoted in the preceding paragraph:

1) The "quality" of coolant-side convection heat transfer is epitomized by:

\[
Z = \psi_I \alpha_b \cdot \nu_p \cdot E_k \cdot (c/d)^{1.2}
\]

where \( \psi_I = \text{"interference" coefficient accounts for heat conduction among coolant channels; } \alpha_b = \text{overall cooling channel cross-section, } c = \text{blade chord; } \nu_p = \text{number of cooling channel passages along the radial direction; } E_k = \text{coolant-side heat transfer enhancement factor (multiplies } h_{cl} \text{ given by Colburn equation); } d = \text{hydraulic diameter of cooling channels. The limit of transpiration cooling corresponds to } Z = \infty, \text{ i.e. zero coolant-side heat resistance.}

2) The blade wall heat resistance is quantified by the following Biot number:

\[
B_i_{bw} = \frac{h_{bw}}{k_b}
\]

where \( h_{bw} = \text{gas-side heat transfer coefficient; } k_b = \text{blade wall thickness, } k_b = \text{thermal conductivity. The limit of transpiration cooling corresponds to } B_i_{bw} = 0, \text{ i.e. zero blade wall heat resistance.}

3) The "quality" of film cooling is expressed by the ratio:

\[
r_{fc} = \frac{\text{cooling flow used for film cooling}}{\text{total cooling flow}}
\]

which is generally much lower than unity because part of the coolant is ejected at the trailing edge and at the blade tip without film-cooling the blade surface. Transpiration cooling does not necessarily correspond to \( r_{fc} = 1 \) because, while the heat flux reduction obtained with transpiration depends only on cooling flow, the reduction accomplished with film cooling also depends on the geometrical arrangement of the film cooling holes; thus, an equivalence between transpiration and film cooling requires to specify the geometrical arrangement.

2.3.3 State-of-the-art and room for improvements

Based on the materials and the typical operating conditions of modern gas turbines it can be assumed that for today's technology \( B_i_{bw} = 0.5 \). The estimate of realistic values of \( Z \) and \( r_{fc} \) is much more problematic due both to lack of data and to wide variations of design styles and technological level among manufacturers. Consequently, instead of attempting an estimate based on their definition, their values have been calibrated to give the best agreement between the predictions of the calculation model and the performances of actual commercial engines; such calibration (Consonni et al., 1991; Consonni, 1992) suggests that for state-of-the-art engines \( Z = 100 \) and \( r_{fc} = 0.25 \). It must be emphasized that these two values reflect average cooling technology for the whole cooled expansion and for a whole set of engines produced by different manufacturers; the values corresponding to a single cascade of a specific engine may be quite different.

Given that expected increases of TIT and \( \beta \) will translate to higher \( h_{cl} \), it is unlikely that progress in manufacturing technology (lower \( t_{bw} \) and perhaps higher \( k_b \) can decrease \( B_i_{bw} \) more than 50-70%; on the contrary, the increase of \( Z \) and \( r_{fc} \) will be limited only by design and manufacturing capabilities. Based on these considerations, the ranges left open for progress in coupled convection/film cooling technology are:

- For \( Z \): from \( \approx 100 \) to \( \infty \)
- For \( B_i_{bw} \): from \( = 0.5 \) to \( 0.15-0.25 \)
- For \( r_{fc} \): from \( = 0.25 \) to \( 1 \)

The relevance of these potential variations is exemplified in Fig.4, which reports the cooling flow required by the first nozzle of a large heavy-duty engine similar to those now being introduced on the 50 Hz market (GE 9001F, Mitsubishi/Westinghouse 501F, Siemens V94.3). The figure shows that:

- Without film cooling \( m_{cl} \) becomes unrealistically large even for high \( Z \).
- Compared to the "current" set of cooling parameters (point A), the set hypothesized for the year 2010 (Point B with \( Z = 200, B_i_{bw} = 0.25, r_{fc} = 0.5 \) gives an approximately 50% reduction of cooling flow.
- Compared to the 2010 set of cooling parameters, transpiration allows a further 50-60% reduction of cooling flow.
- The marginal pay-off produced by improving each single parameter decreases as the other two improve. In other words, if one of the three mechanisms illustrated in 2.2.3 is very efficient, there are less incentives to improve the others.
- At low \( Z \) the influence of \( B_i_{bw} \) becomes small because the blade wall heat resistance is negligible compared to the coolant-side resistance.

