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EFFECT OF THERMAL BARRIER COATINGS ON THE PERFORMANCE OF STEAM- AND WATER-COOLED GAS TURBINE - STEAM TURBINE COMBINED CYCLE SYSTEMS

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SUMMARY

The performance of steam and water-cooled gas turbine/steam turbine combined cycles using thermal barrier coatings on the gas turbine airfoils is presented and discussed. These results are also compared to the performance of combined cycles using air-cooled turbines with and without thermal barrier coatings. The thermal barrier coating assumed for this analysis is a plasma sprayed, two-layer coating consisting of a NiCrAlY bond coat and a yttria-stabilized zirconia overlayer. Due to the low conductivity of the overlayer, the heat transfer from the hot gases to the turbine vanes and blades is reduced, resulting in a reduction in the required cooling flow rate and, thus, improvements in system performance. Alternatively, the use of thermal barriers could allow the turbine inlet temperature to be increased to a point where the percentage coolant flow rate is the same as that required for the uncoated case while keeping the metal temperature constant, and this also improves the combined cycle performance. Both of these effects are investigated. The improvements in performance are calculated for overlayer thicknesses of .038 cm (.015 in) and .076 cm (.030 in) with a bond coat thickness of .010 cm (.004 in). These calculations are performed for a range of turbine inlet temperatures from 1205°C (2200°F) to over 1650°C (3000°F).

The maximum combined cycle efficiency improvement using thermal barrier coatings for the steam-cooled cases is 1.9 percentage points as the turbine inlet temperature is increased from 1205°C (2200°F) to 1370°C (2500°F), while maintaining the same airfoil metal temperatures. The maximum specific power increase is 32.4% when the turbine inlet temperature is increased from 1425°C (2600°F) to 1675°C (3050°F). For the water-cooled cases, the maximum efficiency increase is 2.2 percentage points at a turbine inlet temperature of 1683°C (3062°F), and the maximum specific power improvement is 36.6% when the turbine inlet temperature is increased from 1425°C (2600°F) to 1730°C (3150°F). The combined cycle performance gains are greater for the higher temperature cases since the cooling losses are much higher. The use of thermal barriers at these high temperatures yield greater reductions in these cooling losses and, thus, greater performance improvements. Large temperature differences of over 555°C (1000°F) across the thermal barriers at high temperatures indicate that large thermal stresses may present an obstacle to the implementation of thermal barrier coatings on high temperature gas turbines.

INTRODUCTION

The effect on the performance and cost of electricity (COE) of open cycle gas turbine systems when using thermal barrier coatings has previously been investigated (refs. 1, 2, 3). These studies indicate potential performance gains and reductions in fuel usage and COE for utility gas turbine systems through the use of thermal barrier coatings which allow either a reduction in turbine coolant flow rate for a constant turbine inlet temperature or an increase in turbine inlet temperature for a constant coolant flow rate. These analysis were done for state-of-the-art and near term air-cooled turbines. In this report, results from a performance analysis for more advanced steam and water-cooled open cycle gas turbine/steam turbine combined cycles, both with and without thermal barrier coatings, are compared and discussed. These results are also compared to the performance of an air-cooled gas turbine/steam turbine combined cycle, also with and without thermal barrier coatings.

A thermal barrier coating (TBC) is a thin layer of ceramic material applied to the hot section components of a gas turbine. This type of coating has the potential for protecting the gas turbine components from the erosive and corrosive environment of the hot combustion products and thus, could permit the use of less refined and cheaper fuels. Also, since the thermal conductivity of the TBC is substantially lower than the metal substrate on which it is applied, the TBC provides a thermal insulating barrier between the hot combustion products and the cooled turbine components. Thus, longer-lived components can result from lower metal substrate temperatures, or performance gains can be achieved by reductions in turbine coolant usage or by increasing turbine inlet temperatures to the point where the percentage coolant flow rate is the same as that required for the case without TBC while maintaining the same metal temperatures.

Thermal barrier coatings have been investigated at the Lewis Research Center for the protection of cooled rocket nozzles and gas turbine components (refs. 4, 5, 6). A plasma sprayed duplex TBC has been developed at Lewis and successfully tested (ref. 7). This two-layer TBC consists of a NiCrAlY bond coat and a yttria stabilized zirconia overlayer. Further tests are continuing, with emphasis on improving the coatings' hot corrosion and oxidation resistance.

The study described in this report was done as part of the Critical Research and Advanced Technology Support Project (CRT) which is being performed by NASA-Lewis Research Center for the Department of Energy (DOE) Division of Fossil Fuel Utilization (Interagency Agreement No. EF-77-A-01-2593). The purpose of CRT is to provide federal technical support to DOE to accelerate the development of utility size, advanced, open cycle gas turbine systems using coal-derived fuels. One of the CRT sub-tasks is a study of the performance potential of TBC with various coolants. This report presents the results of this study.

DESCRIPTION OF SYSTEMS STUDIED

Previous studies indicated greater performance gains for combined cycle systems when using TBC compared to the gains achieved with simple or recuperated gas turbines (refs. 1, 2, 3). Thus combined cycle systems were chosen for this study. Also, the combined cycle offers thermodynamic advantages when using water or steam as the gas turbine coolant by thermally integrating the gas turbine coolant streams with the steam bottom cycle. In the case of water-cooling, the heat lost from the gas turbine to the water coolant can be utilized for feedwater heating in the steam bottom cycle or by injecting the heated water coolant into flash tanks, and inducting the steam from the flash tanks into the steam turbine. Likewise for steam cooling the steam coolant can be extracted from the steam turbine after having done useful work.

A plasma-sprayed duplex TBC was assumed for the analysis. This TBC consists of a yttria-stabilized zirconia thermal barrier with a NiCrAlY bond coat. The zirconia has a relatively low thermal conductivity and, hence, insulates the cooled turbine vanes and blades from the high temperature combustion products. The thermal expansion coefficient of the zirconia matches that of gas turbine alloys better than other TBCs of this type, resulting in good adherence at high temperatures. Also, the NiCrAlY bond coat should provide some oxidation and hot corrosion resistance.

The combined cycle cases analyzed are shown in table 1. The turbine inlet temperatures considered for each type of coolant are indicated. Air-cooled turbine cases, both with and without TBC, were included for comparison to the steam and water-cooled cases. As shown, three situations were considered for each TBC case. In the first two, the surface metal substrate and turbine inlet temperature were assumed the same as the case without TBC, and the turbine coolant flow rate was reduced. Coating thicknesses of .038 cm (.015 in) and .076 cm (.030 in) were assumed. In the third situation, the metal temperature was kept the same as the case without TBC, and the turbine inlet temperature was increased to the point where the coolant flow rate was the same as the case without TBC at the original turbine inlet temperature. A coating thickness of 0.38 cm (.015 in) was used in this case. The bond coat thickness was assumed to be .010 cm (.004 in) for all cases.

