

Evaluation of Advanced Combined Cycle Power Plants

Dipl.-Ing. C. Kail, Siemens AG, Erlangen

Summary

The report will analyze and evaluate the most recent and significant trends in combined cycle gas turbine (CCGT) power plants configurations. The various enhancements will be compared with the "simple" gas turbine.

The first trend, a gas turbine with reheat can not convert its better efficiency and higher output into a lower cost of electrical power. The additional investments required as well as increased maintenance costs will neutralize all the thermodynamic performance advantages.

The second concept of cooling the turbine blades with steam puts very stringent requirements on the blade materials, the steam quality and the steam cooling system design. Closed loop steam cooling of turbine blades offers cost advantages only if all its technical problems will be solved and the potential risks associated with the process can be eliminated through long demonstration programs in the field.

The third configuration, a gas turbine with a closed loop combustion chamber cooling system appears to be less problematic than the previous, steam cooled turbine blades. In comparison with an open combustion chamber cooling system, this solution is more attractive due to better thermal performance and lower emissions. Either air or steam can be used as the cooling fluid.

Introduction

The goal of advanced power plant processes is electric power generation at high efficiency, low power generation costs and with a minimal impact on the environment (emissions). In this regard, CCGT power plants are currently superior to other types of power plants. The present Siemens gas turbines are the result of continuous further development of proven technology. Their use in CCGT power plants results in very high efficiencies (> 58 %). The use of film cooling in the turbine blades and the development of the annular combustion chamber and premix burner are the most recent successfully completed measures to increase efficiency and decrease emissions.

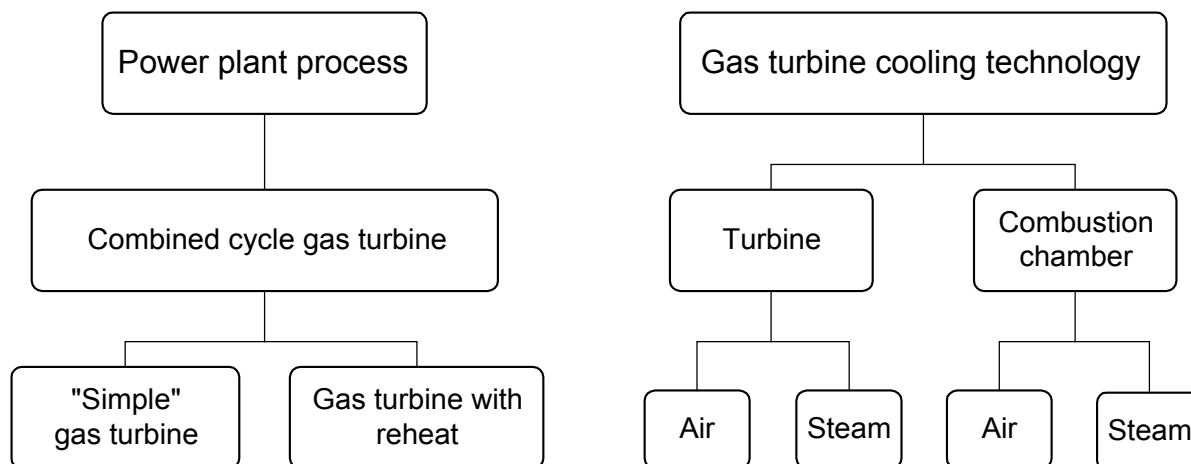


Fig. 1: Primary development directions for CCGT power plants

The trend towards further efficiency increases and lower emissions continues to require significant development efforts. Fig. 1 shows the current main development trends in CCGT power plants. The CCGT process can either be operated with a "simple" gas turbine, as was common to date, or with a gas turbine with reheat. The right side of the figure shows the various ways of cooling the combustion chamber and the turbine. There is a choice between air and steam as a coolant and between open and closed systems. A CCGT power plant with a "simple" gas turbine similar to the Siemens V94.3A type serves as a comparison power plant for the evaluation of the various concepts. The steam process features three pressure levels with reheat.

The results shown below contain elements of the energy and exergy analysis. They are more detailed than the results of a previous paper [1], which contains the results of extensive parameter variations of the gas turbine and steam turbine processes of CCGT power plants.

Methods for Thermodynamic Evaluation and Optimization

The goal of the thermodynamic optimization of power plant processes is the determination of the most favorable combination of the process parameters with regard to efficiency and output. The investigation and optimization of the processes can be performed by an energy or exergy analysis.

Energy is a conserved quantity. When energy is converted from one form to another, a part of the ability to produce work is irreversibly lost. It is not possible to quantify these losses with the energy analysis. The exergy analysis allows the determination of these losses and thus the quality of the energy.

In the energy analysis, energy flows of various forms and qualities are used to determine the characteristic parameters of the process. The energetic efficiency η of the process is calculated from the ratio of the useful output P to the energy input Q_F of the fuel. Like the efficiency, the heat losses rejected to the surroundings Q_{out} are

determined from energy balances.

$$\eta = \frac{P}{\dot{Q}_F} = \frac{\dot{Q}_F - \dot{Q}_{out}}{\dot{Q}_F} \quad (1)$$

For the reversible process, the efficiency can be based on the Carnot efficiency η_C by using the mean temperature $T_{m,out}$ of the heat output and the mean temperature $T_{m,in}$ of the heat input.

$$\eta_C = 1 - \frac{T_{m,out}}{T_{m,in}} = \eta + \frac{T_{m,out} \cdot \sum \dot{S}_{irr}}{\dot{Q}_F} \quad (2)$$

The Carnot efficiency and the real process efficiency are coupled by the entropy production S_{irr} of the process. If energy losses occur in the transfer of the heat energy in the fuel to the process, a corresponding efficiency (e.g. combustion chamber efficiency) must be included on the right side of Equations 1 and 2.

For most components of the process, no energetic efficiency in the sense of useful output over input can be calculated, as no energy losses occur i. e. the quality of these components can only be evaluated by comparing the real irreversible state change with the ideal reversible one. This comparison yields isentropic efficiencies. These can be determined only for those components in which no heat transfer, no mixing of materials and no chemical reactions occur (e.g. for the compressor). Heat exchangers or combustion chambers therefore cannot be qualitatively evaluated by this method.

The second method for the investigation and optimization of power plant processes is the exergy analysis [2, 3]. In this method, the exergy flows occurring in the process are used to calculate exergetic efficiencies and exergy losses. Exergy is a measure of the capacity of a fluid to perform work if it is reversibly brought into equilibrium with the surroundings. This can occur with interaction with the surroundings. Exergy has the character of a potential in a given environment. There is no law of conservation for exergy. Each irreversible state change represents the destruction of exergy.

Neglecting kinetic and potential contributions, exergy E is determined from the enthalpy H , the entropy S and the molar amounts n_i of the mixture as well as from the chemical potentials μ_i of the individual materials at ambient temperature, ambient pressure and the composition of the mixture and the surrounding [4].

$$E = H - H_0 - T_0 \cdot (S - S_0) + \sum_i (\mu_{i,M}^0 - \mu_{i,0}^0) \cdot n_i \quad (3)$$

The dependence of the exergy on the surrounding conditions is no disadvantage for power plant applications. The maximum cycle efficiency, the Carnot

efficiency, is also restricted by the temperature of the surroundings as the minimum temperature for heat removal.

The exergy analysis enables a much more detailed determination of the losses than the energy analysis. A loss and an efficiency can be allocated to each component. In contrast to the energy analysis, losses and efficiencies can also be determined, for example, for heat exchangers and combustion processes.