3. EVOLUTION OF STEAM CYCLE TECHNOLOGY

Steam cycles have been the technology of choice for power generation since the beginning of the century. Until the early 60s this supremacy kept on being strengthened by the spectacular efficiency escalation shown in Fig.5, which was accomplished by: (i) improving turbomachine efficiency; (ii) enhancing the cycle thermodynamic "quality" by introducing feed-water heaters, superheating, single and then double reheat; (iii) decreasing auxiliaries and parasitic losses and (iv) most important, increasing maximum steam temperatures and pressures. The results of these developments were quite remarkable: in the late 50s large (over 200-300 MW) central stations with maximum steam temperatures around 540-560 °C reached net plant efficiencies close to 40%, i.e. about 65% of the Carnot efficiency corresponding to the same temperature range.

The virtual halt of the progress in steam cycle technology experienced after 1960 can be understood by recalling one of its fundamental features: the steam cycle is a "closed" cycle, whereby thermal power must be transferred from an
external source to steam by means of a heat exchanger. The surfaces of such heat exchanger must obviously operate at temperatures higher than steam, and therefore any attempt to realize high steam pressures and temperatures calls for huge quantities of expensive, high-strength, oxidation- and corrosion-resistant, high-temperature materials.

The cost and the technological problems posed by this situation have been the single most important element which brought steam technology to maturity and caused the flattening of the curves shown in Fig.5.

The attempts toward further increases of steam conditions performed in the early 60s were so unsuccessful (Borglin, 1989) that since then utilities turned to more conservative designs: most fossil fired steam power plants commissioned in the 80s feature a single reheat cycle, steam conditions around 540-565 °C and 165-185 bar and net efficiencies in the range 38-40%. Thirty years after the commissioning of the renowned Eddystone no. 1 unit, designed for 345 bar and 650°C (Campbell et al., 1963), the metallurgical development needed to provide appropriate and well-tested materials for advanced steam conditions is still in the R&D phase. To the authors' knowledge, the most advanced steam power plants recently built are two LNG-fired, double-reheat 700 MW stations in Kawagoe (Japan), featuring steam conditions of 310 bar, 566°C/566°C/566°C and a net efficiency of 44.8% (Iwanaga et al., 1990); steam temperatures approaching 600°C will be
adopted in a future plant (Miyazaki e Watanabe, 1990). The same maximum pressure and slightly higher temperatures (593°C/566°C/566°C) are proposed by EPRI for an "advanced" 824 MW unit which should reach a net efficiency of 45%; however, its realization would require substantial technical efforts.

Today’s R&D activity on materials for 600-650°C is centered on the development of 12Cr ferritic steels (Barlow et al., 1990; Thornton, 1990) and on advancements in casting and welding technology; the medium term goal is the manufacture of large rotors, casings, steam pipes, valves and bladings at affordable cost and without penalties in quality, reliability and operational flexibility.

### 3.2 Heat Recovery Steam Cycles

Heat recovery steam cycles play a relevant role in achieving the outstanding performance of today's Combined Cycles: in industrial practice, the two-pressure cycle is widely accepted even for small (20-30 MW) size units, while the three-pressure reheat cycle is often proposed for larger heavy-duty gas turbines (Lugand and Parietti, 1990; Farmer, 1992). Today's most advanced utility-size combined cycles feature machine sizes and steam conditions not far from the ones of conventional steam plants: for instance, a module based on two large 200 MW heavy-duty turbines includes a 250 MW class steam turbine with steam conditions typical of fossil-fired steam plants.

Addressing the advancements achievable by combined cycles in the next decades calls for a re-evaluation of traditional steam plant design limits, particularly when considering gas reheat and supplementary firing. Despite the difficulties encountered in the past, it appears unrealistic to assume that no progress whatsoever will be realized in the future; for this reason the technological scenarios considered in Ch.5 have been characterized not only by different assumptions on gas turbine technology, but also by three different steam temperature limits:

- 565°C, corresponding to today's best technology
- a medium term value of 650°C
- a long term value of 800°C

Based on the considerations in Par.3.1, the value of 650°C appears at hand within 10-20 years; the long-term value will require the use of superalloys. Steam pressure has been limited to 350 bar: the achievement of high supercritical values does not represent a technological barrier, as the Kawagoe plant and its predecessors can testify.

Besides higher steam temperatures and pressures, future combined cycles will also feature steam cycle configurations with increasing complexity.

On the high-temperature end, steam reheat should become standard practice because: (i) it increases cycle efficiency by extending the high temperature part of the cycle; (ii) it reduces the LP turbine moisture content, thus allowing higher maximum steam pressures; (iii) heat is transferred from the gas to steam at variable temperature, thereby reducing heat transfer irreversibilities.

