THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

93-GT-57

COMPARATIVE EVALUATION OF COMBINED CYCLES AND GAS TURBINE SYSTEMS WITH WATER INJECTION, STEAM INJECTION AND RECUPERATION

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ABSTRACT

Combined cycles have gained widespread acceptance as the most efficient utilization of the gas turbine for power generation, particularly for large plants. A variety of alternatives to the combined cycle that recover exhaust gas heat for re-use within the gas turbine engine have been proposed and some have been commercially successful in small to medium plants. Most notable has been the steam injected, high-pressure aero-derivatives in sizes up to about 50 MW. Many permutations and combinations of water injection, steam injection, and recuperation, with or without intercooling, have been shown to offer the potential for efficiency improvements in certain ranges of gas turbine cycle design parameters.

A detailed, general model that represents the gas turbine with turbine cooling has been developed. The model is intended for use in cycle analysis applications. Suitable choice of a few technology description parameters enables the model to accurately represent the performance of actual gas turbine engines of different technology classes. The model is applied to compute the performance of combined cycles as well as that of three alternatives. These include the simple cycle, the steam injected cycle and the dual-recuperated intercooled aftercooled steam injected cycle (DRIASI cycle). The comparisons are based on state-of-the-art gas turbine technology and cycle parameters in four classes: large industrial (123-158 MW), medium industrial (38-60 MW), aeroderivatives (21-41 MW) and small industrial (4-6 MW). The combined cycle's main design parameters for each size range are in the present work selected for computational purposes to conform with practical constraints.

For the small systems, the proposed development of the gas turbine cycle, the DRIASI cycle, are found to provide efficiencies comparable or superior to combined cycles, and superior to steam injected cycles. For the medium systems, combined cycles provide the highest efficiencies but can be challenged by the DRIASI cycle. For the largest systems, the combined cycle was found to be superior to all of the alternative gas turbine based cycles considered in this study.

NOMENCLATURE

A	area	m^2
B	blade axial breadth	m
C	degree of reaction factor	-
c_P	specific heat capacity	kJ/(kg K)
l	airfoil perimeter	m
L	blade height	m
$\dot{M}$	mass flow rate	kg/s
Ma	Mach number	-
p	pressure	bar
Pr	Prandtl number	-
PRC	pressure ratio compressor	-
$\dot{Q}$	heat flow	kW
r	recovery factor	-
R	gas constant	kJ/(kg K)
SP	blade circumferential spacing	m
St	Stanton number	-
T	temperature	K or °C
u	velocity	m/s
$\dot{W}$	power	kW
Y	cooling air mixing loss factor	-

<u>Greek</u>

α	heat transfer coefficient	$kW/(m^2 K)$
δ	sum of the relative deviations	-
ε	heat exchanger efficiency	-
η	efficiency	-
κ	ratio of specific heats (c_P/c_v)	-
ρ	density	kg/m^3
σ	technology level descriptor	-
ϕ	angle of coolant injection (see Fig. 3)	
ω	specific work	kJ/kg air

<u>Subscripts and superscripts</u>

a	ambient
ad	adiabatic
AUX	auxiliary
c	coolant, cooling air
CE	combustor exit
co	compressor
F	fuel
g	gas (hot gas expanding through the turbine)
GT	gas turbine

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
Cincinnati, Ohio — May 24–27, 1993

IC	intercooler
M+G	mechanical and generator (efficiency η_{M+G})
P	polytropic
s	free-stream
ST	steam turbine
t	turbine
TE	turbine exit
w	wall

INTRODUCTION

Gas turbine development with respect to performance has mainly relied on increased turbine inlet temperature and improved blade cooling technology. These improvements have had a positive impact especially on combined cycle performance. Today, using a combined cycle for power generation and cogeneration applications is probably the best way to utilize advanced gas turbine technology. Advanced gas turbine cycles with features like steam injection, intercooling and recuperation have the potential of achieving high efficiencies, but have not gained the widespread acceptance that the combined cycle has enjoyed.

When analyzing gas turbine cycles it, has become more and more important to include the impact of turbine cooling on the turbine expansion path and on cycle performance, as turbine inlet temperatures have been raised. Different approaches have been taken in order to include turbine cooling in gas turbine cycle performance calculations. Rohsenow (1955) divided the turbine expansion path into a cooled and an uncooled portion. For the cooled portion a relation for heat transfer to cooled blades per unit of work ($\dot{Q}/\dot{W}$) was derived. This relation was based on the temperature difference between the hot gas and the wall, a defined loss factor, and an expression for maximum work per stage. At that time this was a unique type of solution to determine the quantity of heat rejected by cooling without using extensive heat-transfer analysis. The Rohsenow method applies only to the case where heat is removed from the turbine blades and is rejected to a sink external to the turbine. Another method was proposed by Louis et al. (1983) intended for different types of cooling methods such as internal impingement and convection cooling, film cooling and closed loop cooling by water. In this model, internally cooled blades are treated as heat exchangers operating at constant metal temperatures and the coolant exit temperature is a function of the heat exchanger effectiveness. The coolant requirements and heat transfer with film cooling are determined using a correlation from Louis (1977). A single stage model for the expansion path divided into three portions is used. In the first portion the coolant is mixed with the hot gas, resulting in a drop of stagnation temperature at constant stagnation pressure followed by a drop in stagnation pressure at constant stagnation temperature. The second portion is an adiabatic expansion with an isentropic efficiency, and the third portion is a repetition of the first one. The GASCAN code (Elmasri, 1986a) enables stage-by-stage calculations of the turbine based on the free-vortex design method. The cooling flows to each blade row are found from semi-empirical cooling effectiveness curves (Elmasri and Pourkey, 1986). A different approach is given by Elmasri (1986b) where a closed-form solution is used for the impact of cooling on cycle performance. This model is based on representing the turbine as an expansion path with continuous, rather than discrete, work extraction. However, the closed-form solution requires constant fluid properties. The model for cooled turbines presented in this paper is based on the work by Elmasri (1986b).