Assumptions

A summary of some of the assumptions used in the analysis is shown in table 2. The compressor pressure ratios were chosen for the particular turbine inlet temperatures shown based on previous studies (refs. 8, 9, 10). The combustor pattern factor is defined as the peak local temperature exiting the combustor minus the bulk mean combustor exit temperature (turbine inlet temperature), all divided by the combustor temperature rise. The same steam turbine throttle conditions were used for all calculations to avoid an additional variable in the comparison between cases. Also to facilitate comparisons, the steam cycle was configured in such a way that the stack gas temperature was 149°C (300°F) for all cases. (This will be further discussed in later sections). The surface metal substrate temperatures assumed for the air and steam-cooled cases are typical for these types of cooling media. The metal temperature of the water-cooled cases is somewhat lower because of the higher heat transfer rates obtained with water coolant. The TBC conductivity shown represents an average value over the temperature range of the coatings for all cases. The conductivity varies little over the temperature ranges studied. Since the bond coat thickness is relatively small compared to the zirconia layer, and its conductivity is relatively large, the bond coat was assumed to have the same temperature as the metal substrate surface.

The gas turbine performance was determined by analyzing the turbine "row by row". The cooling requirements for each stator vane and rotor blade row, as well as the heat loss from the hot gas path, were calculated. The work of each stage was calculated, including the effect of discharged coolant and heat losses on the gas temperature between rows.

The air cooling requirements for each turbine row were calculated based on cooling correlations developed at the Lewis Research Center for advanced impingement-convection methods (ref. 11). For steam cooling, these correlations were modified to account for the heat transfer properties of steam. For water cooling, data developed in Phase 1 of the High Temperature Turbine Technology program (HTTT) were obtained from the General Electric Company and used to calculate the heat lost to turbine components and the water cooling requirements (ref. 12).

For the air and steam-cooled cases, the turbine coolant is discharged into the hot gas path and is assumed to be fully mixed with the hot combustion products before entering the next turbine row. With water cooling, the stator vane coolant is not discharged into the hot gas path. For the airfoil cooling design of the rotor blades, 70% of the blade coolant water is assumed to evaporate to steam and is lost to the gas path; the remaining water is recovered.

In addition to the blading coolant flow, some air from the compressor is also used as blockage flow in the turbine wheel spaces between blade rows to prevent hot combustion products from flowing down into these spaces. For the air and steam-cooled cases, approximately 0.5% of the compressor inlet airflow rate is assumed per stage for blockage flow. For the water-cooled cases, a total blockage flow of about 2% of the compressor inlet flow rate is used, based on General Electric data (ref. 12). For the air and steam-cooled cases, the blockage flow is assumed to be extracted from the compressor discharge and mixed with the turbine exhaust gases. For the water-cooled cases, some of the blockage flow is assumed to be extracted from intermediate compressor stages, as well as from the compressor discharge. This blockage flow is injected into the turbine gas path between stages, as well as at the turbine exhaust. The difference in procedure for blockage flow accounting has little effect on the combined cycle performance.

For the air and steam-cooled gas turbine combined cycle systems, the number of turbine stages was calculated based on the enthalpy drop across the turbine and assumed values of the work-speed parameter (.8) and mean blade speed (1200 ft/sec). For all of the water-cooled cases, a three stage turbine was assumed, based on the General Electric water-cooled design. This design represents a higher stage loading. The number of stages for the air and steam-cooled gas turbines is thus greater than three, and represents a more conservative design approach.

For the cases incorporating TBC, a one diminsional heat transfer analysis was made across the two layer TBC. Based on the required substrate temperature, inlet gas temperature to that row, and the thickness and conductivity of the TBC, the temperature difference across the TBC and heat transfer rate were calculated. No temperature profiles along the blade span or chord were taken into account in this analysis. Althougn these profiles would be needed to actually design a TBC airfoil, it is judged that a one-dimensional analysis of the coating is adequate to predict the performance of a gas turbine/steam turbine combined cycle system.

The gas turbine exhaust temperature and flow rate are used to determine the heat input to the steam cycle. The steam turbine power output is then determined and, thus, the combined cycle performance is calculated. Combined cycle efficiency is expressed in terms of the higher heating value of an average kerosene fuel (HHV-46.114 MJ/Kg (19810 BTU/lb)). Auxiliary power requirements were not included in the performance results. Auxiliary power requirements for clean-fuel fired combined cycles are typically small compared to the total power produced. (For a combined cycle fired with a liquid fuel, G.E estimated plant auxiliary requirements of 1.5% of the total gross power output - ref. 13).

Combined Cycle System With Air-Cooled Turbines

A schematic diagram of the combined cycle system with air-cooled gas turbines is shown in figure 1. Part of the compressor discharge air is used for cooling the turbine vanes and blades, and as blockage flow. Steam is generated in a heat recovery steam generator (HRSG) by recovering heat from the exhaust flow of the gas turbine. The HRSG consists of a superheater, boiler, economizer, and low pressure economizer-boiler. The low pressure economizer-boiler raises steam for use in the deaerator. Thus, steam extractions from the main steam turbine are not needed for this purpose. To reduce the stack temperature to 149°C (300°F) in some cases, additional steam is raised in the low pressure economizer-boiler, and this steam is then inducted into the steam turbine.

Combined Cycle with Steam-Cooled Turbines

A schematic diagram of the combined cycle system with steam-cooled turbines is presented in figure 2. A small percentage of the compressor discharge flow is used for blockage in the turbine. Steam is extracted from the steam turbine for cooling the gas turbine components. The pressures at which this steam is extracted are 1.38 MPa (200 psia) and 1.72 MPa (250 psia) for compressor pressure ratios of 12 and 16 respectively. These extraction pressures were chosen sufficiently high to account for pressure losses in the steam lines between the steam and gas turbine and for pressure losses within the turbine blading. The HRSG arrangement and operation is the same as discussed in the previous section.

Combined Cycle with Water-Cooled Turbines

A schematic diagram of the combined cycle system with water-cooled turbines is shown in figure 3. As with the steam-cooled cases, only a small part of the compressor airflow is used in the turbine as blockage flow. The stator-vane coolant water flows from the deaerator and is pumped up to 8.618 MPa (1250 psia). After cooling the stator-vanes, this heated water is then input to three successive flash tanks as shown. The steam from the flash tanks is inducted into the steam turbine, and the remaining water from the last flash tank is returned to the deaerator. The rotor-blade coolant water leaves a polishing unit and is pumped to 1.72 MPa (250 psia). As this water flows through the blade, 70% of it is evaporated to steam and is lost to the gas path, while the remaining water is recovered. This hot water heats the condenser water flow in a closed heater as shown, and then returns to the polishing unit. Water is added to makeup for the water that was evaporated in the turbine rotor blades. All other components of the system are as described in the section on air-cooled gas turbine/steam turbine combined cycles.

PERFORMANCE EFFECT OF THERMAL BARRIER COATINGS

The performance results for air, steam, and water-cooled combined cycle systems are presented in this section for gas turbines with and without TBC. The performance is expressed in terms of combined cycle efficiency (based on HHV of kerosene), and specific power (power output/compressor inlet air flow rate). Also presented is the temperature difference across the thermal barrier coatings for each vane and blade row. The relative performance gains when using thermal barrier coatings with air, steam, and water-cooled gas turbine/combined cycles are compared and discussed.