Analogous to the energetic efficiency, the exergetic efficiency ε of the process is calculated from the ratio of useful output P to the fuel exergy input E_F . It can also be determined from the losses S_{irr} and E_{out} .

$$\varepsilon = \frac{P}{\dot{E}_F} = 1 - \frac{T_0 \cdot \sum \dot{S}_{irr}}{\dot{E}_F} - \frac{\sum \dot{E}_{out}}{\dot{E}_F} \quad (4)$$

The advantage of the exergy analysis over the energy analysis lies in the detailed information on the losses and efficiencies of all of the components in the process. This extensive information enables more targeted action in the parameter variations to optimize the process, thus possibly also reducing the required effort.

Figures 2 and 3 show the energy and exergy flows in a CCGT process. The representation of the energy flow accounts for the heat of condensation of the gas turbine exhaust gas. For this reason, the energy supplied to the combustion chamber must be based on the higher heating value HHV of the fuel. The higher heating value is calculated from the heat of reaction of the substances participating in the combustion considering the condensation of water, and is larger than the fuel exergy determined by the Gibbs function of the reaction. In contrast to the heat of reaction, the Gibbs function of the reaction is independent of the size of the condensing water fraction, as the Gibbs function of the water is constant in the wet steam range. It is therefore not possible to distinguish between a higher and a lower value in the fuel exergy.

The energy input of the fuel supplied to the combustion chamber in the gas turbine is 100.8 %. It consists of 100 % chemically bound energy and 0.8 % thermal energy (fuel preheating). 34.7 % of the fuel input is transformed into mechanical output, and 65.9 % is passed on to the heat recovery steam generator. The heat recovery steam generator supplies the steam turbine with 47.9 % of the fuel energy supplied to the gas turbine, from which the steam turbine produces a mechanical output of 18.7 % (based on fuel input).

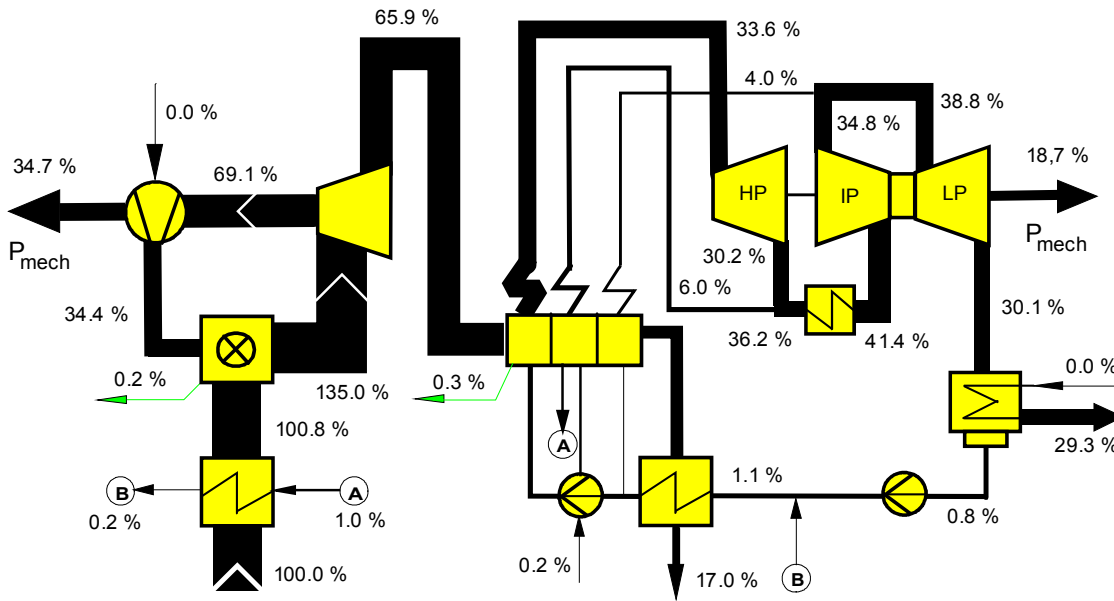


Fig. 2: Energy flow in CCGT process with "simple" gas turbine (based on HHV)

The efficiency of the overall process is obtained from the sum of the mechanical outputs and is 53.4 % (based on HHV, without generator losses). The heat rejected to the surrounding consists of 17.0 % exhaust gas heat and 29.3 % heat of condensation. Overall, this amounts to 46.3 % of the fuel energy supplied to the process.

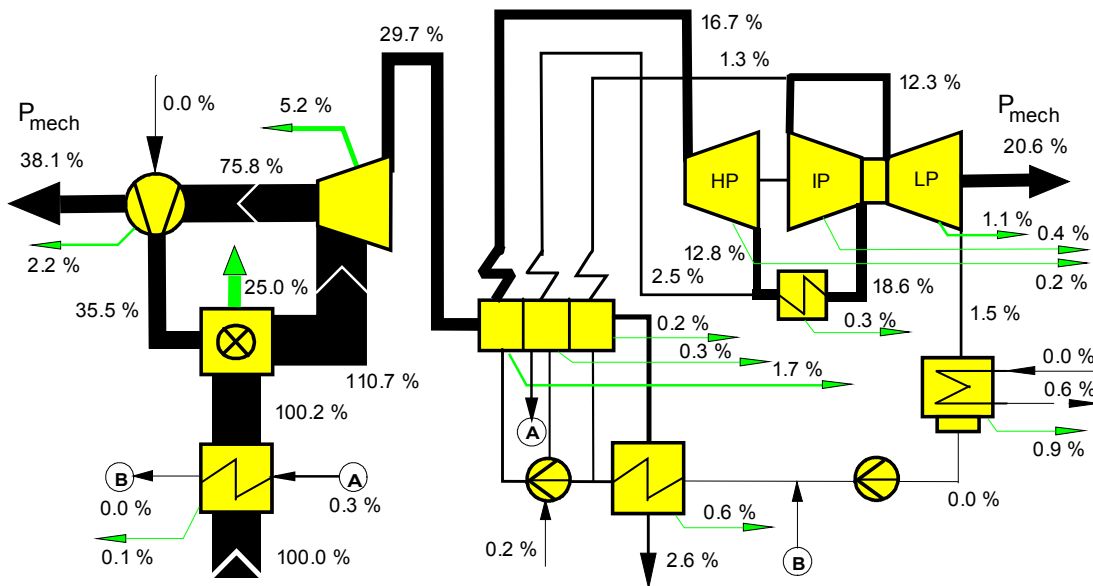


Fig. 3: Exergy flow in CCGT process with "simple" gas turbine

The exergy flow decreases much more rapidly than the energy flow, as an exergy loss occurs in each component. The largest source of losses is the combustion chamber, in which 25 % of the supplied exergy is lost. 29.7 % of the supplied exergy is passed on to the heat recovery steam generator by the gas

turbine, in contrast to 65.9 % of the energy. The steam turbine receives 23.7 % of the exergy in the fuel from the boiler, in comparison with 47.9 % of the energy. Only 3.2 % of the supplied exergy, in contrast to 46.3 % of the energy, is rejected to the surroundings as loss flows.

The main differences between the energy and the exergy balance are the losses. Energy is lost in the exhaust gas and in the condenser, while exergy is mainly lost by the irreversibilities in the components of the system.

Reheat of the Gas Turbine Working Fluid

Reheat of the turbine working fluid increases the mean temperature of the heat supply and hence the efficiency potential of the process. A gas turbine with reheat has a compressor, two combustion chambers and two turbines. The air exiting the compressor is first heated in the high-pressure combustion chamber and is expanded in the high-pressure turbine. Downstream of the HP turbine, the flue gases are reheated in the low-pressure combustion chamber before subsequent expansion in the LP turbine. The pressure ratio is approximately double that of a gas turbine without reheat. The pressure ratios for the results shown here are 16 and 28.