GAS TURBINE MODEL

In principle, raising the turbine inlet temperature increases the efficiency and the specific work output of gas turbine cycles. The surfaces of the components exposed to the hot gas must be maintained below a certain safe working temperature, consistent with mechanical strength and corrosion resistance. As the turbine inlet temperature is raised, the temperature difference between the hot gas and the metal surfaces will increase. The cooling requirements will then increase both with respect to the amount of coolant for a given stage and to the proportion of the expansion path which has to be cooled. Cooling of stages in the expansion path counteracts the effect of increased inlet temperature. When, for a given level of cooling technology, the turbine inlet temperature is raised beyond a certain value, the cooling penalties are such that cycle efficiency falls.

In air cooled turbines, mixing the cooling air into the main flow causes losses in both stagnation temperature and pressure. The thermodynamic penalty due to those losses is usually more substantial than that due to heat extraction from the expanding hot gas through the walls. Since those penalties increase with the flow rate of the cooling air, minimizing this flow rate is desirable. This is achieved by improving internal heat transfer between the coolant and the blade and by using the spent coolant to shield the blade from the hot gas, which essentially reduces the blade external heat-transfer coefficient. Both of those effects could be achieved if practical transpiration cooling systems were developed. Today they are partially apparent in the technology of film cooling. The older technology of internal air cooling does not utilize the spent coolant to reduce the external heat-transfer coefficient. Thus, its performance can be considered as a lower limit on film cooling, whereas transpiration cooling represents an upper limit. Other methods of reducing cooling air requirements are currently being developed. One such method which is being applied to some machines, is to externally precool the cooling air going to the first row of stators. The use of other coolant fluids like steam or liquid water has also been proposed.

The objective for the development of a gas turbine model in the present work is to study gas turbine cycle performance. No attempt is made to do a detailed physical stage-by-stage analysis including features such as velocity triangles and prediction of stage efficiencies. Due to the large number of parameters influencing cooling losses, precise calculations can only be attempted for specific machines and conditions. Such computations can yield satisfactory predictions for the performance of a particular machine. To obtain a general understanding of the parameters governing cooling losses, and to compare different levels of machine technology and different gas turbine cycles with respect to performance, one often employs approximate models. Such models should not be simplified to a degree where satisfactory numerical results cannot be provided. In the present work a gas turbine model is developed that is sufficiently simplified to facilitate parametric studies of gas turbine cycle performance, while retaining an adequate level of detail to provide useful numeric results.

The turbine is treated as an expander whose walls continuously extracts work. The stagnation temperature of the gas relative to those walls is approximated by a very-close-to-continuously-varying function, rather than the steplike variation of a real machine. This is depicted in Fig. 1 which shows that the assumed temperature profile underestimates the relative gas-to-surface temperature difference for a stator, while overestimating it for the rotor. The stage average, however, is essentially correct.

The connection between this idealized model of the expansion path and the real machine is accomplished by setting its work flux equal to the machine stage average (Elmasri, 1986b).

$$\frac{d\dot{W}}{dA_w} = \frac{\dot{W}_{stage}}{A_{w,stage}} \qquad (1)$$

The stage work is proportional to the local gas-flow rate and the square of the pitchline velocity. That is

$$\dot{W}_{stage} = C\,\dot{M}_g\,u^2 \qquad (2)$$

where the proportionality constant C depends on the stage geometry and velocity triangles, and is typically in the range $1.0 < C < 1.5$. An element of the expansion path is shown in Fig. 2. The heat flux is

$$\frac{d\dot{Q}}{dA_w} = \alpha\,(T_{ad} - T_w) \qquad (3)$$

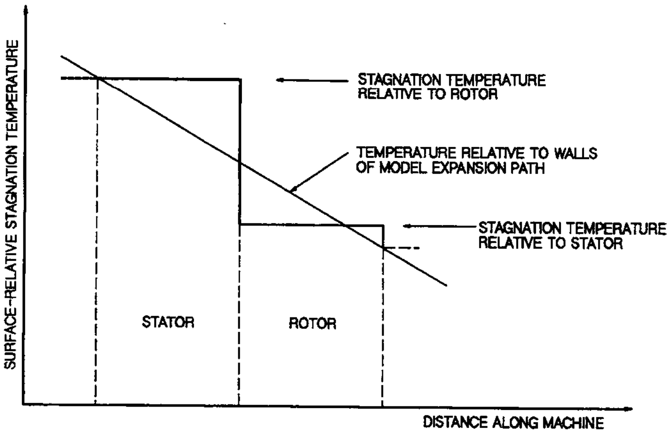

Fig. 1. Schematic representation of the temperature profile in the in the expansion path with continuous work extraction (after Elmasri, 1986b).