Combined Cycle with Air Cooling

The performance results for the air-cooled gas turbine/steam turbine combined cycle systems are presented in table 3. With the turbine inlet temperature held constant at 1205°C (2200°F) and the metal surface temperature maintained at 815°C (1500°F), the addition of a thermal barrier coating decreases the vane and blade cooling flow by 50% and 68%, with zirconia coating thicknesses of .038 cm (.015 in) and .076 cm (.030 in), respectively.

The efficiency of a combined cycle system can be expressed as follows:

$$\eta_{cc} = \eta_{gt} + (1-\eta_{gt})\ (f)\ \eta_{st}$$

Where

η_{cc} = combined cycle efficiency

η_{gt} = gas turbine efficiency (including generator losses)

$f = \frac{\text{recovered gas turbine reject heat}}{\text{total gas turbine reject heat}}$

η_{st} - steam cycle efficiency (including generator losses)

Reductions in the air cooling flow affect the combined cycle efficiency in three ways. First, the gas turbine performance increases with decreased cooling flow because of increased turbine power per pound of fuel input. Secondly, since less coolant is injected into the gas turbine gas path, the turbine exhaust temperature increases relative to the uncoated case. This results in more heat being recovered in the HRSG (i.e., higher f factor) and thus higher steam flow rates and more power output from the steam turbine. Finally, the steam cycle efficiency increases slightly with the use of thermal barriers for the particular configurations chosen because with the higher "f" factor, more reject heat is recovered to raise steam at the steam turbine throttle conditions, and less steam at .20 MPa (15 psig) is raised in the low pressure economizer-boiler and inducted into the steam turbine. The overall effect of higher "f" factor and steam cycle efficiency more than offsets the lower (1- gt) factor. Thus, the second term in the combined cycle efficiency equation increases, and hence the combined cycle efficiency improves.

In the last column of table 3, the performance results are shown for a .038 cm (.015 in) TBC case when the turbine inlet temperature is increased to approximately 1370°C (2500°F). The vane and blade cooling flow and metal temperature in this case is the same value as the uncoated case at 1205°C (2200°F). As shown, the performance improvement for this case is greater than that for the two TBC cases in which the coolant flow is reduced while keeping the turbine inlet temperature constant. Without the TBC, more advanced film or transpiration cooling would have to be used to attain this performance at a turbine inlet temperature of 1370°C (2500°F). Thus, the use of TBC might be an alternative to more advanced cooling methods to achieve higher turbine inlet temperature and better performance.

In table 4, the temperature differences across the TBC are shown for the gas turbine/combined cycle cases with TBC. Values are shown for each cooled turbine row. The values in the parentheses for the first stage vanes are the maximum temperature differences across the thermal barrier obtained using the peak temperature at the combustor exit. This peak temperature is a result of the pattern factor as discussed previously. The nonuniformity of the temperature exiting the combustor and entering the first stage vanes is a result of the design of the combustor and the distribution of dilution air into it. Pattern factors of from .2 to .3 are typical of well-designed combustors (A value of .2 was chosen for this analysis - see table 2). Peak temperatures ranged from 171°C (308°F) to 265°C (477°F) higher than the bulk mean temperature for the cases considered in this analysis. The temperature difference across the TBC resulting from the peak combustor exit temperature is seen to be much higher than that corresponding to the bulk mean combustor exit temperature. These large temperature differences are a concern, since they might result in excessive thermal stresses across the TBC of the first stage vanes and lead to coating failure. Thermal stresses were not calculated in this study. However, recent burner rig tests indicate that TBCs can withstand large temperature gradients (up to 600°C (1111°F)) over many 1-hour cycles (ref. 14). In these tests, air-cooled blades with TBCs were exposed to high gas velocities and rapid thermal transients.

In table 4, the number of stages is shown to be four at a turbine inlet temperature of 1205°C (2200°F) and five at 1370°C (2500°F). Also, the first four turbine rows (first two stages) are cooled at 1205°C (2200°F), with seven rows cooled at 1370°C (2500°F). It is interesting to note that for the case without TBC at a turbine inlet temperature of 1205°C (2200°F) (not shown in the table) only the first three blade rows are cooled. Although the total vane and blade cooling flow is decreased by 50 and 68% with TBC thickness of .038 cm (.015 in) and .076 cm (.030 in) respectively, the higher gas temperature entering the fourth turbine row requires that this row be cooled. The gas temperature entering the fourth row is higher because of the reduced cooling in the preceeding rows. Thus, in some cases more turbine rows must be cooled even though the total vane and blade cooling flow is reduced with the use of TBC. This then would require that cooling passages be included in the downstream blade rows, which are more difficult to machine with present equipment because of their greater length.

Combined Cycle with Steam Cooling

The performance results for the steam-cooled gas turbine/steam turbine combined cycle systems are shown in table 5. As mentioned previously, steam to be used as turbine coolant is extracted from the steam turbine. After cooling the turbine vanes and blades, the steam is discharged to the gas path, mixes with the gases entering that turbine row, and expands in later stages. Since no compressor air is used for cooling vanes and blades and the mass flow rate through the turbine increases as the steam is discharged to the gas path, the efficiency and specific power of the steam-cooled gas turbine is much higher than the air-cooled engine. However, the steam turbine power is decreased because of the large percentage of the steam turbine flow that is extracted for cooling. This can be seen by comparing the percent of total power output produced by the steam cycle for the uncoated steam-cooled case in table 5(a) with the uncoated air-cooled case in table 3. Also note that les coolant is required using steam as the coolant because of its improved heat transfer characteristics compared to air.

At a turbine inlet temperature of 1205°C (2200°F), the reductions in cooling flow and increased combined cycle performance compared to the case without TBC is shown for coating thicknesses of .038 cm (.015 in) and .076 cm (.030 in). The percentage reduction in cooling flow rates, when cooling flow is expressed as a percentage of the compressor airflow rate, are approximately the same as for the air-cooled cases. The cooling flow reductions when the steam cooling flow rate is expressed as a percentage of the steam turbine flow are slightly higher since the amount of steam raised in the HRSG per pound of compressor inlet flow is higher for the TBC cases relative to the case without TBC.

The percentage increase in the efficiency is greater for the steam-cooled cases shown in table 5(a) compared to the air cooled cases in table 3, but the percentage increase in specific power with TBC is lower for the steam-cooled cases at 1205°C (2200°F). In the steam-cooled case with TBC, the reduction in steam cooling flow results in a decrease in gas turbine power and an increase in the steam turbine power output. The gas turbine power output decreases because less coolant flow is discharged to the gas path after cooling the vanes and blades and expanded in downstream stages. The steam turbine power output increases for two reasons; 1) less steam is extracted for cooling with more of it expanding down to the condenser pressure, and 2) the gas turbine exhaust temperature is higher in the TBC cases and therefore more steam is raised in the HRSG, with a resulting increase in steam turbine power output per pound of compressor inlet airflow. Thus, even though the specific power with steam cooling still increases relative to the case without TBC, the increase is not as great as with air cooling. This can also be seen from the percent of total power output in table 5, where the percent of power produced from the gas turbine decreases and the steam turbine power output percentage increases with coating thicknesses of .038 cm (.015 in) and .076 cm (.030 in).