Fig. 4 shows the "simple" gas turbine process and the gas turbine process with reheat in a T-S diagram. The total temperature is plotted on the ordinate and consists of the static temperature and a dynamic fraction. The dimensionless entropy change is plotted on the abscissa. The entropy change of the gas turbine process is based exclusively on the production of entropy. The production of entropy S_{irr} in this figure was made dimensionless by multiplication with the surrounding temperature T_0 and division by the fuel exergy E_F supplied to the process in each case. It therefore corresponds to the exergy loss of the process (see Eq. 4). This representation enables a direct comparison of the losses occurring in the processes. The fuel energy supplied and the work performed by both processes cannot be compared because the entropy change was normalized based on the corresponding fuel exergy supplied.

For the calculation of the gas turbine processes, a highly detailed model was developed to account for the cooling air requirement of the combustion chamber and of each individual blading row of the turbine. Each turbine stage is assigned an aerodynamic efficiency which is corrected by a factor to account for the mixing losses of the cooling air. The compressor is calculated block-by-block corresponding to the number of cooling air extraction points.

The amount of nitrogen oxide (NO_x) produced in the combustion process increases with increasing pressure. The combustion temperatures of the gas turbine with reheat must therefore be lower in order to achieve the same emissions (corrected to 15 % Oxygen).

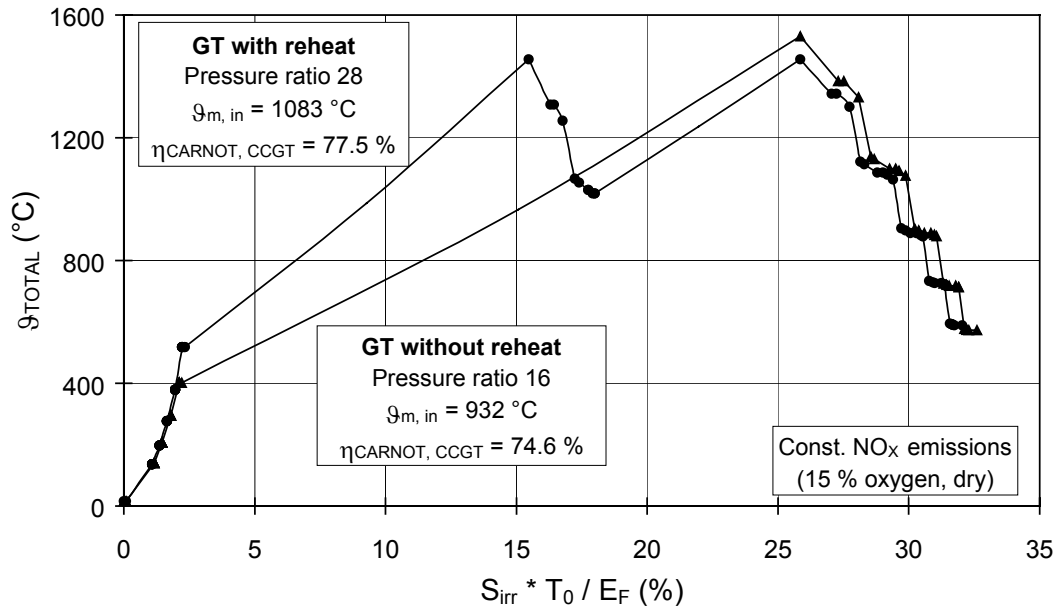


Fig. 4: "Simple" gas turbine and gas turbine with reheat in T-S diagram

The mean temperature of the heat supplied to the gas turbine with reheat is 151 K higher than for the gas turbine without reheat (Fig. 4). This is due primarily to the reheat, but also to the higher compression ratio. The mean temperature of the heat removal is the same in both cases. It is 32.5 °C under the assumption of a condenser pressure of 0.04 bar and the resulting exhaust gas temperature of 87 °C. The Carnot efficiency increases by 2.9 percentage points due to the higher temperature of the heat supply in the process with reheat.

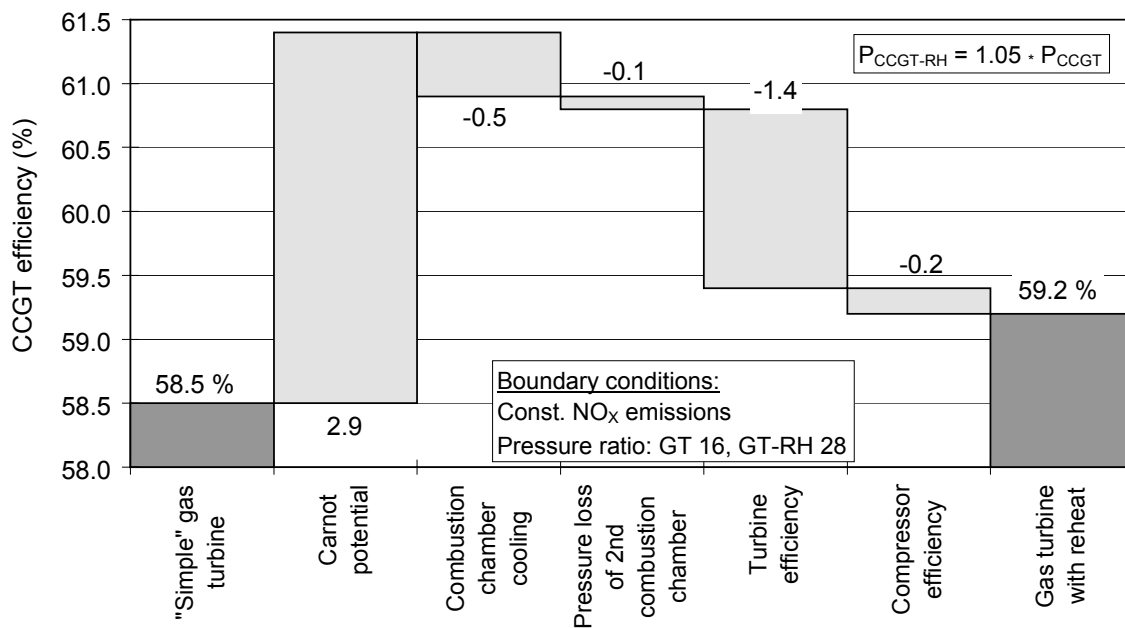


Fig. 5: CCGT efficiency of "simple" gas turbine and gas turbine with reheat (based on LHV)

Fig. 5 shows the CCGT efficiency of the "simple" gas turbine and the gas turbine with reheat and the corresponding changes in the losses of the CCGT process. If both processes had the same quality, the efficiency difference would correspond to the difference of the Carnot efficiencies. However, the reheat produces additional losses and therefore does not achieve the improvement potential determined by the Carnot efficiency. The losses of the combustion chamber cooling are higher for the gas turbine with reheat, as two combustion chambers must be cooled, consequently the CCGT efficiency decreases by 0.5 percentage points. The additional pressure drop in the second combustion chamber decreases the efficiency by an additional 0.1 percentage points. The turbine in the system with reheat produces an additional loss of 1.4 percentage points. This loss results from the relatively poor expansion efficiency of the HP turbine. It has a low aerodynamic efficiency due to the small blade heights. This is further degraded by the high cooling air consumption which is very high due to the high pressure ratio and the resulting high cooling air temperatures. The diffuser located downstream of the high-pressure turbine, which reduces the flow velocity to a suitable level for the second combustion chamber, has a pressure drop of 1 to 2 % and is an additional source of losses in comparison with the gas turbine without reheat. The isentropic efficiency of the compressor is poorer than for the "simple" gas turbine due to the high pressure ratio. This reduces the CCGT efficiency by 0.2 percentage points. Overall, three quarters of the Carnot potential is lost through the higher losses of the process. The CCGT efficiency of the gas turbine with reheat is 0.7 percentage points higher than that for the "simple" gas turbine. The CCGT output is increased by 5 %.