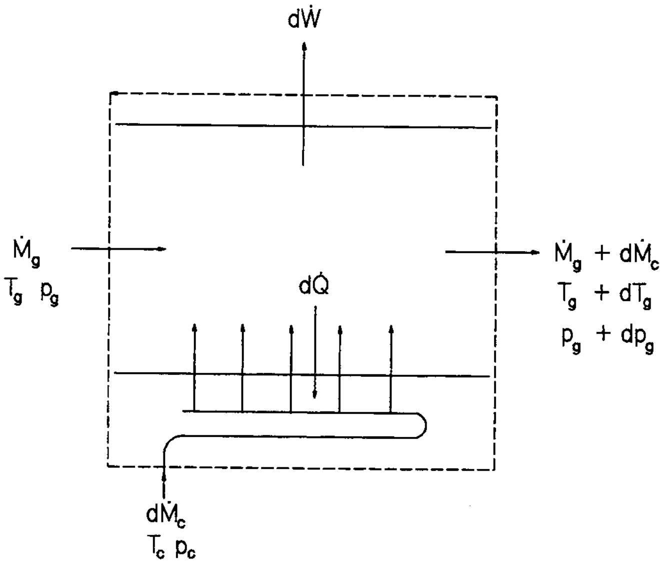

Fig. 2. Element of the cooled expansion path

where T_{ad} is the adiabatic wall temperature. This temperature is close to the gas stagnation temperature, given by the "recovery factor" r,

$$r = \frac{T_{ad} - T_s}{T_g - T_s} \tag{4}$$

where T_s is the free-stream static temperature. For turbulent flow the recovery factor can be expressed as a function of the Prandtl number (Kreith and Black, 1980), i.e.,

$$r = Pr^{\frac{1}{3}} \quad . \tag{5}$$

For a typical gas composition from a gas turbine combustor the recovery factor is close to one, and therefore the adiabatic wall temperature is assumed to be equal to the local stagnation temperature (T_{ad}=T_g). The heat flux through the wall will then be

$$\frac{d\dot{Q}}{dA_w} = \alpha\,(T_g - T_w) \quad . \tag{6}$$

Assuming that the heat transfer coefficient for the expansion path is the same as for the stage average, one may write

$$\alpha = \frac{St\, c_{P,g}\, \dot{M}_g}{A_g} \tag{7}$$

where the Stanton number is defined as

$$St = \frac{\alpha}{\rho\, u\, c_{P,g}} \tag{8}$$

The heat transfer through the blade wall can be expressed by the following equation:

$$d\dot{Q} = c_{P,c}\, \varepsilon\, (T_w - T_c)\, d\dot{M}_c \tag{9}$$

where $d\dot{M}_c$ is the cooling air flow rate supplied at T_c, and ε is the heat exchanger effectiveness for an internally cooled blade, i.e.,

$$\varepsilon = \frac{T_{c,out} - T_c}{T_w - T_c} \quad . \tag{10}$$

$T_{c,out}$ in Eq. (10) is the temperature of the cooling air entering the gas path. Further, the work extracted through the walls is given by

$$d\dot{W} = -\,\dot{M}_g\, c_{P,g}\, dT_g \quad . \tag{11}$$

dT_g is obtained from

$$\frac{dT_g}{T_g} = \left(\frac{p + dp}{p}\right)^{\frac{R\eta_{P,t}}{c_P}} - 1 \tag{12}$$

The efficiency ($\eta_{P,t}$) in Eq. (12) is the uncooled turbine polytropic efficiency. At this point it is possible to derive an expression which relates the required cooling air mass flow to the work extraction for an element of the expansion path. By combining Eqs. (1), (2), (6), (7), (9) and (11) the following relation is found,

$$\frac{d(\dot{M}_c\, c_{P,c})}{\dot{M}_g\, c_{P,g}} = \sigma\, \frac{dT_g}{T_a}\, \frac{(T_g - T_w)}{(T_w - T_c)} \tag{13}$$

where σ is

$$\sigma = \frac{St\, \frac{A_{w,stage}}{A_g}}{C\,(\kappa - 1)\, Ma^2\, \varepsilon} \tag{14}$$

and where the Mach number is

$$Ma = \frac{u}{\sqrt{T_a\, \kappa R}} \quad . \tag{15}$$

For a cascade, the ratio of wall area to gas flow path inlet area is

$$\frac{A_w}{A_g} = \frac{l}{SP} + \frac{2B}{L} \tag{16}$$

where l = airfoil perimeter, SP = blade circumferential spacing, B = axial breadth, and L = blade height. Typical values of A_w/A_g is around 4 for a row and 8 for a stage. Stanton numbers based on the cascade inlet area, according to Elmasri (1986b), are typically about 0.005. The Mach number, Ma, based on the pitchline velocity is typically in the range of 0.6 to 0.85 for current gas turbines. With κ=1.4 and C=1.2, Eq. (14) gives the estimate of $\sigma \cdot \varepsilon$ to be 0.12-0.26. Heat exchanger effectiveness, ε, depends on the cooling air delivery temperature and the blade internal heat transfer area, the heat transfer coefficient and the temperature difference. The value of ε by Louis et al. (1983) is assumed to be in the range of 0.3 to 0.5, where the lower value is typical for film cooled blades with multiple compressor extractions, and the upper value is typical for internally cooled blades with cooling air taken from the compressor discharge. The parameter σ characterizes the relative heat to work loadings on the machine surfaces, and can be regarded as a descriptor of the level of technology for a cooled machine.

Mixing the cooling air with the hot gas results in an irreversible loss of stagnation pressure. Consider the flow in a duct between two sections an infinitesimal distance apart (Fig. 3). In this element of duct length, cooling air is injected with an angle ϕ into the gas stream at the flow rate $d\dot{M}_c$. To find the loss of stagnation pressure, a momentum balance is applied (Shapiro, 1953), and arranged in the following manner:

$$\frac{dp}{p} = -\frac{d\dot{M}_c}{\dot{M}_g}\kappa M^2 Y \qquad (17)$$

The factor Y, when being different from unity, permits other values for the mixing pressure loss than that of perpendicular mixing.