On the low-temperature end there may be modifications to the economizer section. When the gas turbine outlet temperature (TOT) exceeds 750-800°C and/or the thermal power used for steam reheat is small, steam production is so large that the gas heat capacity is insufficient to heat the liquid. In such case the "quality" of the thermodynamic cycle can be improved by introducing regenerative feedwater heaters (RFH) fed with steam bled from the turbine; as discussed by Lozza (forthcoming), this translates to higher steam production — and thus larger power output — for the same stack temperature. The plant lay-out becomes increasingly complex, because the liquid flow must be split between RFH heat exchangers and the HRSG economizer.

### 4. METHOD OF CALCULATION

The performances of gas-turbine-based cycles discussed in the next chapters have been calculated by a computer program whereby the plant is modeled by assembling a network of nine basic components: compressor, gas turbine expander, splitter, mixer, heat exchanger, combustor, pump, steam cycle and shaft. By specifying the component interconnections it is virtually possible to study any cycle configuration. The calculation method, the assumptions and the validation vs. existing engines have been described in a previous paper (Consonni et al., 1991). Here we will summarize the most significant features and outline the modifications undertaken to account for future technology developments:

- **Compressor:** polytropic efficiency is evaluated as a function of the size parameter \(SP = \sqrt[0.5]{\Delta h_{is,stage}}\), where \(V\) is a properly averaged value of the volume flow rate (see functional expression in Tab.1).

- **Gas turbine:** the cooled expansion is calculated as a sequence of small steps, each consisting of an expansion followed by gas-coolant mixing. At each step, the coolant flow required to maintain \(T_{binx}\) within an assigned value is found by the heat flux balance across the blade wall. Spent coolant is injected into the main flow and is accelerated at the expense of static pressure drops; the conditions before injection are evaluated after accounting for pumping in rotating cascades and for pressure drops from the bleed points. The expansion polytropic efficiency is a function of the size parameter \(SP\) already defined above. Exit kinetic energy is partly recovered by a diffuser.

- **Combustor:** except for stoichiometric cycles, composition and properties of combustion gases are calculated by assuming complete fuel oxidation. In stoichiometric cycles this hypothesis has been removed, and replaced by a chemical equilibrium calculation performed according to the method of element potentials developed by Reynolds (1986). The chemical species considered at equilibrium are \(\text{Ar, CO, CO}_2, \text{H}_2O, \text{N}_2, \text{NO}, \text{O}_2\); the addition of other species was found to have negligible effects on the thermodynamics of the cycle. Downstream of the combustor the gas composition is "frozen" for the remainder of the cycle. The two hypotheses of full oxidation and chemical equilibrium mark the extremes of the range which can be covered. The calculation of actual
operating conditions requires complex chemical kinetic calculations and estimates of hot section residence times which are beyond the scope of this investigation.

- **Steam cycle**: this component consists of a very large number of "sub-components": network of HRSG heat exchangers, steam turbine, condenser, deaerator, feedwater pumps, auxiliaries, etc. Each evaporation pressure level (up to four, excluding the deaerator pressure) may comprise (i) evaporation and superheating or (ii) reheating of steam bled from a turbine or (iii) both. The arrangement of the HRSG and the prediction of steam turbine efficiency are discussed in other papers (Consonni et al., 1991, Lozza, 1990, Lozza and Bombarda, 1991). The HRSG calculation method was modified to handle RFHs, which now are automatically activated when the gas heat capacity is insufficient to heat the liquid (details in Lozza, forthcoming); the gas is always cooled down to the minimum stack temperature by heating as much liquid as possible, while RFHs provide heat to the remaining liquid flow fraction.

4.1 Cycle optimization

The calculation method includes a numeric procedure allowing the optimization of a multi-variable non-linear function under both linear and nonlinear constraints. This procedure has been systematically utilized to maximize the cycle efficiency by acting on cycle parameters like steam evaporation pressures and, if present, intercooling and reheat pressures. The results reported in the next chapters always refer to optimized cycles.

4.2 Assumptions

The cycle calculation requires a large number of assumptions: the most important ones are listed in Tab.1 and are, in the authors’ opinion, a good representation of the state-of-the-art. The air flow of 600 kg/s — typical of large, 200 MW class heavy-duty gas turbines — sets the scale of the plant and is relevant to the evaluation of turbomachine performances. The air flow of 600 kg/s is also typical of large, 200 MW class heavy-duty gas turbines — sets the scale of the plant and is relevant to the evaluation of turbomachine performances.