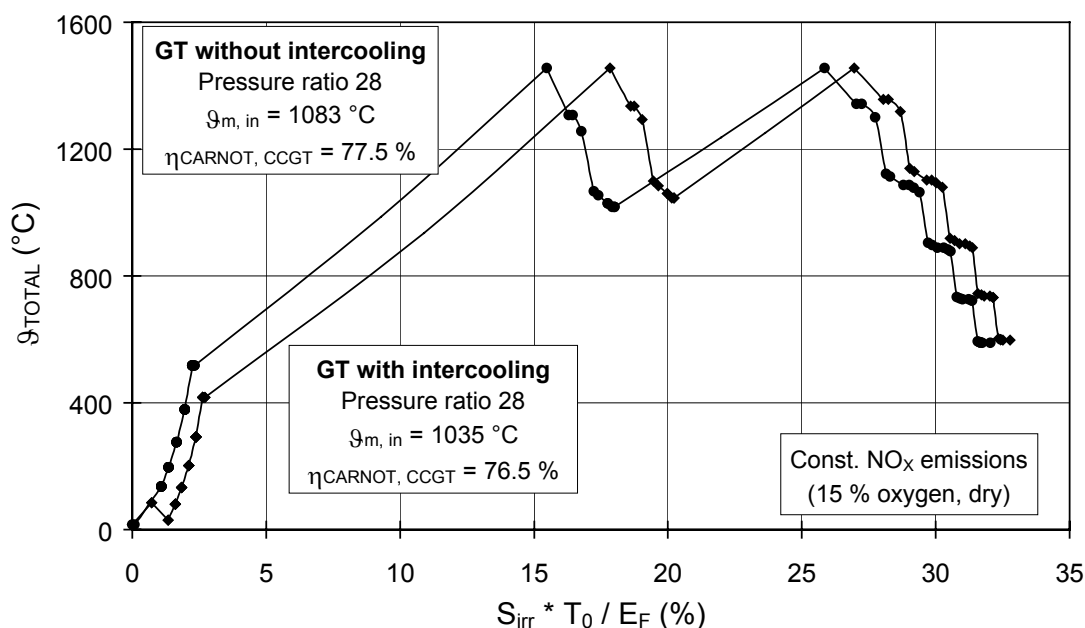


Fig. 6: Reheated gas turbine with and without intercooling in the T-S diagram

Intercooling of the compressor air decreases the mean temperature of the

compression and hence the required compressor power. This is particularly effective at high pressure ratios, and could therefore be a reasonable extension for the reheated gas turbine. Fig. 6 shows both processes on the T-S diagram. Intercooling reduces the compressor outlet temperature and thus also the mean temperature of the heat supply to the first combustion chamber. The heat removed by intercooling is no longer useful due to its low temperature, and must be rejected to the surroundings. This heat is rejected at a higher temperature than the waste heat from the boiler exhaust gas and the heat of condensation from the steam turbine, thus increasing the mean temperature of the heat removal. The Carnot-efficiency in the system with intercooling is 1 percentage point lower than that of the system without intercooling due to the 48 K decrease in the mean temperature of the heat supply and the 2.4 K increase in the mean temperature of the heat removal.

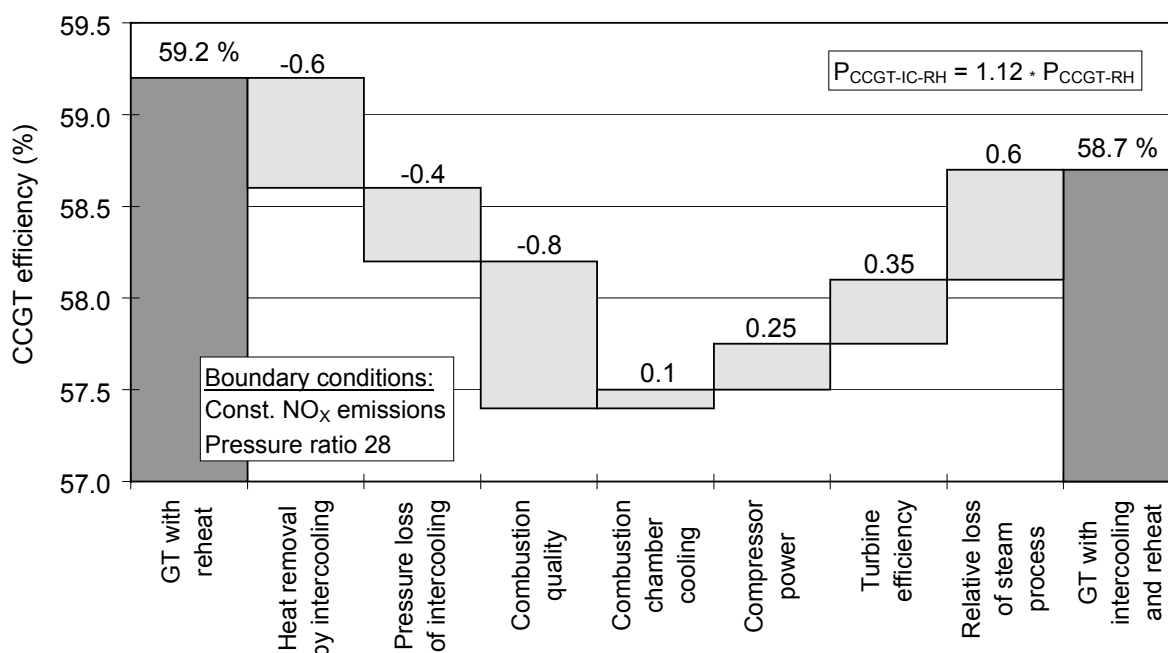


Fig. 7: CCGT efficiency of reheated gas turbine with and without intercooling (based on LHV)

Fig. 7 shows the CCGT efficiencies of the reheated gas turbine with and without intercooling and the changes in the losses. The changes in the process can be distinguished as efficiency decreasing and efficiency improving effects. Heat removal of the intercooling results in an additional loss of 0.6 percentage points. The pressure drop of the intercooling produces a deterioration of 0.4 percentage points. This is almost on the same order of magnitude as the losses due to heat removal. The poorer combustion quality (exergetic efficiency) of the system with intercooling is the result of the lower compressor outlet temperature and produces an additional loss of 0.8 percentage points. The lower cooling air temperatures obtained by intercooling enable the amount of cooling air to be reduced. This leads to improvements in the combustion chamber cooling (0.1 percentage points) and in the turbine efficiency

(0.35 percentage points). These improvements primarily affect the high pressure stage, as it is affected by the high cooling air temperatures. The required compressor power decreases due to the lower mean temperature of the compression, thus raising the CCGT efficiency by 0.25 percentage points. The quality and output of the steam process are influenced only slightly by the intercooling. They remain constant as a first approximation. The ratio of the losses from the steam process to the supplied fuel input is the determining factor for the CCGT efficiency (Eq. 2 and Eq. 4) and decreases by 0.6 percentage points corresponding to the increased fuel supply. The sum of all changes in losses corresponds to the change in the CCGT efficiency and is -0.5 percentage points. This is only half as large as the change in the Carnot efficiency. The CCGT output increases by 12 % due to intercooling.

If the pressure ratio is increased beyond the value of 28 selected here, the positive effects of intercooling increase more rapidly than the negative ones, resulting in a continuous decrease in the CCGT efficiency loss, which eventually becomes a gain. This turning point lies between pressure ratios of 35 and 40. The gain in output due to intercooling increases continuously with increasing pressure ratio.