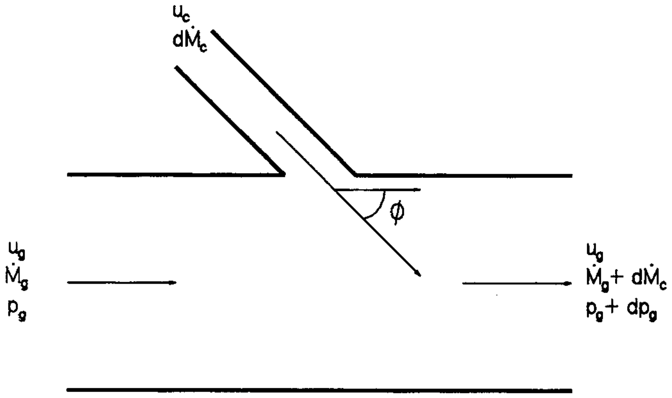

Fig. 3. Element of duct with gas injection

The model of one element of the expansion path is then described by Eqs. (11), (12), (13) and (17). The expansion path is divided into a large number of elements, and for each element those four equations are employed. Fig. 4 shows the computational model of one element in an enthalpy/entropy-diagram. The element is split into the following parts: adiabatic expansion (work extraction), mixing of hot gas and cooling air at constant pressure (loss of stagnation temperature), and mixing of hot gas and cooling air at constant enthalpy (loss of stagnation pressure).

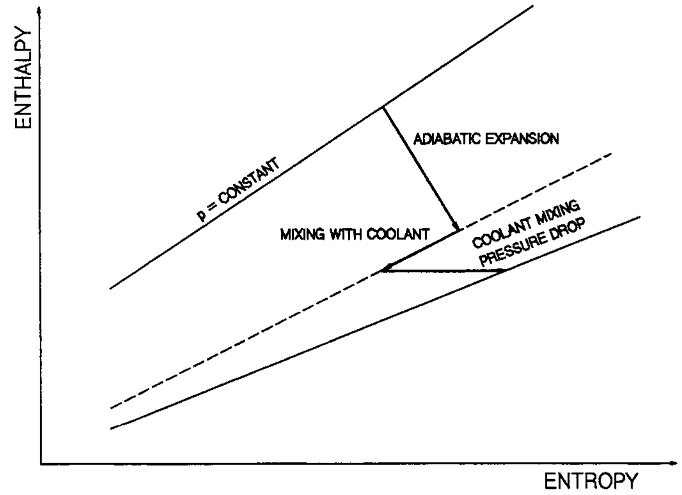

Fig. 4. Computational model for an element of the expansion path

The compression path is computed with a large number of stages in order to 1) take into account variable specific heat capacity due to temperature dependency and changes in air composition if an evaporative intercooler is used, 2) and have multiple cooling air bleeds. Considering stage 'i' of the compression path the following equations are used. The work equation is

$$\dot{W}_i = \dot{M}_i c_{P,i} \Delta T_i \quad . \qquad (18)$$

The ΔT_i is computed from the stage inlet temperature, T_i, and the stage inlet and outlet pressures, p_i and p_{i+1}, with the following equation:

$$\frac{\Delta T_i}{T_i} = \left(\frac{p_{i+1}}{p_i}\right)^{\frac{R}{\eta_{P,co} c_P}} - 1 \quad . \qquad (19)$$

The stage flow rate is the compressor inlet mass flow, $\dot{M}_{co,inlet}$, minus the cooling air mass flow, $\dot{M}_c$, extracted from stages at lower pressures than the present stage, plus the mass added in the evaporative intercooler, $\dot{M}_{IC}$, if the stage is at a pressure above the intercooler pressure. That is,

$$\dot{M}_i = \dot{M}_{co,inlet} - \sum_{n=1}^{i-1} \dot{M}_{c,n} + \dot{M}_{IC} \quad . \qquad (20)$$

The cooling air flow rate is determined by Eq. (13). For a given cooled turbine stage, the corresponding compressor stage at which cooling air is bled is selected in such a way that the pressure of the cooling air equals the pressure of the cooled turbine stage multiplied by a factor slightly larger than unity. If this required bleed pressure is larger than the compressor discharge pressure, cooling air is supplied from the compressor discharge. In the present work emphasis was placed on including precise calculation of thermophysical properties (Kunz, 1982; Olikara and Borman, 1975; Valland and Pedersen, 1988). For the combined cycle calculations, the computational model for the steam cycle was given by Bolland (1991).

GAS TURBINE CHARACTERIZATION

In order to provide useful numeric results from the cycle analysis, efforts were made to employ the technology level of existing gas turbines. A selection of gas turbines was carried out. These machines were categorized in four classes: large industrial (A), medium industrial (B), aeroderivative machines (C) and small industrial (D). For each class of machines an "average" gas turbine was defined and used in the cycle analysis. The technology parameters, described in the previous section, for each gas turbine served as a basis when defining these average gas turbines. The data for the gas turbines were mainly based on the GTPRO (1992) database, and included air flow, compressor discharge temperature, power output, fuel flow, turbine exit temperature and an estimate of the cooling air flow rate downstream of the first stator row of the turbine. To quantify representative technology parameters for each gas turbine, the parameters in the model (combustor exit temperature T_{CE}, turbine uncooled polytropic efficiency η_P and the level of technology descriptor σ) were found such that the sum of relative deviations (δ) between the known machine data and those calculated by the model was minimized. A simplification was made by setting the cooling air mixing loss factor (Y) to unity because its significance with respect to gas turbine performance is much smaller compared to the other three parameters mentioned above. The minimization problem is then

$$\delta = \min\{ \left| \frac{\dot{W}_{GT,model} - \dot{W}_{GT,known}}{\dot{W}_{GT,known}} \right| + \left| \frac{\dot{M}_{F,model} - \dot{M}_{F,known}}{\dot{M}_{F,known}} \right| + \left| \frac{T_{TE,model} - T_{TE,known}}{T_{TE,known}} \right| \} \quad . \qquad (21)$$

The results of the gas turbine characterization are given in Table 1. Only a few of the gas turbines listed in Table 1 are able to take steam injection flow rates substantially exceeding those used for NO_X control. Very few gas turbines are intended for or may easily be redesigned for compressor intercooling or recuperation (Thomas and Higson, 1985; Staudt et al., 1989). In the present work it is assumed that some modifications can be made to the gas turbines. These are: 1) allow a larger nozzle throat area for the turbine 2) changes in the cooling air flow paths 3) air can be taken out of the machine for recuperation and be put back.