In the last column of table 5(a), the turbine inlet temperature was increased to approximately 1370°C (2500°F) with a TBC thickness of .038 cm (.015 in) and the coolant flow rate and metal temperature the same as in the case without TBC. As was shown for the air-cooled case in table 3, increasing the turbine inlet temperature while keeping the same coolant flow rate and metal substrate temperature results in larger performance gains than decreasing the coolant flow rate at a constant metal temperature and turbine inlet temperature of 1205°C (2200°F).

In table 5(b), the results obtained when applying a TBC to a 1425°C (2600°F) steam-cooled gas turbine/steam turbine combined cycle are presented. At this turbine inlet temperature, the cooling flow rates, the ratio of gas turbine to steam turbine power output, and combined cycle performance are higher than shown in table 5(a). The percentage gain in efficiency relative to the uncoated case when TBC is used is larger in table 5(b) since the reductions in coolant flow are larger. For example, the reductions in coolant flow, expressed as a percent of compressor inlet flow, from table 5(b) are 5.4 (i.e., 10.4-4.0) and 7.1 (10.4-3.3) percent for coating thicknesses of .038 cm (.015 in) and .076 cm (.030 in) respectively while the same cooling reductions from table 5(a) are 2.4 (4.7-2.3) and 3.2 (4.7-1.5) percent.

In the last column of table 5(b), the results obtained when raising the turbine inlet temperature to approximately 1675°C (3050°F) while maintaining the same coolant flow rate and metal temperature as the uncoated case at 1425°C (2600°F) are presented. The specific power gain is larger compared to the other TBC cases, but the percentage increase in efficiency is the same as the 1425°C (2600°F) case with a .038 cm (.015 in) TBC, and lower than the case with a .076 cm (.030 in) TBC. Although the turbine inlet temperature is increased, resulting in higher gas turbine efficiency, the steam cycle efficiency is low, since approximately 40 percent of the steam turbine flow is extracted for cooling. This low steam cycle efficiency results in a lower combined cycle efficiency than can be obtained at a turbine inlet temperature of 1425°C (2600°F) with a .076 cm (.030 in) TBC.

The temperature differences across the TBC for the steam-cooled gas turbine/steam turbine combined cycle cases are shown in table 6. The values shown for turbine inlet temperatures of 1205°C (2200°F) and 1370°C (2500°F) (table 6(a)) are similar to those for the air-cooled cases presented in table 4.

The TBC temperature differences for the 1425°C (2600°F) and 1675°C (3050°F) steam-cooled cases are shown in table 6(b). Six turbine rows are cooled for the case without TBC at a turbine inlet temperature of 1425°C (2600°F) (not shown in the table). As explained for the air-cooled case, an additional turbine row is cooled with the turbine using TBC since less coolant is injected into the gas path proceeding the 7th row, resulting in higher gas temperatures entering that row, thus requiring it to be cooled.

Combined Cycle with Water Cooling

The performance results for the water-cooled combined cycle systems are presented in table 7. The turbine inlet temperatures of 1449°C (2639°F) and 1684°C (3062°F) shown in table 7(a) and (b) are those used by the General Electric Company in the HTTT program (ref. 12), and correspond to first stage rotor inlet temperatures of 1425°C (2600°F) and 1650°C (3000°F) respectively. As shown previously for the air and steam-cooled gas turbine cases, the use of thermal barrier coatings at constant turbine inlet temperature results in increased performance relative to the uncoated case. However, for water-cooled turbines, the reductions in cooling flow with TBC do not significantly affect combined cycle performance, since the coolant is not extracted from the gas turbine compressor or steam turbine. The major performance penalty associated with water cooling is the heat loss from the hot gas path to the cooled turbine components. This results in lowering the amount of power produced by the gas turbine and decreasing the turbine exhaust temperature. Lower exhaust temperature results in less heat recovery in the HRSG and thus, less power output from the steam turbine. Although some of this heat is recovered from the coolant by flashing part of the hot coolant to steam and inducting it into the steam turbine, and by using part of the water coolant for feedwater heating (see figure 3), this heat is reintroduced into the cycle at a lower temperature level. As shown in table 7, the use of

TBC significantly reduces the heat loss from the turbine, which results in higher combined cycle performance. The reduction in heat loss and increase in performance are seen to be higher for the higher turbine inlet temperatures (compare tables 7(b) and 7(a)) because of the higher heat transfer rates.

The calculated temperature differences across the thermal barrier coatings for the water-cooled gas turbine/steam turbine combined cycle cases are shown in Table 8. As mentioned previously, a three stage turbine was assumed for all of the water-cooled cases, based on the General Electric water-cooled design used in the HTTT program. As shown in the table, the temperature differences are very high, especially for the first stage vanes at the 1684°C (3062°F) turbine inlet temperature. Again, a major concern of using thermal barrier coatings is their ability to withstand thermal stresses and the harsh operating environments associated with advanced, high temperature gas turbines. The temperature differences shown for the first stage vanes in table 8 are approximately the same magnitude as those in the burner rig tests mentioned previously (ref. 14).

Results are presented only for the water-cooled cases where the turbine inlet temperature is held constant and the cooling loss to the turbine components is reduced with the use of TBC. The effect on performance of increasing the turbine inlet temperature with TBC is not shown for the water-cooled systems, since this would result in higher turbine inlet temperatures than those considered in the HTTT program. Since such high inlet temperatures present severe NO_x emission problems (which have not been addressed in this report) and little is known about the ability of the TBC to withstand such high temperatures, those cases were not included in the tables.

COMPARISON

The efficiency and specific power for combined cycle systems at the three turbine inlet temperatures examined in this report are shown in figure 4(a). Air cooling is used at 1205°C (2200°F), steam cooling is used at 1205°C (2200°F) and 1425°C (2600°F), and water cooling is used at approximately 1425°C (2600°F) and 1650°C (3000°F). The combined cycle performance is shown for the cases without and with TBC in which the turbine inlet temperature is held constant, and the performance is increased by using TBC, enabling a reduction in the required coolant flow and heat losses in the turbine. The combined cycle efficiency improvements with the use of TBC (relative to the efficiencies without TBC) are greater at the higher temperatures. With a .076 cm (.030 in) coating thickness, the efficiency increases are .8 percentage points for air cooling at 1205°C (2200°F), 1.8 percentage points for steam cooling at 1425°C (2600°F) and 2.2 percentage points for water cooling at 1684°C (3062°F). At the higher turbine inlet temperatures, where steam and water are used for cooling, the heat losses from the working fluid to the cooled blades or vanes are higher than the lower temperature cases using air cooling. Therefore, the use of the insulating TBC has a greater effect.

Although the efficiency improvements with TBC are higher at the higher turbine inlet temperature for the steam and water-cooled cases, the specific power improvements, as a percentage of the specific power of the cases without TBC, are greater for the air-cooled case. With a .038 cm (.030 in) coating thickness, the percentage increase in specific power is 6.4% at 1205°C (2200°F) with air cooling, 3.8% at 1425°C (2600°F) with steam cooling and 4.9% at 1684°C (3062°F) with water cooling.

In figure 4(b), the combined cycle performance is shown for the TBC cases in which the metal temperature is held constant and the turbine inlet temperature is increased to a point where the percentage coolant flow is the same as that required for the case without TBC. The dashed line for the water-cooled case indicates an increase in turbine inlet temperature to 1733°C (3150°F), with a corresponding increase in combined cycle performance. Although gas turbines with turbine inlet temperatures above 1705°C (3100°F) represent very far term technology advancements, this one case is presented for comparison with the air and steam-cooled cases.