The power generation costs of the CCGT power plant with reheated gas turbine with or without intercooling offer no advantage over the "simple" gas turbine under the present cost situation. The efficiency and output advantages are not only compensated by the higher costs of the gas turbine, but also by the higher maintenance costs. Several additional compressor stages, a second high temperature turbine stage and a second combustion chamber with burners, larger casing dimensions and greater wall thicknesses as well as the need for higher quality materials increase the costs of the gas turbine with reheat. In addition, the comparatively large high temperature region results in higher maintenance costs over those of the "simple" gas turbine. The availability of the gas turbine with reheat is likely to be lower due to the higher complexity of the process. Additional technical risks such as premature ignition in the second combustion chamber under disturbed flow conditions diminish the attractiveness of the gas turbine with reheat.

Closed Steam Cooling of Turbine Blades

Output and efficiency of gas turbines generally increases with the turbine inlet temperature. Cooling of the turbine blades enables an increase in the turbine inlet temperature beyond the allowable material temperatures. Most of the gas turbines currently in operation have air-cooled blades. The cooling air first cools the blades convectively from within and then exits through openings. It can then be used to develop a protective film over the blade surface. The admixture of the cooling air to the flue gas decreases the flue gas temperature and disturbs the flow field, deteriorating the expansion efficiency of the stage. The pressure loss of the cooling air from compressor extraction to the mixing point in the turbine requires additional compressor power.

Steam is inherently a better cooling medium than air due to its higher heat capacity and its lower viscosity. Closed steam cooling of the turbine blades avoids the losses connected with the mixing process of the cooling medium to the flue gases. In CCGT power plants, steam for cooling the gas turbine is available from the downstream steam process. The condition of the exhaust steam from the HP turbine is roughly 30 bar and 350 °C and is therefore suited for cooling in the gas turbine. Its conditions can be changed as required over a certain range by changing the main steam temperature and the ratio of reheat pressure to main steam pressure. The high pressure exhaust steam is reheated by cooling the blades and is subsequently supplied to the IP turbine. Parallel flow through the row of blading to be cooled is desirable to prevent the outlet pressures of the steam superheated in the boiler and the steam superheated in the gas turbine from differing too greatly.

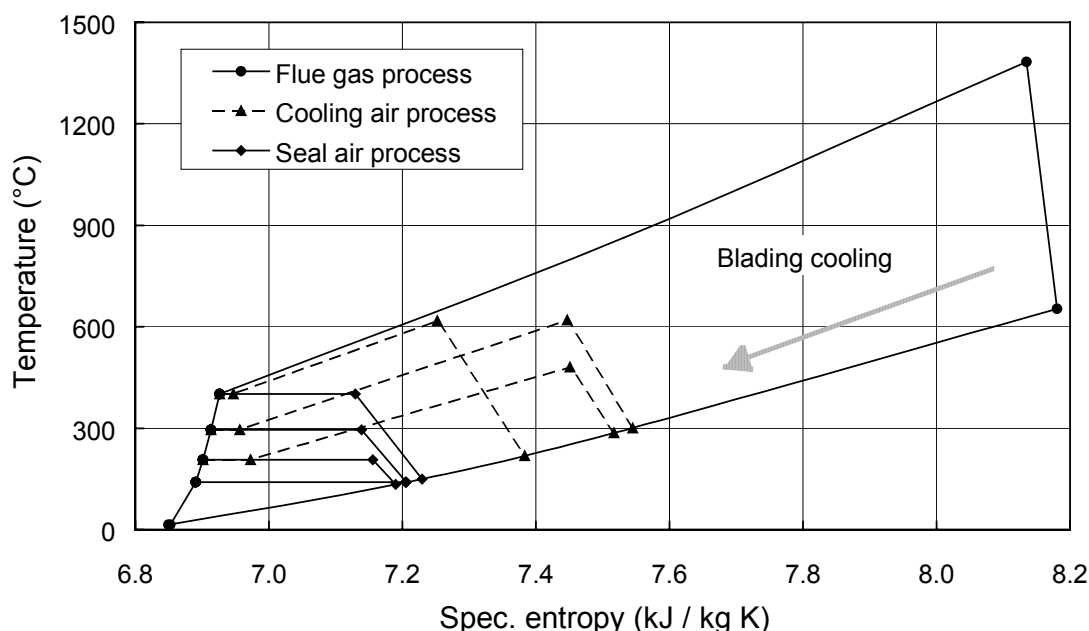


Fig. 8: Separation of the gas turbine process into flue gas, cooling air and seal air processes

As a first step towards estimating the efficiency potential of the gas turbine with steam cooling, the gas turbine process with air cooling is separated into a flue gas, cooling air and seal air process, each of which is calculated separately. This procedure enables determination of the quality of each process and estimation of the potential for improvement offered by closed cooling systems. Fig. 8 shows the flue gas, cooling air and seal air processes in the T-S diagram. The cooling air process was subdivided into three subprocesses corresponding to the number of stages to be cooled. The seal air process consists of four subprocesses, as all four turbine stages require seal air. The efficiency of the cooling air process only reaches roughly one-third of the efficiency of the flue gas process due to the low temperature of its heat supply and due to the pressure losses. The seal air process consists of compression, throttling and an expansion. It does not produce any output; rather it consumes

(dissipates) output.

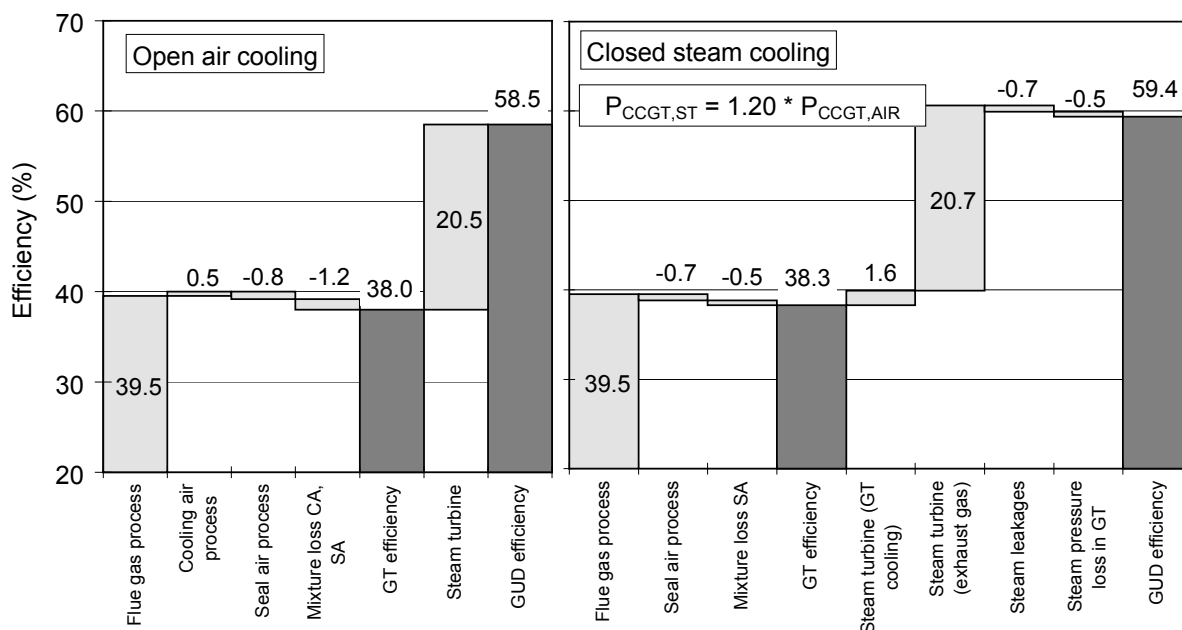


Fig. 9: CCGT efficiency of air-cooled and steam-cooled gas turbines (based on LHV)

Fig. 9 shows the results of the process calculation and the corresponding efficiencies of the CCGT processes. The power generated or supplied is plotted on the ordinate as a ratio to the fuel energy supplied. The results are shown for an open air cooling system in the left diagram, and for a closed steam cooling system in the right diagram. The efficiency of the flue gas process is 39.5 % in both cases. For the case of open air cooling, the cooling air process produces an additional output of 0.5 %. The seal air process does not produce any output. It consumes power and thus decreases the gas turbine efficiency by 0.8 percentage points. The decrease in the expansion efficiency in the real process due to the losses described above by admixture of cooling and seal air during expansion costs an additional 1.2 percentage points. After all of the losses are accounted for, the gas turbine efficiency is 38.0 %. The downstream steam process produces an output of 20.5 % resulting in a CCGT efficiency of 58.5 %.