Table 1. Results from the characterization of gas turbines

Class	Types	[1]$\dot{W}_{GT}$	[1]η_{GT}	[1]T_{TE}	[1]PRC	σ	T_{CE}	$\eta_{P,t}$	$\eta_{P,co}$	[2]$\dot{M}_c$
		MW	%	°C	-	-	°C	%	%	%
A	Siemens V84.3 ABB GT13E GE 9171E GE 7221FA	141	35.0	550	14.2	0.41	1234	87.8	90.7	14.9
B	GE 6541B Siemens V64.3 ABB Type 8 West. 251 B12A	48.4	32.5	532	14.8	0.49	1203	86.2	89.7	15.6
C	LM 2500PE LM 5000PD LM 6000PA TPM FT8	30.2	37.6	471	23.4	0.40	1269	88.3	90.0	19.6
D	Ruston Typhon Ruston Tornado Solar Taurus Dresser R. DC990	4.6	28.8	487	12.2	0.63	1019	85.5	86.8	5.9

[1] Inlet/outlet pressure drop = 10 mbar, ambient temperature 15 °C, relative humidity 60%, auxiliary power requirement is here not taken into account. Efficiency based on fuel lower heating value and generator terminal output.

[2] The total amount of cooling air downstream the combustor exit as percentage of the compressor inlet flow rate

CYCLE DESCRIPTION

Four types of cycles were studied; simple cycle, steam injected cycle, DRIASI (dual-recuperated intercooled aftercooled steam injected) cycle and combined cycle. The steam injected and the DRIASI cycles are shown in Fig. 5 and Fig. 6, respectively. When comparing the simple cycle and the steam injected cycle, the latter offers a higher efficiency mainly due to reduced cycle energy rejection temperature, and higher specific work due to the small compression work for liquid water compared to air. The increase in efficiency and specific power for a steam injected cycle over a simple cycle is thermodynamically limited by a pinch-point restriction of the exhaust gas heat recovery process. Another method of increasing gas turbine cycle efficiency is to preheat the combustion air by a heat exchange from the turbine exhaust (recuperator). The increase in efficiency for a recuperated cycle over a simple cycle is thermodynamically limited by the heat exchanger effectiveness of the recuperator and the compressor discharge temperature. The idea behind the DRIASI cycle (Elmasri, 1988a,b; Bolland, 1990) is to combine steam injection, recuperation and water injection into an effective gas turbine exhaust heat recovery scheme. Water injection into the compressor or/and after the compressor helps bring down the stack temperature, thus reducing the temperature of the cycle heat rejection. On the other hand, water and steam injection increases the loss of latent heat from the stack. Evaporative intercooling of the compressor with water increases the heat capacity of the turbine coolant and reduces the required coolant flow rate. By combining water injection with recuperation, a reduction of the temperature of heat addition to the process is avoided. The pinch-point limitation of the steam injected cycle is cancelled by locating a portion of the recuperator (LTR) below the pinch-point temperature of the heat recovery process. The water injection, together with the low temperature recuperator, allows steam generation at temperatures below the pinch-point because the water is vaporized at a partial pressure well below that of the gas turbine combustor. A visualization of the heat recovery process for the DRIASI cycle is given in Fig. 7, where the numbers correspond to those in Fig. 6. The combined cycle takes advantage of the high temperature of cycle heat addition of the Brayton cycle and the low temperature of cycle heat rejection of the Rankine cycle. By using a dual or triple pressure steam Rankine cycle the heat transfer losses between the Brayton and Rankine cycles are rather small as well as the stack loss (Bolland, 1991). The cooling system of combined cycles for class A gas turbines is typically based on a direct water cooled condenser, which allows the condenser pressure to be very low. For smaller combined cycles there is a tendency towards using cooling towers or air cooled condensers. The condenser pressure for such cooling systems is higher. In the present work

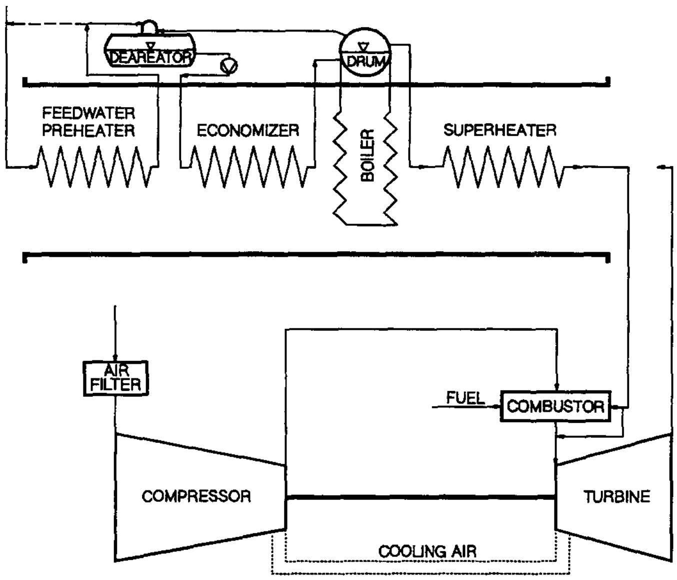

Fig. 5. Steam injected gas turbine cycle

it is assumed that combined cycles with gas turbines of classes B, C and D are using cooling towers with an increased condenser pressure, compared to combined cycles with class A gas turbines. The steam cycle level of technology, here meaning if it is a triple pressure, dual pressure or single pressure cycle, is in current design practice very dependent upon the power output of the steam cycle. This design practice is also reflected in the assumptions made for the present work, and is described in the Appendix.