As shown, maintaining the same coolant flow rate and metal temperature and increasing the turbine inlet temperature for the air and steam-cooled cases at 1205°C (2200°F) result in greater efficiency improvements than was shown in figure 4(a) (by maintaining a constant turbine inlet temperature). However, for steam cooling at 1425°C (2600°F) and water cooling at 1449°C (2639°F), the efficiency improvement when increasing the turbine inlet temperature to1675°C (3050°F) and 1733°C (3150°F) respectively, is not as great as that shown in figure 4(a). As was explained earlier in the report (see results section for steam-cooled gas turbine/steam turbine combined cycle), the high percentage of steam turbine throttle flow being used to cool the gas turbine at 1675°C (3050°F) results in a lower steam cycle efficiency and thus lower combined cycle efficiency than the steam cooled case at 1425°C (2600°F) with a .038 cm (.030 in) coating (shown in table 5(b)). Likewise for the water-cooled turbines, the reduction in the heat loss from the turbine gas path by using TBC and holding the turbine inlet temperature constant (as shown in figure 4(a)), results in greater efficiency improvements than increasing the turbine inlet temperature while maintaining the same heat losses. However, it should be noted that the increase in specific power is much greater when the turbine inlet temperature is increased, and higher specific power generally results in a lower capital cost on a $/kwe basis and, thus, lower cost of electricity.

The improvements in combined cycle specific power shown in figure 4(b) are greater for the steam and water-cooled cases at high turbine inlet temperatures than the air and steam-cooled cases at 1205°C (2200°F) because at these temperatures, the use of thermal barriers allows a greater increase in turbine inlet temperature (i.e., 250°C (450°F) and 284°C (511°F) for 1425°C (2600°F) and 1449°C (2639°F), respectively, compared to 167°C (300°F) at 1205°C (2200°F). Since the specific power is a strong function of the turbine inlet temperature, the greater increase in turbine inlet temperature results in a greater improvement in the combined cycle specific power.

As shown in figure 4, the steam-cooled combined cycle systems have lower efficiency and higher specific power than the air-cooled cases at the same turbine inlet temperature. Also, the water-cooled cases are shown to have lower efficiency and higher specific power compared to the steam-cooled cases. Care should be taken in comparing the performance of the air, steam, and water-cooled systems presented in this report, since the purpose of this report is to compare the performance improvements when using TBC with these three cooling methods, and not a comparison of the relative benefits of the cooling methods themselves. None of the systems were optimized with respect to performance. For example, the selection of other compressor pressure ratios, other steam bottoming cycles, or other configurations than those considered in this analysis would affect the relative performances of combined cycles using these cooling methods. The selections of these parameters for this study were primarily based on previous studies (refs. 8, 9, 10, 11). Although steam cooling appears competitive with water cooling, further detailed parametric performance analyses, beyond the scope of this report, would have to be done to verify this conclusion. Also, detailed cost data would be needed to compare the relative economics of steam and water cooling.

For the air-cooled case shown in figure 4(b), the use of TBC allows the turbine inlet temperature to be increased to 1370°C (2500°F) with the same cooling flow rate and metal temperature as the case without TBC. As mentioned earlier, advanced impingement/convection cooling is assumed for these cases. Without the use of TBC, this turbine inlet temperature and performance could be attained with air cooling only through the use of more advanced film or transpiration cooling methods. Thus, the use of TBC with air cooling may extend the range of turbine inlet temperatures where advanced impingement/convection methods may be used, and would provide an alternative to the more advanced turbine cooling methods.

CONCLUDING REMARKS

The effect on the performance of steam and water-cooled gas turbine/steam turbine combined cycle systems when using TBC has been calculated and compared. These results have also been compared to the performance improvement of combined cycles using air-cooled turbines when using TBC. Combined cycle performance improvements with the use of TBC, relative to the performance without TBC, were generally greater for the steam and water-cooled turbines at nominally 1425°C (2600°F) and 1650°C (3000°F) than for the air-cooled turbines at 1205°C (2200°F). The maximum combined cycle efficiency improvement for the air-cooled cases investigated in this analysis is 1.7 percentage points when the turbine inlet temperature is increased from 1205°C (2200°F) to 1370°C (2500°F) through the use of TBC. A maximum specific power increase of 26.1% also occurs for the same increase in turbine inlet temperature. For the steam-cooled cases, the maximum efficiency increase is 1.9 percentage points with an increase in the turbine inlet temperature from 1205°C (2200°F) to 1370°C (2500°F), while the maximum specific power increase is 32.4% when the turbine inlet temperature is increased from 1425°C (2600°F) to 1675°C (3050°F). For the water-cooled cases, the maximum efficiency increase is 2.2 percentage points at a turbine inlet temperature of 1684°C (3062°F), and the maximum specific power improvement is 36.6% when the turbine inlet temperature is increased from 1449°C (2639°F) to 1733°C (3150°F).

The combined cycle performance improvements through the use of TBC are greater for the steam and water-cooled cases. The reason for this is that at the high turbine inlet temperatures, where steam and water cooling techniques are likely to be used, the reduction in cooling losses are much more significant than at the lower turbine inlet temperatures where air cooling is used. The large temperature differences across the TBC calculated for the steam and water-cooled cases at high turbine inlet temperatures indicate that thermal stresses might present an obstacle to the implementation of TBC on such high temperature gas turbines. The particular temperature differences calculated in this analysis depend on the thermal conductivity assumed. Higher conductivity coating material would result in lower temperature differences for a given thickness. However, for a given thickness, higher conductivity would lessen the effectiveness of the coating. Likewise, decreasing the thickness of the coating to lower the temperature across it would result in lower performance improvements for a given conductivity. Further testing of TBC at high gas temperatures and heat flux are required to determine which trade-offs look attractive.

The use of TBC would result in lower overall coolant usage and losses compared to turbines without TBC. However, in some instances, the use of TBC may result in more turbine stages requiring coolant flow. This may not be desirable for two reasons. First, the machining of cooling passages in longer vanes and blades is more difficult than in smaller airfoils. Also, since the vanes and blades of later stages must withstand higher stress levels, the presence of cooling holes would make the design of these airfoils more difficult.

A primary potential benefit of the use of TBC with air-cooled turbines is that TBC may extend the range of turbine inlet temperatures where advanced impingement/convection air cooling may be used. Without TBC, high efficiency gas turbines at turbine inlet temperatures of up to 1205°C (2500°F) can be achieved with air cooling only through the use of more advanced film or transpiration cooling.

The calculations presented indicate that the performance of steam-cooled gas turbines in combined cycles appears to be competitive with that of water-cooled turbines. More detailed performance and cost data beyond the scope of this report, however, are needed to determine the comparative benefits of these two cooling methods.

The performance predictions presented in the report are optimistic, in that they do not assume any performance degradation of the TBC during its operative lifetime. Actual performance improvements will depend on the ability of the TBC to withstand the harsh operating environments at high turbine inlet temperatures, especially when dirty fuels are used. Further testing of the TBC in these harsh environments and further improvements in the erosion and corrosion resistance of the coatings, will determine whether these performance improvements can be achieved.