5. DISCUSSION OF RESULTS

5.1 Technological scenarios

Performance predictions have been carried out with reference to the following scenarios (Tab.2):

| Tab. 1: Main assumptions adopted to derive the results discussed in Ch.5. The underlined values refer to present state-of-the-art, and change according to the scenarios described in Ch.5. |
|----------------|-----------------|
| Air flow at compressor inlet | 600 kg/s |
| Ambient air | 15°C, 1.013 bar, 60% RH |
| Fuel: methane | 15°C, 40 bar, LHV = 50.01 MJ/kg |
| Fuel compression | Isothermal, η = 55%, Δp/p_{max} ≥ 50% |
| Inlet air pressure loss | 1 kPa |
| Combustor pressure loss | 3 kPa |
| Discharge pressure loss | 1 kPa in simple cycles and 3 kPa with HRSG |
| Intercooler pressure loss | 3 kPa |
| Intercooler air exit temperature | 40°C |
| Air leakage at compressor outlet | 0.8% |
| Compressor air leakage | 0.4% of input heat |
| Compressor polytropic efficiency | 0.896 \times max(1, 1 - 0.02688 \times \log_{10}(SP)) |
| Turbine polytropic efficiency | 0.891 \times max(1, 1 - 0.02688 \times \log_{10}(SP)) |
| Parameters involved in blade cooling | \begin{align*}
Z &= 100, r_c = 0.25, B_{bw} = 0.5
\end{align*} |
| Maximum metal temp. (nozzle/other cascades/reheat combustor) | 830/800/900°C |
| Minimum/maximum steam generation pressure | 3/350 bar |
| Minimum/maximum steam reheat pressure | 15/110 bar |
| Condensing/deaerator pressure | 0.05/1.4 bar |
| Maximum steam temperature | 565°C |
| Minimum gas exit temperature | 80°C |
| Minimum ΔT at pinch-point/approach-point | 10/25°C |
| Liquid subcooling at drum inlet (only if p < 170 bar) | 10°C |
| Heat losses from HRSG | 0.7% of available heat |
| RH/SH/economizers pressure losses | 8/8/20% |
| Cooling system auxiliaries power | 0.5% of rejected heat |
| Steam turbine revolution speed | 3000 RPM |
| Steam turbine leaving loss | 24.2 kJ/kg (220 m/s) |
| Equivalent steam leakage in turbine | 1% of live steam flow |
| Compressor/gas/steam turbine organic loss | 0.3/0.3/0.5% |
| Electric generator efficiency | 99% |

(a) for a complete list refer to Consonni et al., 1991
(b) valid for outputs ≥ 100 MW - for smaller units, refer to Lozza, 1990.

A) Present-state-of-the-art, corresponding to the most advanced heavy-duty gas turbines introduced into the market in the early 90's. The relevance of turbomachinery efficiency is investigated by considering the case A with η_{pc} and η_{in} increased by one percentage point.

B) Technological level which presumably will be reached within two decades with higher operating temperatures, better turbomachinery aerodynamics and advances in cooling techniques. The TIT increase of 250°C is stipulated according to the trend depicted in fig.1, and is partly ascribed to an increase of 100°C of material temperature capability (refer to fig.3 to compare with

---

5 The steam expansion line is calculated by a stage-by-stage procedure whereby efficiency is related to volume flow rate and revolution speed. The method includes corrections for moisture and accounts for steam leakages, moisture removal effectiveness, turbine discharge area and flow splitting.
Tab. 2: Hypotheses defining the scenarios considered in figures 6, 11 and 12.

<table>
<thead>
<tr>
<th>Scenario</th>
<th>TIT, °C</th>
<th>( \frac{T_{bmax,ae}}{T_{bmax,cl}} ) °C</th>
<th>( \frac{\eta_{ps,ae}}{\eta_{ps,cl}} )</th>
<th>Cooling technique</th>
<th>MST, °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>1250</td>
<td>830/800</td>
<td>0.896/.921</td>
<td>100/.25/.5</td>
<td>565</td>
</tr>
<tr>
<td>A'</td>
<td></td>
<td></td>
<td>0.906/.931</td>
<td>200/.5/.25</td>
<td></td>
</tr>
<tr>
<td>B-</td>
<td>1500</td>
<td>930/900</td>
<td>0.906/.931</td>
<td>100/.25/.5</td>
<td>650</td>
</tr>
<tr>
<td>B</td>
<td></td>
<td></td>
<td>200/.5/.25</td>
<td>transpiration</td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>1500</td>
<td>&gt;TIT</td>
<td>0.906/.931</td>
<td>uncooled</td>
<td>650</td>
</tr>
<tr>
<td>D</td>
<td>stoich. (full oxid.)</td>
<td>&gt;TIT</td>
<td>0.906/.931</td>
<td>uncooled</td>
<td>800</td>
</tr>
<tr>
<td>D'</td>
<td>stoich. (chem. eq.)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 6: Efficiency and specific work of simple gas turbine cycles at various compression ratios, under the technological scenarios defined in Tab. 2.