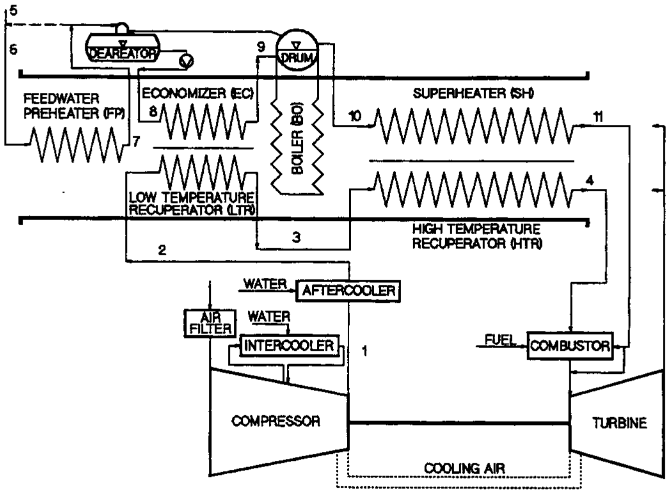

Fig. 6. Dual recuperated intercooled aftercooled steam injected cycle (DRIASI)

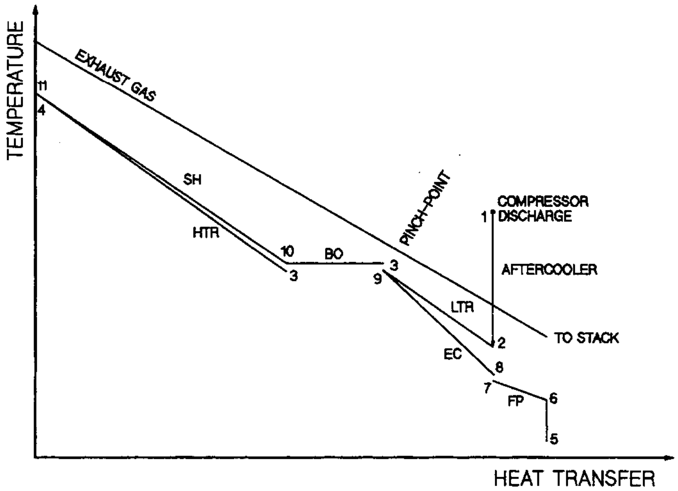

Fig. 7. Heat recovery diagram for the dual recuperated intercooled aftercooled steam injected cycle

RESULTS AND DISCUSSION

Parametric studies are presented over a broad range of cycles pressure ratios (*PRC*) and turbine inlet temperatures (T_{CE}), and the results for the four types of cycles are given in separate diagrams (Fig. 8 to Fig. 11) for each class of gas turbines. In these diagrams the cycle efficiency (η_{cycle}) and specific work (ω) are given as functions of the pressure ratio and turbine inlet temperature. Different definitions of turbine inlet temperature exist, but in the present work the combustor exit (or first nozzle inlet) temperature is used. A list of computational assumptions is given in the Appendix. The definitions of cycle efficiency and specific work used in the present work are given in Eqs. (22) and (23).

$$\eta_{cycle} = \frac{\dot{W}_{GT}\,\eta_{M+G,GT} + \dot{W}_{ST}\,\eta_{M+G,ST} - \dot{W}_{AUX}}{\dot{M}_F\ LHV} \quad (22)$$

$$\omega = \frac{\dot{W}_{GT}\,\eta_{M+G,GT} + \dot{W}_{ST}\,\eta_{M+G,ST} - \dot{W}_{AUX}}{\dot{M}_{co,inlet}} \quad (23)$$

The thick lines in the diagrams are for constant pressure ratio and varying turbine inlet temperature (1000-1400 °C) while the thin lines are for constant turbine inlet temperature and varying pressure ratio (5 or 10 to 20 or 30).

The results of the computational model for the simple cycle are in agreement with known simple cycle gas turbines. Current design practice is obviously a compromise between having high efficiency and high specific power. The results of the steam injected cycles reveal both higher efficiency and specific power compared to known systems like the aeroderivative steam injected cycles. The reason for this is that in the present work the rate of steam injection is limited by the pinch-point constraint, while for existing systems the limitation may typically be gas generator speed (Hånde, 1992). The results for combined cycles are in accord with current design practice.

The steam injected cycle is superior to the simple cycle in both efficiency and specific work. The difference in efficiency is typically in the range of 8-10%-points when considering typical pressure ratios and turbine inlet temperatures for the different machine classes. A characteristic difference between the simple cycle and the steam injected cycle is that the increase in specific power per increase in turbine inlet temperature is much larger for the steam injected cycle. Another characteristic difference is that for a given pressure ratio the efficiency peaks at a higher turbine inlet temperature for the steam injected cycle.

The DRIASI cycle is typically 3-4%-points more efficient compared to the steam injected cycle, while specific work is slightly lower for the DRIASI cycle. The efficiency for the DRIASI cycle peaks over a broad range of pressure ratios and turbine inlet temperatures. This is a remarkable characteristic of the DRIASI cycle which means that high efficiency is obtainable for moderate pressure ratios and rather low turbine inlet temperatures. For the class D gas turbines it is impractical to have pressure ratios exceeding 12-15 and the cooling technology is not developed to the same level as for larger gas turbines. Therefore, the class D gas turbines are especially suited to this type of cycle arrangement, when high efficiency is wanted. Even for gas turbines with very advanced cooling technology, high efficiencies may be obtainable at rather moderate turbine inlet temperatures.