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TABLE 1. - SUMMARY OF COMBINED CYCLE CASES INVESTIGATED

Turbine inlet[c] temp. °C (°F)	Air cooled				Steam cooled				Water cooled			
	w/o TBC	With TBC			w/o TBC	With TBC			w/o TBC	With TBC		
		[a]0.038 (0.015)	[a]0.076 (0.030)	[b]0.038 (0.015)		[a]0.038 (0.015)	[a]0.076 (0.030)	[b]0.038 (0.015)		[a]0.038 (0.015)	[a]0.076 (0.030)	[b]0.038 (0.015)
1650 (3000)									X	X	X	X
1425 (2600)					X	X	X	X	X	X	X	X
1205 (2200)	X	X	X	X	X	X	X	X				

[a] Coating thickness cm (in.) same turbine inlet temperature as uncoated-reduced cooling flow.

[b] Coating thickness cm (in.) same cooling flow as uncoated-increased turbine inlet temperature.

[c] Turbine inlet temperatures are 1448° C (2639° F) and 1684° C (3062° F) corresponding to first stage rotor inlet temperatures of 1425° C (2600° F) and 1650° C (3000° F) respectively.

TABLE 2. - SUMMARY OF ASSUMPTIONS

Gas turbine:	
Compressor pressure ratio:	
Turbine inlet temp. = 1205° C (2200° F)	12
Turbine inlet temp. = 1425° C (2600° F), 1650° C (3000° F)	16
Compressor polytropic efficiency	0.9
Turbine polytropic efficiency	0.9
Combustor pattern factor	0.2
Steam cycle:	
Throttle conditions, MPa/°C, (psig/°F)	8.375/510 (1200/950)
Turbine adiabatic efficiency	0.8
Condenser pressure, MPa (in Hg)	0.0084 (2.5)
Minimum HRSG ΔT, °C (°F)	28 (50)
Stack temperature, °C (°F)	149 (300)
Generator efficiency	0.987
Maximum gas turbine blading surface metal temperature °C (°F)	
Air and steam cooling	815 (1500)
Water cooling	538 (1000)
Zirconia conductivity MJ/sec-m^2-°C, (Btu/hr-ft^2-°F)	4.3×10^{-6} (0.75)
NiCrAlY conductivity MJ/sec-m^2-°C (Btu/hr-ft^2-°F)	
Air and steam cooling	2.2×10^{-5} (3.9)
Water cooling	1.9×10^{-5} (3.4)

TABLE 3. - COMBINED CYCLE PERFORMANCE GAINS USING THERMAL BARRIER COATINGS WITH AIR-COOLED TURBINES

		Turbine inlet temperature = 1205° C (2200° F)			Turbine inlet temp. = [c]1370° C (2500° F)
		Without TBC	With[a] TBC 0.038 cm (0.015 in.)	With[a] TBC 0.076 cm (0.030 in.)	With[a] TBC 0.038 cm (0.015 in.)
Efficiency		0.458	0.463 [b](+1.1%)	0.466 [b](+1.7%)	0.475 [b](+3.7%)
Specific power, kW-sec/kg (kW-sec/lb)		480.8 (218.1)	503.3 (228.3) [b](+4.7%)	511.7 (232.1) [b](+6.4%)	606.3 (275.0) [b](+26.1%)
Percent of total power output	Gas turbine	69	69	69	68
	Steam turbine	31	31	31	32
Vane and blade cooling flow, percent of compressor inlet flow		6.6	3.3 [b](-50%)	2.1 [b](-68%)	6.6

[a] 0.010 cm (0.004 in.) bond coat.
[b] Percent change from uncoated case.
[c] Same metal temperature and cooling flow rate as 1205° C (2200° F) case.

TABLE 4. - TEMPERATURE DIFFERENCE ACROSS THERMAL BARRIER COATINGS WITH AIR COOLED TURBINES

		Turbine inlet temperature = 1205° C (2200° F)		Turbine inlet temp. = [c]1370° C (2500° F)
TBC thickness		[a]0.38 cm (0.015 in.)	[a]0.076 cm (0.030 in.)	[a]0.038 cm (0.015 in.)
1st Stage	Vane	[b]173° C (312° F) [249° C (449° F)]	[b]237° C (426° F) [341° C (614° F)]	[b]255° C (459° F) [349° C (628° F)]
	Blade	113° C (203° F)	161° C (290° F)	185° C (333° F)
2nd Stage	Vane	68° C (123° F)	103° C (186° F)	139° C (251° F)
	Blade	30° C (54° F)	48° C (87° F)	92° C (166° F)
3rd Stage	Vane	Uncooled	Uncooled	67° C (121° F)
	Blade	Uncooled	Uncooled	33° C (59° F)
4th Stage	Vane	Uncooled	Uncooled	21° C (38° F)
	Blade	Uncooled	Uncooled	Uncooled
5th Stage	Vane	None	None	Uncooled
	Blade	None	None	Uncooled

[a] 0.010 cm (0.004 in.) bond coat.

[b] Maximum temperature difference using peak combustor temperature.

[c] Same metal temperature and cooling flow rate as 1205° C (2200° F) case.

TABLE 5. - COMBINED CYCLE PERFORMANCE GAINS USING THERMAL BARRIER COATINGS WITH STEAM-COOLED GAS TURBINES

(a) Turbine inlet temperature = 1205° C (2200° F)

		Turbine inlet temperature = 1205° C (2200° F)			Turbine inlet temp. = ^{c}1370° C (2500° F)
		Without TBC	With^a TBC 0.038 cm (0.015 in.)	With^a TBC 0.076 cm (0.030 in.)	With^a TBC 0.038 cm (0.015 in.)
Efficiency		0.452	^{b}0.461 (+2.0%)	^{b}0.463 (+2.4%)	^{b}0.471 (+4.2%)
Specific power, kW-sec/kg (kW-sec/lb)		508.4 (230.6)	517.4 (234.7) ^{b}(+1.8%)	520.7 (236.2) ^{b}(+2.4%)	648.6 (294.2) ^{b}(+27.6%)
Percent of total power output	Gas turbine	76	72	71	73
	Steam turbine	24	28	29	27
Vane and bade cooling flow, percent of compressor inlet flow		4.7	^{b}2.3 (-51%)	^{b}1.5 (-68%)	4.7
Vane and blade cooling flow, percent of steam turbine flow		32.9	^{b}15.6 (-52.6%)	^{b}9.9 (-69.9%)	^{b}22.7 (-31.0%)

[a] 0.010 cm (0.004 in.) bond coat.
[b] Percent change from uncoated case.
[c] Same metal temperature and cooling flow rate as 1205° C (2200° F) case.

TABLE 5. - Concluded.