The cooling air process is not required for the system with steam cooling. The seal air process produces somewhat lower losses, by 0.1 percentage points, as the flue gas mass flow is increased in the absence of cooling air extraction. The losses of 0.5 percentage points occurring in the real process by admixture are lower than for open air cooling, as here only seal air and no cooling air is mixed to the flue gas flow. The efficiency of the gas turbine with closed steam cooling is 38.3 %, 0.3 percentage points higher than for open air cooling. The output of the steam process is produced by the gas turbine cooling heat and by the exhaust gas heat from the gas turbine. These outputs are 1.6 % and 20.7 % (based on fuel energy). The closed steam cooling system is not completely leak-tight. Leaks must be anticipated, especially in

the rotor area. These leaks lead to efficiency penalties of 0.7 percentage points. The large pressure drop of the cooling steam in the gas turbine results in additional losses in the steam process, decreasing the efficiency by 0.5 percentage points. The resulting CCGT efficiency of the gas turbine with closed steam cooling is 59.4 %, and is thus 0.9 percentage points higher than for the gas turbine with open air cooling. The CCGT output increases by 20 %.

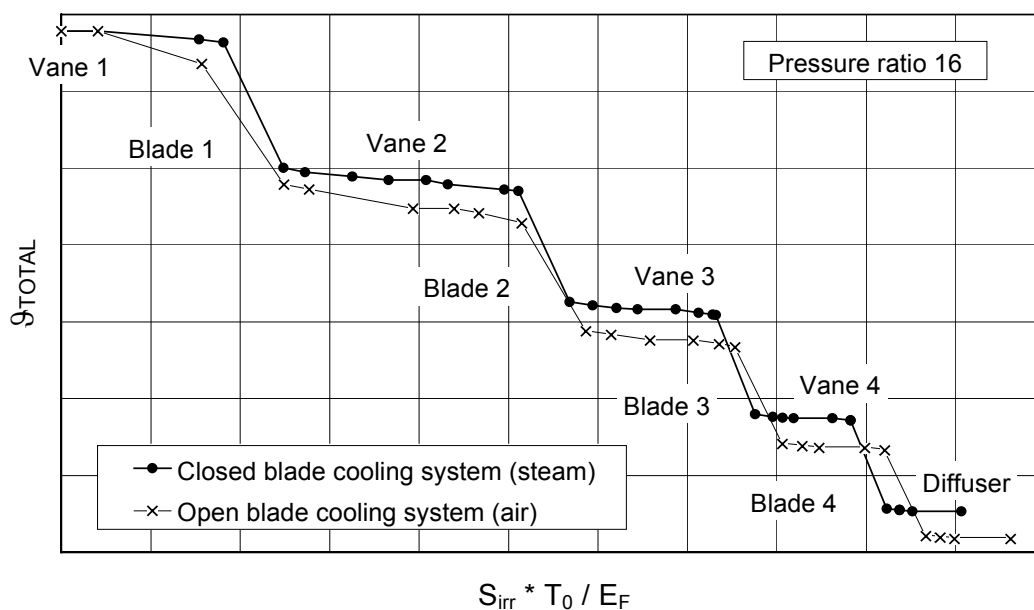


Fig. 10: Expansion curves for open air cooling and closed steam cooling of the turbine

Fig. 10 shows the expansion curves for the air-cooled and steam-cooled gas turbine in the T-S diagram. A comparison of the expansion curves clearly shows the differences in the rotating blade efficiencies and the temperature losses due to cooling. The differences are the greatest in the first stage. In the second and third stage they are smaller, and they disappear altogether in the fourth stage because it is not cooled.

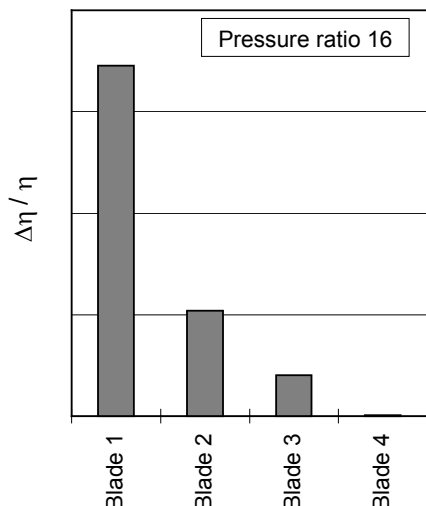


Fig. 11: Improved rotating blade efficiency due to steam cooling

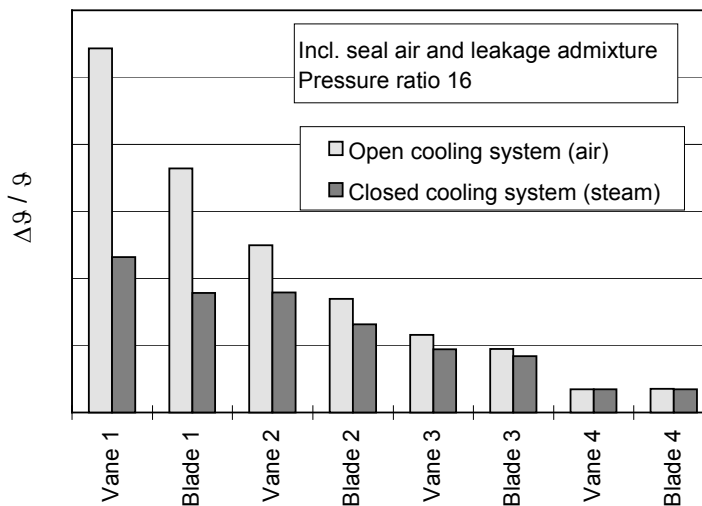


Fig. 12: Decrease of flue gas temperature for open air cooling and closed steam cooling

The relative changes in the rotating blade efficiencies for a change from open air cooling to closed steam cooling are shown in Fig. 11. Losses due to the admixture of cooling air (see above) are the largest in the first stage. In the second stage, the mixing loss is only one third, and in the third stage only one tenth that in the first stage. The isentropic efficiency of the overall expansion is approx. 2 percentage points higher for the case of closed steam cooling than for open air cooling.

Fig. 12 shows the relative temperature losses of the air cooling and the steam cooling for each individual row of blades. The ratio of the temperature losses for air cooling to those for steam cooling is roughly 2:1 in the first stage, roughly 4:3 in the second stage and nearly 1:1 in the third stage. The ISO turbine inlet temperature, which is a measure of the thermal load on the blading, is roughly 80 K higher than for air cooling due to the lower temperature losses associated with steam cooling. The exhaust gas temperature is roughly 40 K higher for the same pressure ratio.

The improvement potential of closed steam cooling is larger for the rotating blades than for the stationary vanes for several reasons. The admixture of the stationary vane cooling air with the flue gas deteriorates the expansion efficiency (rotating blade efficiency) less than that of the rotating blade cooling air because it does not have such a large influence on the flow field around the rotating blades. In addition, the pressure losses of the cooling air for the rotating blades are larger than for the stationary vanes due to the larger pressure drop along the rotating blades. The cooling heat to be removed is smaller for the rotating blades than for the stationary vanes due to the size difference. Film cooling is not as effective for the rotating blades as for the stationary vanes because the film is more likely to be disturbed by less favorable flow conditions and blading geometry. The transition from film cooling to convection cooling is thus also associated with a smaller loss in efficiency for the rotating blades than for the stationary vanes.