The large combined cycles are superior to the other systems with respect to efficiency. Typical values for the efficiency of the combined cycle with class A gas turbines are in the range of 52-55%, while the DRIASI and the steam injected cycles are at 49 and 45-46, respectively. For the combined cycles with gas turbines from classes B, C and D, the differences in efficiency compared to the DRIASI cycle are much smaller. The combined cycle efficiency is reduced as a function of gas turbine technology level and steam cycle power output. For the gas turbine classes B, C and D, the condenser pressure is higher because the use of a cooling tower is assumed. The steam turbine is less efficient because it is smaller, and the steam cycle level of technology is typically reduced as power output is decreased. For the smallest machines (class D), the DRIASI cycle provides comparable or superior efficiencies to the combined cycle.

Medium and small combined cycles are, in most cases, designed for cogeneration of power and heat, and extraction and/or backpressure systems offers the possibility of having a high degree (up to 90%) of fuel utilization as well as a certain flexibility between the heat and power generation. The DRIASI cycle has some of the same advantageous capabilities of the combined cycle in cogeneration applications (Elmasri, 1988a,b; Bolland, 1990).

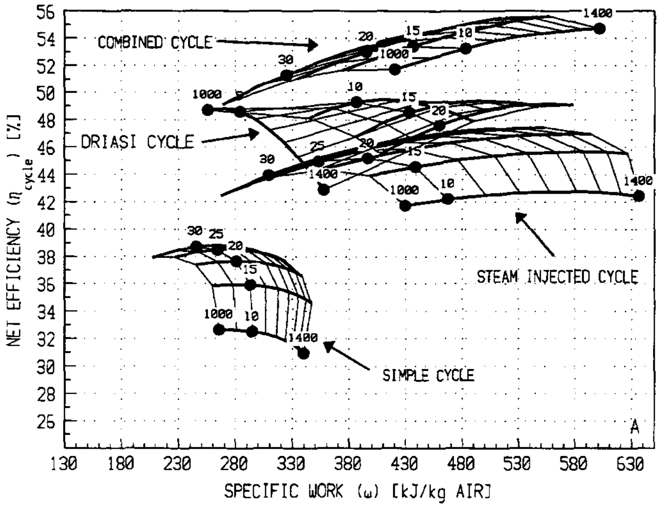

Fig. 8. Computational results for machine class A (large industrial)

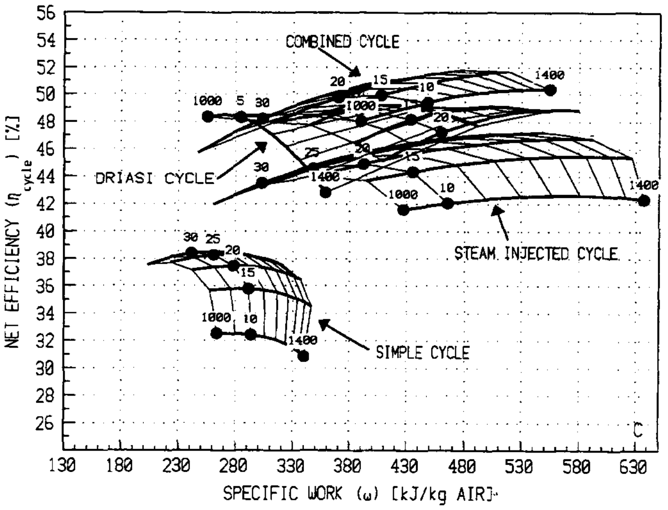

Fig. 10. Computational results for machine class C (aeroderivatives)

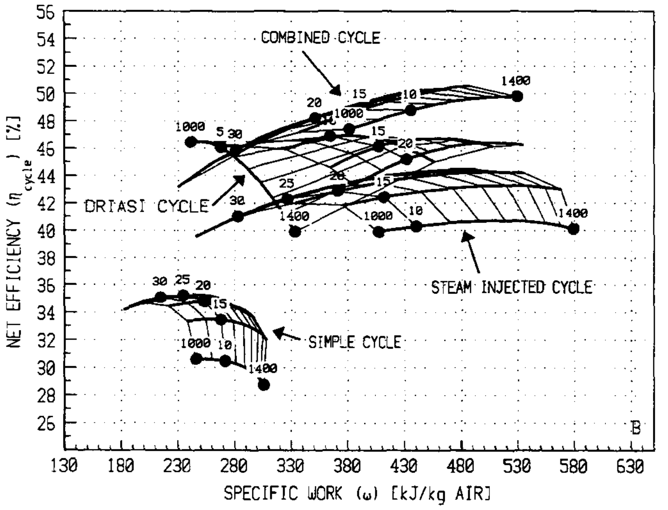

Fig. 9. Computational results for machine class B (medium industrial)

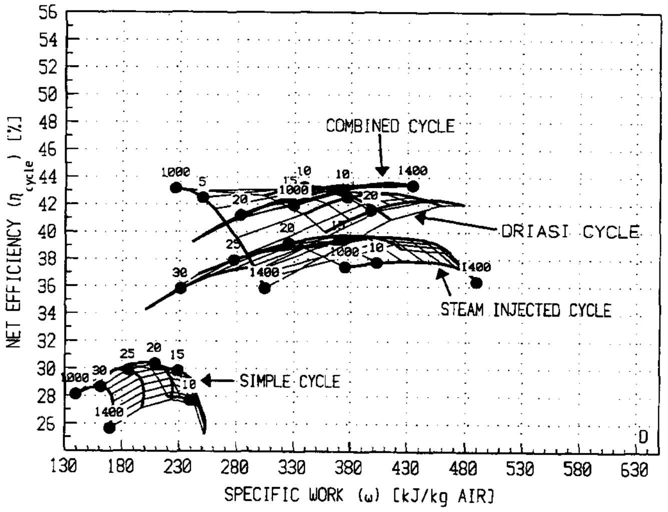

Fig. 11. Computational results for machine class D (small industrial)

CONCLUSIONS

A detailed, general model that represents the gas turbine with turbine cooling has been developed. The model is intended for use in cycle analysis applications. Suitable choice of a few technology description parameters enables the model to accurately represent the performance of actual gas turbine engines of different technology classes.