5.2 Simple cycle gas turbines

Fig. 6 presents results for simple cycle turbines. The physical significance of the parameters used as coordinate axes is well-known: efficiency defines the plant performance in terms of fuel consumption — by far the major cost item for base-load gas-turbine-based plants — as well as thermal pollution and CO\(_2\) emissions. Specific work is related to the power output per unit of turbomachine cross flow area, thus setting the limit of single-shaft plant power output and establishing a connection with plant investment cost.

Today's heavy-duty machines (curve A) operate at pressure ratios corresponding to maximum specific work (\( \beta = 15 \)), thereby suffering an efficiency penalty of about 5 percentage points (36% vs. 41%) with respect to operation at the high pressure ratios adopted in aero-derivative units (\( \beta = 30 \)).

As long as the pressure ratio is moderate, TIT augmentation produces large gains in specific work but not in efficiency: for instance, the efficiency improvements achievable by going from curve A' to curve B (same turbomachinery efficiency, same cooling technology but different TIT) exceed 1 percentage point only for \( \beta > 30 \). The same applies
to cooling technology: for \( \beta < 30 \) the difference between curves \( B^- \) and \( B^+ \), which delimit the range between today's and ultimate cooling technology, are below 1 percentage point. The gain in efficiency is limited even when going from transpiration-cooled (\( B^+ \)) to fully uncooled (C) turbines. At today's pressure ratios (\( \beta \leq 30 \)), cycle efficiency does not exceed 46% even at the temperatures (\( \approx 2300^\circ C \)) produced by stoichiometric combustion (points D).

Fig. 7 illustrates that — similarly to what shown in Fig. 2 but due to a different mechanism — also in the uncooled case there is a value of TIT which maximizes efficiency. At TIT higher than this value the losses due to the discharge of exhaust gases to ambient outweigh the advantage of increasing the average temperature at which heat is introduced into the cycle. The descent of \( \eta \) with TIT is exacerbated by the dissociation of combustion products, which reduces the heat released by the fuel. One more interesting point evidenced by Fig. 7 is that when TIT reaches values above present practice, the full oxidation hypothesis gives results significantly different from those based on chemical equilibrium.

5.3 Bottoming steam cycles

The main flaws of simple cycles are (i) exergy losses due to combustion and (ii) discharge of gases with high exergy content. The results discussed above show that technological improvements and compression ratio augmentation cannot completely eclipse these two drawbacks. On the other hand, the latter can be effectively eliminated by recovering the gas exergy content in a bottoming steam cycle.

Let us consider three-pressure-level cycles with single reheat (reheat pressure equal to IP evaporation pressure) or double reheat (LP reheat pressure equal to IP evaporation pressure), a configuration which achieves high efficiency with acceptable plant complexity. Fig. 8 shows results for optimized cycles operating at the three values of maximum steam temperature (MST) discussed in Par. 3.2.

The second-law efficiency \( \eta_{\text{II}} \) on the ordinate axis is defined as the ratio between net electric steam turbine power output and gas exergy content (the reference for exergy is \( T_0 = 15^\circ C \)). The figure shows that:

- \( \eta_{\text{II}} \) increases sharply until it reaches a maximum for \( T_g \) about 150-200°C higher than MST and then slowly decreases. In all cases, the maximum of \( \eta_{\text{II}} \) occurs for \( T_g \) much higher than \( T_0 \) of present gas turbines.
- The lower \( \eta_{\text{II}} \) realized at moderate \( T_g \) are mainly due to

The advantages of three- vs. two-pressure-level cycles are significant over a wide range of gas temperatures (Lozza and Bombarda, 1991). A higher number of evaporation pressure levels would not achieve significant improvements.
incomplete heat recovery. This can be justified by observing that (i) stack temperatures are high (the minimum value of 80°C is reached only when \( T_g > 650°C \)); (ii) for the same stack temperature, the heat fraction discharged is larger when \( T_g \) is low.

- At very high \( T_g \) efficiency decreases due to large AT in the hot HRSG section. In this gas temperature range an increase of MST would be very appealing: going from 565°C to 800°C increases steam plant output by 10%.
- Maximum \( \eta_t \) vary from 72 to 79%, a range which places steam recovery cycles among top-class energy conversion systems, and confirms that steam is an excellent working fluid for CC applications.
- Double reheat can increase cycle power output by 2-4%, but only at high \( T_g \).
- At \( T_g = 650°C \) heat recovery is complete. At higher gas temperatures part of the feedwater is heated by RFHs.
- As \( T_g \) increases, IP and LP steam production decrease and eventually disappear (see dots in fig.8). High MST and/or double reheat shift these occurrences toward higher \( T_g \) because steam production is lower and there is more room for LP/IP steam generation.