(b) Turbine inlet temperature = 1425° C (2600° F)

		Turbine inlet temperature = 1425° C (2600° F)			Turbine inlet temp. =[c] 1675° C (3050° F)
		Without TBC	With[a] TBC 0.038 cm (0.015 in.)	With[a] TBC 0.076 cm (0.030 in.)	With[a] TBC 0.038 cm (0.015 in.)
Efficiency		0.475	[b]0.490 (+3.2%)	[b]0.493 (+3.8%)	[b]0.490 (+3.2%)
Specific power, kW-sec/kg (kW-sec/lb)		661.6 (300.1)	681.9 (309.3) [b](+3.1%)	686.5 (311.4) [b](+3.8%)	875.7 (397.2) [b](+32.4%)
Percent of total power output	Gas turbine	82	76	74	78
	Steam turbine	18	24	26	22
Vane and blade cooling flow, percent of compressor inlet flow		10.4	[b]5.0 (-52%)	[b]3.3 (-68%)	10.4
Vane and blade cooling flow, percent of steam turbine flow		56.9	[b]27.7 (-55%)	[b]16.6 (-71%)	[b]39.9 (-30%)

[a] 0.010 cm (0.004 in.) bond coat.

[b] Percent change from uncoated case.

[c] Same metal temperature and cooling flow rate as 1425° C (2600° F) case.

TABLE 6. - TEMPERATURE DIFFERENCE ACROSS THERMAL BARRIER COATINGS FOR STEAM COOLED TURBINES

(a) Turbine inlet temperature = 1205° C (2200° F)

		Turbine inlet temperature = 1205° C (2200° F)		Turbine inlet temp. = [c]1370° C (2500° F)
TBC thickness		[a]0.038 cm (0.015 in.)	[a]0.076 cm (0.030 in.)	[a]0.038 cm (0.015 in.)
1st Stage	Vane	[b]173° C (311° F) [249° C (448° F)]	[b]237° C (426° F) [341° C (614° F)]	[b]259° C (466° F) [354° C (637° F)]
	Blade	111° C (200° F)	159° C (287° F)	187° C (336° F)
2nd Stage	Vane	66° C (118° F)	101° C (182° F)	138° C (249° F)
	Blade	28° C (51° F)	47° C (84° F)	92° C (165° F)
3rd Stage	Vane	Uncooled	Uncooled	66° C (118° F)
	Blade	Uncooled	Uncooled	32° C (58° F)
4th Stage	Vane	Uncooled	Uncooled	19° C (35° F)
	Blade	Uncooled	Uncooled	Uncooled
5th Stage	Vane	None	None	Uncooled
	Blade	None	None	Uncooled

[a] 0.010 cm (0.004 in.) bond coat.

[b] Maximum temperature difference using peak combustor temperature.

[c] Same metal temperature and cooling flow rate as 1205° C (2200° F) case.

TABLE 6. - Concluded.

(b) Turbine inlet temperature = 1425° C (2600° F)

		Turbine inlet temperature = 1425° C (2600° F)		Turbine inlet temp. = [c]1675° C (3050° F)
TBC thickness		[a]0.038 cm (0.015 in.)	[a]0.076 cm (0.030 in.)	[a]0.038 cm (0.015 in.)
1st Stage	Vane	317° C (570° F) [b][423° C (761° F)]	413° C (743° F) [b][551° C (992° F)]	451° C (811° F) [b][584° C (1051° F)]
	Blade	235° C (423° F)	321° C (578° F)	350° C (630° F)
2nd Stage	Vane	166° C (299° F)	241° C (433° F)	271° C (488° F)
	Blade	113° C (204° F)	172° C (309° F)	209° C (376° F)
3rd Stage	Vane	74° C (134° F)	120° C (216° F)	159° C (287° F)
	Blade	38° C (68° F)	67° C (120° F)	113° C (204° F)
4th Stage	Vane	19° C (34° F)	38° C (68° F)	83° C (149° F)
	Blade	Uncooled	Uncooled	50° C (90° F)
5th Stage	Vane	Uncooled	Uncooled	33° C (60° F)
	Blade	Uncooled	Uncooled	Uncooled
6th Stage	Vane	None	None	Uncooled
	Blade	None	None	Uncooled

[a] 0.010 cm (0.004 in.) bond coat.

[b] Maximum temperature difference using peak combustor temperature.

[c] Same metal temperature and cooling flow rate as 1425° C (2600° F) case.

TABLE 7. - COMBINED CYCLE PERFORMANCE GAINS USING THERMAL BARRIER COATINGS WITH WATER COOLED TURBINES

(a) Turbine inlet temperature = 1449° C (2639° F)

		Turbine inlet temperature = [a]1449° C (2639° F)		
		Without TBC	With[b] TBC 0.038 cm (0.015 in.)	With[b] TBC 0.076 cm (0.030 in.)
Efficiency		0.469	0.483 [c](+3.0%)	0.488 [c](+4.1%)
Specific power, kW-sec/kg (kW-sec/lb)		675.6 (306.3)	695.9 (315.5) [c](+3.0%)	703.0 (318.7) [c](+4.0%)
Percent of total power output	Gas turbine	72	71	71
	Steam turbine	28	29	29
Vane and blade cooling, percent of compressor inlet flow		15.0	10.0 [c](-33.3%)	7.6 [c](-49.3%)
Total turbine heat loss, MJ/kg-compressor inlet (Btu/lb-compressor inlet)		0.115 (49.6)	0.0794 (34.1) [c](-31.2%)	0.0619 (26.6) [c](-46.4%)

[a] 1425° C (2600° F) rotor inlet temperature for uncoated case.
[b] 0.010 cm (0.004 in.) bond coat.
[c] Percent change from uncoated case.

TABLE 7. - Concluded.

(b) Turbine inlet temperature = 1684° C (3062° F)

		Turbine inlet temperature = [a]1684° C (3062° F)		
		Without TBC	With[b] TBC 0.038 cm (0.015 in.)	With[b] TBC 0.076 cm (0.030 in.)
Efficiency		0.472	0.487 [c](+3.2%)	0.494 [c](+4.7%)
Specific power, kW-sec/kg (kW-sec/lb)		865.5 (392.6)	894.2 (405.6) [c](+3.3%)	907.6 (411.7) [c](+4.9%)
Percent of total power output	Gas turbine	71	70	70
	Steam turbine	29	30	30
Vane and blade cooling, percent of compressor inlet flow		19.7	12.2 [c](-38.1%)	9.1 [c](-53.8%)
Total turbine heat loss, MJ/kg-compressor inlet (Btu/lb-compressor inlet)		0.186 (79.7)	0.119 (51.2) [c](-35.8%)	0.0903 (38.8) [c](-51.3%)

[a] 1650° C (3000° F) rotor inlet temperature for uncoated case.
[b] 0.010 cm (0.004 in.) bond coat.
[c] Percent change from uncoated case.

TABLE 8. - TEMPERATURE DIFFERENCE ACROSS THERMAL BARRIER COATINGS WITH WATER COOLED TURBINES

(a) Turbine inlet temperature = 1449° C (2639° F)

		Turbine inlet temperature = [a]1449° [illegible] (2639° F)	
TBC thickness		[b]0.038 cm (0.015 in.)	[b]0.076 cm (0.030 in.)
1st Stage	Vane	356° C (640° F) [c][437° C (787° F)]	509° C (916° F) [c][626° C (1126° F)]
	Blade	313° C (564° F)	440° C (792° F)
2nd Stage	Vane	188° C (339° F)	287° C (517° F)
	Blade	128° C (231° F)	201° C (361° F)
3rd Stage	Vane	71° C (127° F)	121° C (218° F)
	Blade	60° C (108° F)	99° C (178° F)

[a] 1425° C (2600° F) rotor inlet temperature for uncoated case.