A characterization of 16 existing gas turbines has been carried out with the developed gas turbine model, resulting in descriptions of "average" gas turbine technology levels of four different classes.

A parametric study (pressure ratio and turbine inlet temperature) has been carried out for four types of power cycles including four classes of gas turbine technology level. For the small systems, the proposed development of the gas turbine cycle, the DRIASI cycle, are found to provide comparable or superior efficiencies to combined cycles, and superior to steam injected cycles. For the medium systems, combined cycles provide the highest efficiencies but can be challenged by the DRIASI cycle. For the largest systems, the combined cycle was found to be superior to all of the alternative gas turbine-based cycles considered in this study.

The DRIASI type of cycle with its special characteristics, points toward another direction in gas turbine design development. The current trend in gas turbine technology development is towards higher pressure ratios and turbine inlet temperatures for medium and small machines, and towards mainly higher turbine inlet temperatures for large machines. The DRIASI cycle opens up for improved gas turbine performance without any further increases in the turbine inlet temperature or pressure ratio. Instead, emphasis is on developing heat exchanger and water evaporation units. It has been shown that high efficiencies are attainable with redesigned gas turbines with a low or moderate level of technology.

ACKNOWLEDGEMENTS

The discussions held with Dr. Maher Elmasri were invaluable during the course of this work.

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APPENDIX

Assumptions used in the present work are given below. If four different numbers are given, these refer to gas turbine classes A-D, respectively.

Ambient: 15 °C, 1.013 bar, 60% RH
Cooling water temperature:15 °C
Fuel: CH_4, *LHV*=50056 kJ/kg, *T*=25 °C

Gas turbine cycle

Combustion efficiency (%):99.7, 99.7, 99.5, 99.0
Total-pressure losses (%):
air filter:1.0, combustor:4.0, diffuser:1.0, pipe to recuperator:0.6, pipe from recuperator:0.6, recuperator hot and cold side:0.6, intercooler: 2.5, aftercooler:2.5, superheater & boiler & economizer & feedwater preheater cold side (each):5.0, superheater & boiler & economizer & feedwater preheater hot side (each):0.6, steam injection steam pipe:10.0,
Heat losses:
pipe to recuperator:1.0 K, pipe from recuperator:1.0 K, recuperator (HTR) & superheater (each):0.2 %, recuperator (LTR) & boiler & economizer & feedwater preheater (each):0.1 %, steam injection steam pipe:0.5 K, from turbine to HRSG inlet:1.0 K.
Minimum recuperator temperature difference:30 K
Exhaust gas-superheated steam temperature difference:30 K
Minimum steam generator pinch-point:10 K
Economizer approach temperature:5 K
Mechanical and generator efficiency (%):
(average values including gear box when used) (%):98.2, 96.9, 98.1, 94.9
Auxiliary power requirement: 200 kW (per 100 kg/s of compressor air)
Maximum turbine blade metal temperature:850 °C
Minimum pressure difference between compressor bleed point and the cooled turbine stage:$p_{bleed} = p_{turbine\ stage} * 1.25$
Intercooler & aftercooler: the amount of water injected is such that the exit air is 15 K above the dew-point temperature. Water injection temperature is 15 °C.
Intercooler location in compressor: It is optimized in each cycle calculation.

Steam cycle

Class A: Triple pressure reheat steam cycle, p_{steam}:HP=120 RH=30 IP=16 LP=3 bar
Class B: Dual pressure steam cycle, p_{steam}:HP=80 LP=5 bar
Class C: Dual pressure steam cycle, p_{steam}:HP=60 LP=5 bar
Class D: Single pressure steam cycle, p_{steam}:HP=45 bar

<u>HRSG</u>

Pinch-point:10 K
Exhaust gas-superheated steam temperature difference:
20-30 K (30 for the hot end)
Economizer approach temperature:5 K
Pressure drop steam pipes to turbine (%):HP=5, RH=6, IP=7, LP=10
Heat loss steam pipes to turbine:1 K
Pressure drop superheater & reheater & boiler & economizer & feedwater preheater (cold side):5 %
HRSG pressure drop exhaust gas:40 mbar
Deaerator pressure:1.2 bar
Maximum steam temperature:550 °C

<u>Steam turbine</u>

Pressure drop throttle/control valves:2 %
Pressure drop reheat return pipe:3 %
Isentropic efficiencies (%) (before correction for moisture):
Class A: HP=92 RH=92 IP=92 LP=89
Class B: HP=90 LP=87
Class C: HP=88 LP=85
Class D: HP=83
LP section leaving loss:30 kJ/kg
Wilson line steam quality:0.975 [kg/kg], below this steam quality the steam turbine efficiency is corrected for moisture (Bolland, 1990;1991)
Auxiliary power fraction:0.25 %
Mechanical and generator efficiency (%):98.2, 97.0, 96.0, 95.0
Condenser pressure (bar):0.04, 0.07, 0.07, 0.07
Pumps: mechanical efficiency:92 %, isentropic efficiency:80 %