High gas temperatures change substantially the optimum cycle lay-out: fig.9 shows the plant configuration for optimized cases at 600°C and 1000°C (MST = 565°C). In the first case, the HRSG includes a quite complicated network of heat exchangers, with significant LP and IP steam production; the maximum pressure is supercritical, while the optimum reheat pressure is moderate (15 bar) in order to benefit at best from variable temperature heat transfer. When \( T_g = 1000°C \) the optimized double-reheat cycle is single-pressure (350 bar), with feedwater heating partially accomplished by regenerative heat exchangers. Notice that the water flow fraction handled by RFHs is variable: since the specific heat of water increases with temperature — while the gas heat capacity is approximately constant — RFH2 must provide more heat than RFH1.

### 5.3.1 Supplementary firing

The impact of supplementary firing on the overall combined cycle efficiency can be appraised by Fig.10. The "supplementary firing efficiency" \( \eta_{sf} \) on the ordinate axis is the ratio between the additional steam plant output and the extra heat input due to an infinitesimal increase of \( T_g \) (based on 100% combustion efficiency, which is a very good approximation of actual systems). If \( \eta_{sf} \) exceeds \( \eta_{CC} \), we can say that supplementary firing would improve overall efficiency. The average \( \eta_{sf} \) corresponding to a finite temperature increase can be found by integrating the curve of \( \eta_{sf} \) vs. gas enthalpy. The figure shows that:

![Fig.9: Plant scheme and relevant flow parameters for a three-pressure-level reheat cycle with gas temperature of 600°C (upper part) and for a single-pressure double-reheat cycle with gas temperature of 1000°C (lower part). The percentages shown over regenerative heat exchangers (RFH0-1-2) indicate the fraction of total liquid flow handled by each of them (the remainder is heated in the HRSG). \( P_{gm} \) and \( P_{en} \) indicate gross mechanical and net electric power outputs (MW), respectively.](image-url)
• \( \eta_{af} \) largely exceeds the steam cycle efficiency across the whole temperature range because (i) the added steam cycle portion is a high-temperature, high-efficiency cycle and (ii) in most cases higher \( T_g \) result in lower stack temperatures, implying that the additional heat that enters the cycle is larger than the heat introduced by supplementary firing;

• For \( MST = 565^\circ C \) \( \eta_{af} \) rarely exceeds today's CC efficiency, so that significant overall efficiency improvement cannot be expected. However, one should consider that steam plant output will sharply increase at about the same efficiency, with potential interesting economic benefits; this is the case, for instance, of aero-derivative-based CCs, which are generally penalized by small-size, low-temperature steam turbines.

• \( \eta_{af} \) improves at higher MST: for instance, with \( MST = 650^\circ C \) and gas temperatures ranging 600-800°C, \( \eta_{af} \) largely exceeds 55%. However, since future unfired CC should exhibit even higher efficiencies (see fig.11), it is unlikely that supplementary firing will be used to boost efficiency.

• At the highest gas temperatures \( \eta_{af} \) tends to the efficiency of a conventional steam cycle, i.e. to values much lower than those of CCs.

It must be noticed that the results of fig.10 hold only if it is possible to achieve very low stack temperatures; if supplementary firing is carried out by low quality fuels (heavy oil, coal) with high sulphur content, the stack temperature must be higher and \( \eta_{af} \) will decrease significantly.

\[ \eta_{af} = \frac{W_{net}}{Q_{in}} \]

**Fig.10:** Electric conversion efficiency of an infinitesimal amount of heat added to the exhaust gases as a function of gas temperature. The discontinuities occurring at \( T_g = MST + \Delta T = 25^\circ C \) are due to the different slopes of the constant-approach and constant-MST lines of fig.8.

### 5.4 Unfired combined cycles

Fig.11 shows the performances of unfired gas/steam cycles with gas turbines characterized by the parameters of Tab.2 and the optimized bottoming steam cycles described in the previous section. According to these calculations, today's combined cycles can reach net efficiencies of about 56%\(^7\). The thermodynamic "quality" of these cycles is remarkable, since their efficiency is 70% that of a Carnot cycle operating between temperatures of 1250°C (=TIT) and 32.9°C (steam condensation at 0.05 bar).

Opposite to the situation occurring in simple gas cycles, where cycle efficiency is very sensitive to turbomachinery efficiency, improvements in turbomachinery aerodynamics (curves A'\(^+\)) produce minor effects on the overall efficiency of combined cycles. This is due to the capability of the bottoming steam cycle to compensate for losses occurring elsewhere: for instance, since gas turbine losses eventually result into higher gas discharge temperatures, they can be partly recovered by the steam cycle.

Future gas turbine technological advancements yield remarkable performance improvements: under the scenario stipulated for the year 2010 (curve B), specific work increases by about 40%, while efficiency exceeds 61%. Thermodynamic "quality" is even higher than that of today's scenario (\( \eta \) over 73% of Carnot cycle efficiency).