The CCGT power plant with a steam-cooled gas turbine has a higher efficiency and offers advantages in the specific plant costs over the air-cooled gas turbine due to the large increase in output. The maintenance costs of the gas turbine with steam cooling will be higher due to the anticipated shorter service life of the blades, thus reducing the advantage in the investment costs. High temperature gradients in the blades and the thermal barrier coatings and the resulting high thermal stresses are a reason for the reduced service life. Hot corrosion may also have to be anticipated. The cooling of the thin front and rear edges of the blades with a closed steam cooling system can only be implemented with difficulty. Extremely stringent purity requirements must be placed on the cooling steam to prevent deposits. The leakage-tightness of the closed cooling system is critical to the attainment of high efficiency. This means an enormous additional effort over open air cooling.

A CCGT power plant with steam-cooled turbine blades is only attractive from an economic perspective if the problems are solved satisfactorily and both the maintenance costs as well as the availability are on a similar order of magnitude as for the air-cooled gas turbine. The gas turbine with steam cooling is not suited for simple-cycle operation without a steam process.

Closed Combustion Chamber Cooling System

The cooling system of the combustion chamber can be designed either as an open or closed system. The open system consists of a combination of convection and film cooling. Coolant first flows along the outside of the part to be cooled, providing convection cooling before it reaches the hot side through openings and forms a protective film on the surface. Air is generally used as a coolant because the exhaust gas losses would increase sharply if steam were used. The cooling air is extracted from the compressor outlet mass flow and does not participate in combustion.

In a closed system, the coolant extracts heat by convection. Either air or steam can be used. In closed air cooling, the total compressor outlet flow is used for cooling the combustion chamber before it participates in the combustion process. This cooling system works like a recuperator. In closed steam cooling, a part of the exhaust steam from the HP steam turbine is used to cool the combustion chamber. Cooling steam is then mixed with the steam superheated in the boiler and is fed to the IP steam turbine.

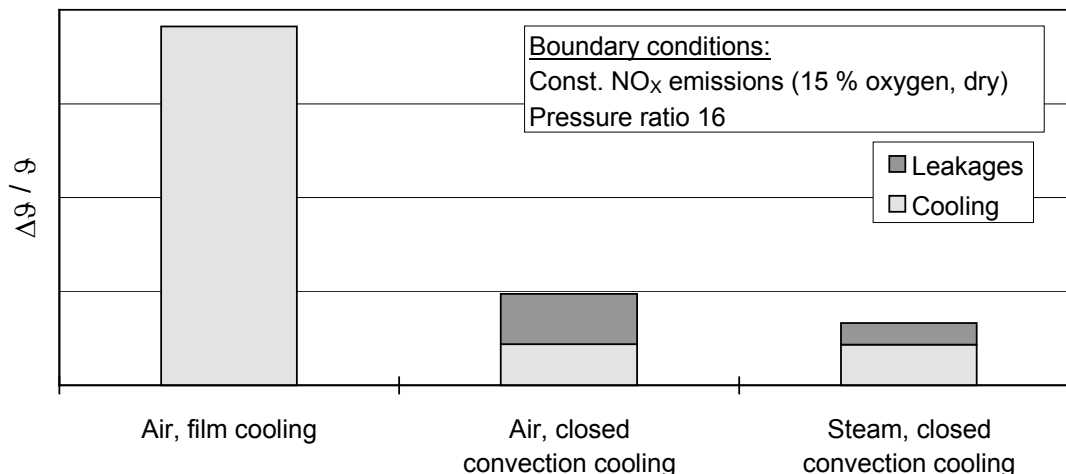


Fig. 13: Reduction of flue gas temperature in open and closed combustion chamber cooling systems

The temperature loss caused by cooling is greater in an open system than in a closed system due to the admixture of the coolant. Fig. 13 shows the temperature losses of the combustion chamber cooling for open air cooling, closed air cooling and closed steam cooling. Losses of the open system are four to five times as large as those of the closed system. Temperature losses of the two closed systems differ only negligibly. Losses for closed steam cooling are somewhat lower than that for closed air cooling because of the lower leakage rates. The steam leakages are smaller than the air leakages because the cooling steam mass flow is only 5 to 10 % of the cooling air mass flow.

The change from an open to a closed combustion chamber cooling system can be exploited to increase the efficiency or to reduce emissions. An efficiency increase is obtained for the same combustion temperatures and a reduction in emissions for the same combustion chamber outlet temperatures. Figs. 14 and 15 show the CCGT efficiencies of the open and closed combustion chamber cooling systems and the corresponding changes in the losses for constant combustion temperatures.

The results of the comparison of the open air cooling system with closed air cooling are shown in Fig. 14. The recuperative effect of the closed cooling system improves the combustion quality (exergetic efficiency) and thus increases the CCGT efficiency by 0.3 percentage points. Convection cooling produces lower losses than film cooling due to the absence of the admixture of the cold cooling air mass flow to the hot gas flow. This increases the efficiency by 1 percentage point. The leakages in the closed cooling system decrease the CCGT efficiency by 0.2 percentage points. The increased pressure drop in the combustion chamber decreases the efficiency by 0.1 percentage points. The higher turbine inlet temperature increases the cooling air requirement of the turbine and thus leads to a degradation of the expansion efficiency. The lower expansion efficiency decreases the total efficiency by 0.4 percentage points. The relative losses of the compressor decrease due to the higher

fuel supply rate (Eq. 2 and Eq. 4). This causes the CCGT efficiency to increase by 0.1 percentage points. The CCGT efficiency of the closed air cooling system is 59.2 % and is thus 0.7 percentage points higher than for the open air cooling system. The CCGT output increases by 8 %.

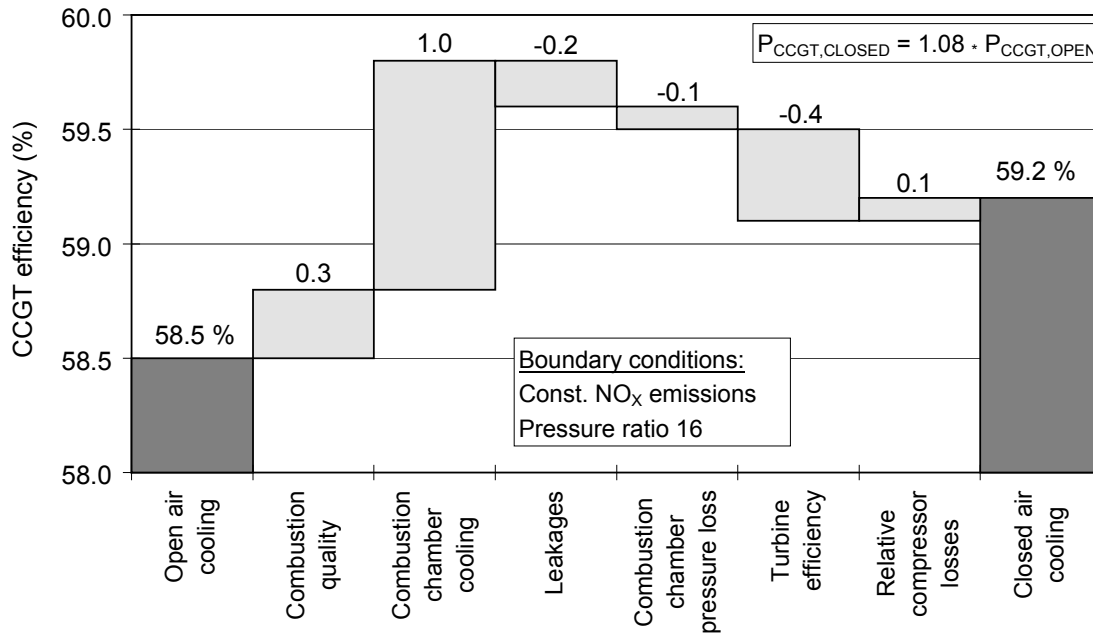


Fig. 14: CCGT efficiency for open and closed air cooling of the combustion chamber (based on LHV)