Although optimum pressure ratios are close to 30, retaining present values around 15 would cause only minor efficiency penalties (less than one percentage point). For a given TIT, improvements in cooling technology yield relatively low efficiency gains: for TIT = 1500°C, the replacement of present cooling technology with transpiration produces less than 1 percentage point gain in overall combined cycle efficiency; uncooled (adiabatic) turbines would give a further gain of about one point.

Turbine outlet temperatures (TOT) do not increase dramatically: a 1500 °C class gas turbine operating at optimum pressure ratio has approximately the same TOT (=590°C) of today's gas turbines: therefore, "conventional" steam conditions (565 °C) will be still adequate. Even at lower pressure ratios TOT is in a region where the benefits brought about by higher steam temperatures are minor: for instance, for \( \beta = 20 \) and TOT=645°C, the adoption of the conventional steam temperature of 565°C gives efficiency and power output penalties of only 0.4%.

The combined cycle will continue to benefit from TIT increases up to stoichiometric combustion, especially if associated to a parallel increase of \( \beta \), reaching ultimate efficiencies above 70% and specific works of 2000 kJ/kg.

### 5.5 Influence of reheat and intercooling

Given the widespread interest aroused by reheat and intercooling (Stamblir, 1992), it is worthwhile investigating the impact of these "complications" for each technological scenario, with the exception of stoichiometric combustion, which is obviously incompatible with reheat.

---

\(^7\) This value is almost 3% higher than the efficiency of 54.5% quoted for recent 200 MW class advanced gas turbines (Farmer, 1992). The difference is due to several assumptions regarding the characteristics of both the steam cycle (more sophisticated HRSG arrangement, higher steam pressure) and the gas turbine (film cooling also downstream the first nozzle).
Fig. 11: Efficiency and specific work of combined gas/steam cycles at various compression ratios, under the technological scenarios defined in Tab. 2.

Fig. 12 shows that for present state-of-the-art and $\beta > 25-30$ reheat would increase efficiency by about 2 percentage points and specific work by about 30%. Further addition of intercooling gives further gains in specific work but no benefits on cycle efficiency and requires larger pressure ratios. Similar results are found also for future scenarios: 2 percentage points efficiency gain with reheat, no further efficiency gain with intercooling, large gains in specific work. These results are not surprising, because the thermodynamic "quality" of unfired combined cycles is so high that the improvements to be expected by "complicating" the shape of the cycle are marginal.

Intercooling may prove more appealing if considered in conjunction with the higher TIT made possible — for the same cooling technology — by the lower temperatures at the compressor exit. In practice, this option can be implemented to boost the performance of existing non-intercooled engines, like proposed by GE with its LM8000 (Anon., 1989): the situation is somewhat similar to that depicted in Fig. 2, with the $\eta$-TIT curve of the intercooled cycle peaking at values of TIT higher than the optimum of the non-intercooled cycle. The detailed investigation of these implications of intercooling goes beyond the scope of this work, especially because such analysis should also consider — for the sake of completeness — other means to increase TIT like pre-cooling of cooling air before its use in the gas turbine.

Regarding reheat, it should be emphasized that its introduction requires substantial technological efforts both for the gas turbine and the steam section. The former must operate at higher $\beta$ and requires new, specifically developed components like the reheat combustor and the reheat turbine. The latter operates with TOTs around 770°C, thus calling for high MST: if MST is limited below 565°C efficiency decreases by about 1%, or approximately one third of the gain obtained with reheat. Finally, the comparison between curves CC-C and RhCC-B shows that uncooled gas turbines produce larger efficiency gains than reheat. This underlines the importance of ceramic materials but, on the other hand, it does not allow to draw conclusions on future developments, because the actual realization of ceramic engines poses challenges fully comparable to those posed by reheat.

5.6 Environmental aspects

Although the focus of this work is just on thermodynamic performance, it is important to emphasize that the prospects of future power generation relies heavily on its environmental implications. With this regard, it is well known that the only concerns aroused by gas-fired combined cycles come from $CO_2$ and $NO_x$ emissions. Very high efficiency and low carbon content of natural gas make the former lower than those of any other fossil-fuel-based power technology, while the latter have recently undergone dramatic reductions due to improvements in combustion technology. Should these improvements be unable to keep up with the pace of tightening emission standards, it would be worthwhile verifying the results obtained here under a constraint on $NO_x$ emissions. This because meeting such constraint may require alternate cycle configurations (e.g. steam injection) or limit some crucial parameter ($\beta$, TIT), thus creating an interaction between emission regulations and the evolution of the thermodynamic cycle.

6. CONCLUSIONS

The results presented in the paper shed light on the performance improvements which should be accomplished by gas turbine technology in the next decades. The development of the power industry will also be influenced by other factors not treated in this paper: availability and cost of fuel, capital costs, environmental regulation, reliability, etc. However, at present it is evident that the radical transformation now
underway - the transition from steam- to gas-turbine-based power plants - will continue, yet it will be strengthened by the same driving forces that originated this trend: "clean" fuels (either natural or synthetic), low capital costs, high efficiency.

According to the thermodynamic calculations carried out in the paper, unfired combined cycles will continue to represent the most attractive plant arrangement for a long time. Even at extremely high TITs, combined cycles can achieve efficiencies of about 70% of the Carnot cycle operating between TIT and the condensation temperature of the bottoming cycle. Based on reasonable assumptions on technological trends, it can be predicted that within two decades large heavy-duty gas turbines will operate at TIT around 1500 °C, with unitary outputs over 300 MW and simple cycle efficiencies over 40%. The combination of these gas turbines with highly complex steam bottoming cycles will produce overall plant efficiencies above 60%.

The thermodynamic "quality" of combined cycles is so good that further cycle modifications — intercooling, reheat, regenerators, bottoming cycle working fluid different from water — have few chances to yield efficiency gains high enough to justify their development.

APPENDIX A: Transpiration cooling flow

To appraise the performance achievable by gas engines with ultimate cooling technology the coupled convection/film cooling model already used in previous investigations (Macchi et al., 1991) has been extended to transpiration cooling.

Contrary to the difficulty of its practical realization, the prediction of transpiration cooling flow is rather simple because, as long as there is no heat input by radiation, the temperature of the coolant ejected at the blade surface must equal the gas-side surface temperature \(T_{bg}\) (Grootenhuis, 1959). Thus, at the design condition with \(T_{bg} = T_{bmx}\) the heat flux convected from the gas must equal the heat flux necessary to take the coolant from its inlet temperature \(T_{cin}\) up to \(T_{b}^n\):

\[
q = h \cdot (T_{g} - T_{b}^n) = G_{cl} \cdot c_{p,cl} \cdot (T_{b}^n - T_{cin})
\]

This equation alone does not solve the problem because the gas-side heat transfer coefficient \(h_g\) is itself a function of \(G_{cl}\). Given the purpose of our work and the trends shown by the experimental data collected by Jeromin (1970), the function \(h_g(G_{cl})\) can be approximated by a linear relationship, an assumption also adopted by El-Masri (1983):

\[
h_g = St_g \cdot G_{cl} \cdot c_{p,g} = St_{g,0} \cdot \left(1 - \frac{b}{b_{crit}}\right) \cdot G_{cl} \cdot c_{p,g}
\]

where \(St_{g,0}\) is the gas-side Stanton number without transpiration, \(b = G_{cl}/G_g\), and \(b_{crit}\) is the value of \(b\) corresponding to the detachment of the boundary layer from the surface, i.e. zero heat flux. The two equations above are satisfied only if:

\[
b = \frac{G_{cl}}{G_g} = \frac{b_{crit}}{1 + \frac{c_{p,g} \cdot b_{crit} \cdot T_{bmx} - T_{cin}}{c_{p,cl} \cdot St_{g,0} \cdot T_{gr} - T_{bms}}}
\]

The value of \(b_{crit}\) as \(Re_g \to \infty\) has been determined theoretically by Kutateladze and Leont'yev (1972). For isothermal and homogeneous (i.e. same gas) injection \(b_{crit}/St_{g,0} = 4\). For the injection of a foreign gas at a temperature different from \(T_g\) it is not possible to reach a closed expression of \(b_{crit}\); however, a good approximation is:

\[
\frac{b_{crit}}{St_{g,0}} = \frac{12}{1 + 2 \cdot \frac{\rho_f}{\rho_g}}
\]
where the coolant density to be used is the one at the blade surface (i.e., at $T_{\text{base}}$). Substitution of this equation into Eq.(6) now gives $G_{cl}/G_s$. The value of $S_{f,0}$—assumed constant over the whole blade—is determined from an empirical correlation which averages experimental data for turbine blades (Consonni, 1992, p.5.14):

$$S_{f,0} = \frac{0.45}{Re_s^{0.37}Pr_s^{0.29}}$$  \hspace{1cm} (8)

The non dimensional cooling flow is finally obtained from:

$$m_{cl} = \frac{G_{cl}}{G_s} S_{cl}$$  \hspace{1cm} (9)$$

where the ratio between surface to be cooled $S_{cl}$ and gas flow cross-section $A_s$ is determined from turbomachinery similarity considerations (Consonni, 1992, App.A).

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