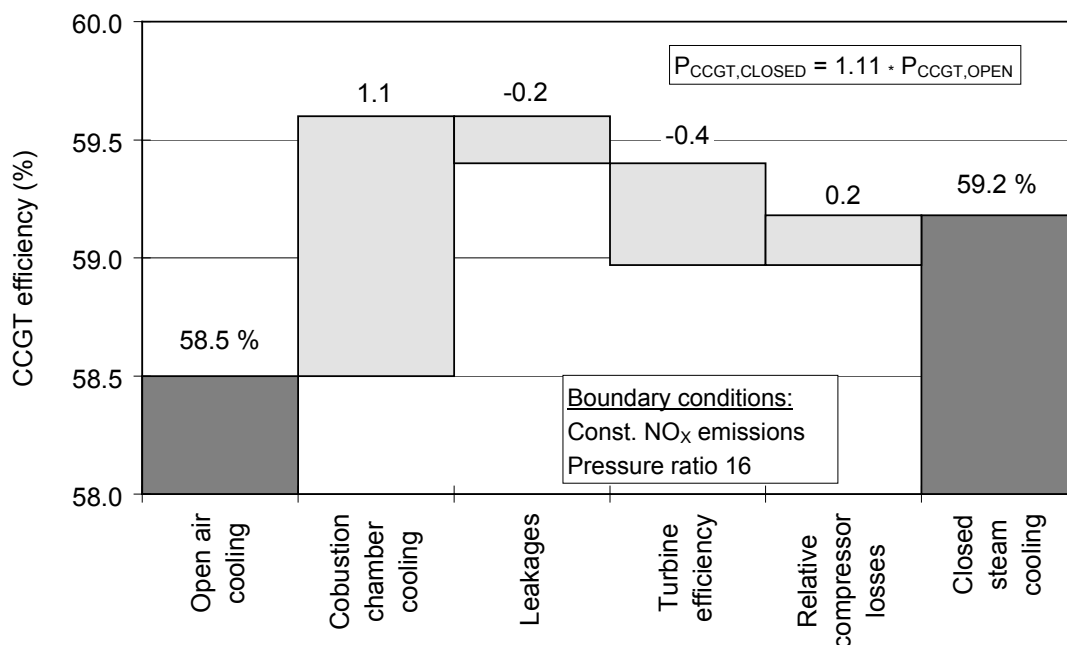


Fig. 15: CCGT efficiency for open air cooling and closed steam cooling of the combustion chamber (based on LHV)

Fig. 15 shows the results of the comparison of open air cooling with closed

steam cooling. As for closed air cooling, the quality of the combustion chamber cooling is considerably better for closed steam cooling than for open air cooling. The CCGT efficiency is increased by 1.1 percentage points. The steam leakages decrease the efficiency by 0.2 percentage points. The higher turbine inlet temperature increases the turbine losses due to the increased cooling air requirement, thus decreasing the efficiency by 0.4 percentage points. The decrease in relative compressor losses improves the process efficiency by 0.2 percentage points. After considering all changes in losses, the CCGT efficiency is 59.2 % and is hence 0.7 percentage points higher than for open air cooling, as is also the case for closed air cooling. The CCGT output is increased by 11 %, somewhat more than in the case of closed air cooling.

Under the assumption of constant nitrogen oxide emissions, the gas turbine with a closed combustion chamber cooling system has a higher CCGT efficiency and a higher CCGT output than the gas turbine with an open combustion chamber cooling system. If steam is used, the output increase is greater than if air is used. However, this output advantage of steam cooling is offset by the increased costs for additional piping and valves. The high thermal stresses of the closed cooling systems are more easily dealt with in combustion chambers than in turbine blades, as there are no additional mechanical loads. Leakage-tightness is more easily realized for the closed combustion chamber cooling system, as no rotating parts must be cooled. The steam purity requirements are not as high for combustion chamber cooling as for turbine blade cooling. The closed combustion chamber cooling system is therefore less problematic than the closed blade cooling system and is therefore also attractive from the standpoint of costs. This holds both for the use of air as well as for the use of steam.

Justified plant costs

The efficiencies, outputs and relative justified plant costs of the processes are summarized in Table 1. The justified plant costs C are calculated under the assumption, that all processes should have the same cost of electrical power. Differences in the availability of the processes are neglected, because the value of the availability is strongly dependent on the individual situation. In the case of the CCGT with the gas turbine with reheat (maintenance costs +10 %), the justified plant costs are 8 % higher as in the case of the CCGT with the "simple" gas turbine. However, the additional investments of the CCGT with the gas turbine with reheat are 15 % to 20 % higher than the plant costs of the CCGT with the "simple" gas turbine. This results in 2 % higher costs of electricity.

Table 1: Efficiencies, outputs and justified plant costs of the processes examined

| Gas turbine process | CCGT efficiency (gross %) | CCGT output (gross MW) | Relative justified plant costs (C/C_0) (%) |
|---|------------------------------|---------------------------|--|
| "Simple" gas turbine | 58.5 | 360 | 100 |
| Gas turbine with reheat | 59.2 | 379 | 108 |
| Gas turbine with intercooling and reheat | 58.7 | 425 | 123 |
| Closed steam cooling of the turbine blades | 59.4 | 432 | 131 |
| Closed air cooling of the combustion chamber | 59.2 | 388 | 114 |
| Closed steam cooling of the combustion chamber | 59.2 | 399 | 118 |

Intercooling decreases the specific plant costs because of an increased output. The difference between the justified and the real plant costs is smaller as in the case of the gas turbine with reheat and without intercooling. The costs of electricity are therefore only slightly higher (0.7 %) than the value of the CCGT with the "simple" gas turbine. The relative justified plant costs of the closed loop cooling systems are 131 % in the case of the steam cooled turbine, 118 % in the case of the steam cooled combustion chamber and 114 % for the air cooled combustion chamber.

Conclusions

In comparison with the "simple" gas turbine, the gas turbine with reheat cannot transform its efficiency and output advantages in the CCGT process into lower electric power generation costs, as the additional investments and higher maintenance costs overwhelm the thermodynamic advantages. Intercooling improves the cost situation of the gas turbine with reheat. However, the CCGT with the "simple" gas turbine achieves the lowest costs of electrical power and is therefore from an economic point of view the best process.

The concept of steam-cooled turbine blades places very stringent requirements on the blade materials, on the quality of the cooling steam and on the design of the closed cooling system. If the technical problems can be satisfactorily solved and the availability and the maintenance costs are similar to these of the air cooled gas turbine, closed steam cooling of the turbine blades offers cost advantages due to the higher efficiency and the significantly increased output over open air cooling.

The gas turbine with a closed combustion chamber cooling system is less problematic than the gas turbine with closed blade cooling system. It is also attractive from a cost consideration in comparison with the open combustion chamber cooling system due to its efficiency and output advantage. Air or steam can be used as a coolant.

Nomenclature

Roman / Greek

C: plant costs
 \dot{E} : exergy flow
 H: enthalpy
 n: molar amount
 P: power output
 \dot{Q} : heat flow
 \dot{S} : entropy flow
 T: temperature
 η : efficiency
 μ : chemical potential

Subscripts / Superscripts

C: Carnot
 F: fuel
 in: inlet
 irr: irreversible
 m: mean value
 M: mixture
 out: outlet
 0: ambient, reference case

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