[b] 0.010 cm (0.004 in.) bond coat.

[c] Maximum temperature difference using peak combustor temperature.

TABLE 8. - Concluded.

(b) Turbine inlet temperature = 1684° C (3062° F)

		Turbine inlet temperature = [a]1684° C (3062° F)	
TBC thickness		[b]0.038 cm (0.015 in.)	[b]0.076 cm (0.030 in.)
1st Stage	Vane	536° C (964° F) [c][655° C (1179° F)]	726° C (1307° F) [c][888° C (1559° F)]
	Blade	467° C (840° F)	620° C (1116° F)
2nd Stage	Vane	255° C (459° F)	386° C (694° F)
	Blade	216° C (388° F)	323° C (581° F)
3rd Stage	Vane	128° C (231° F)	209° C (376° F)
	Blade	87° C (157° F)	151° C (271° F)

[a] 1650° C (3000° F) rotor inlet temperature for uncoated case.

[b] 0.010 cm (0.004 in.) bond coat.

[c] Maximum temperature difference using peak combustor temperature.

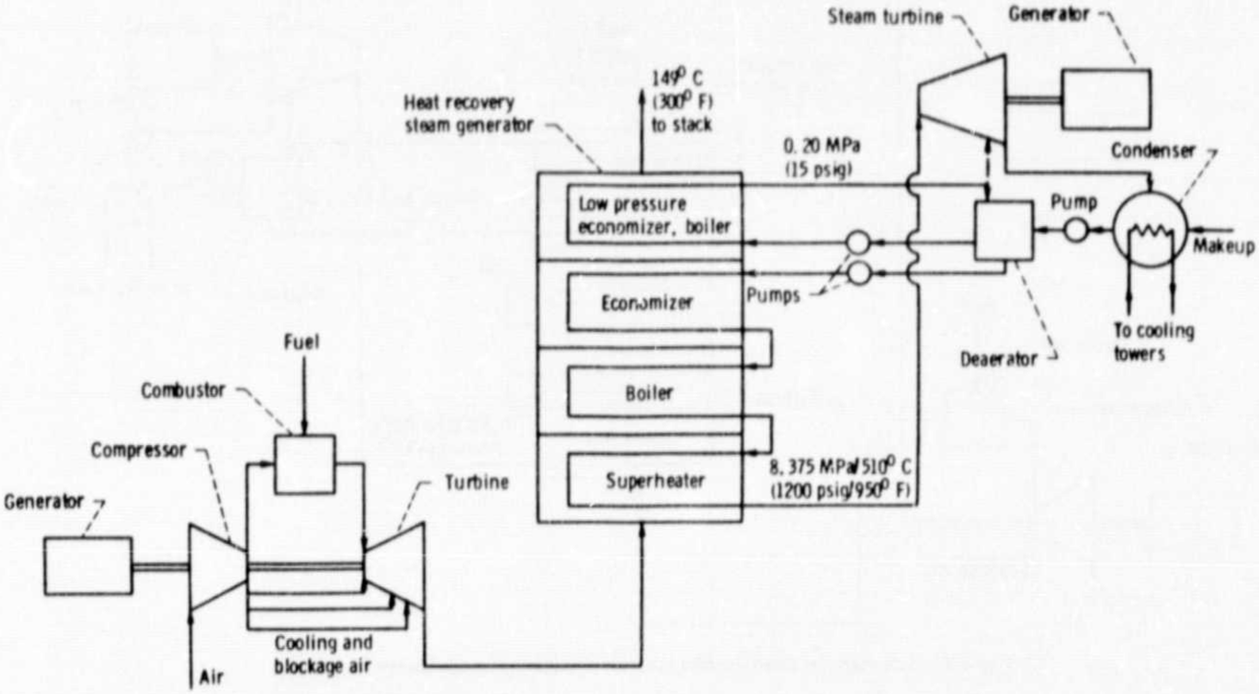


Figure 1. - Schematic for air-cooled gas turbine/steam turbine combined-cycle system.

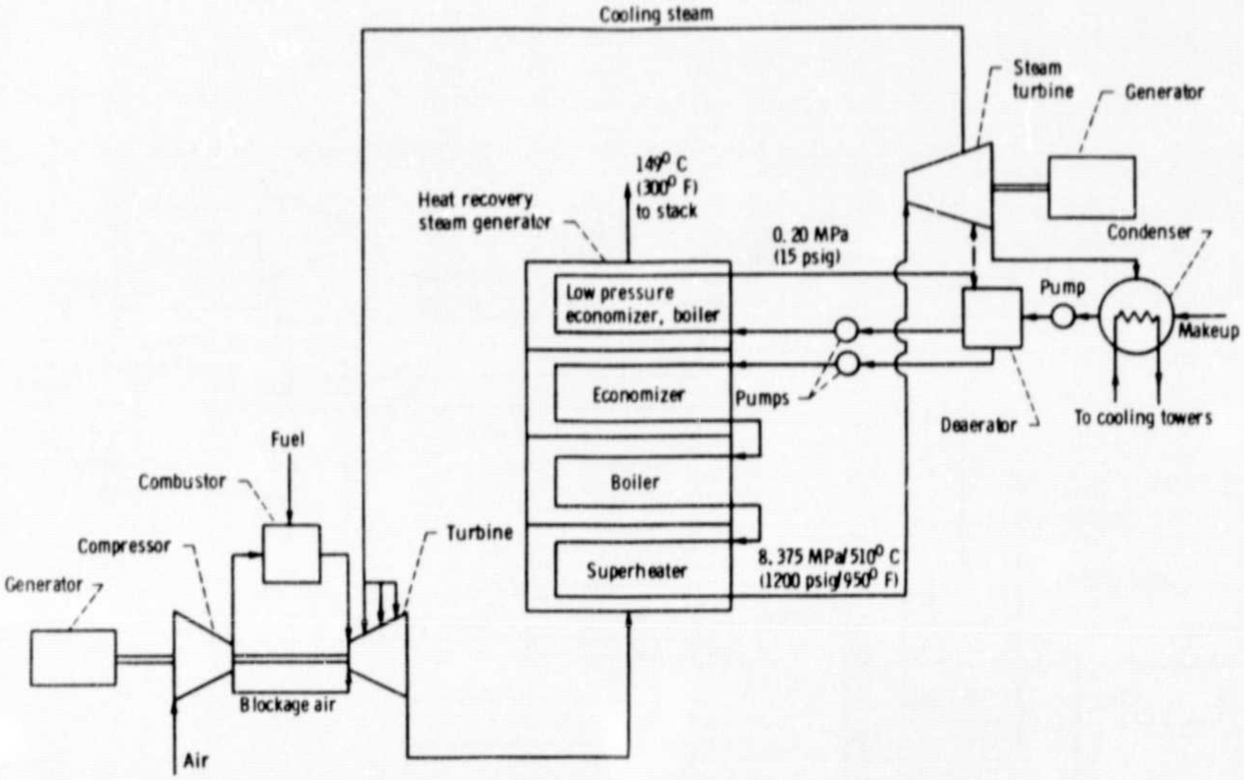


Figure 2. - Schematic for steam-cooled gas turbine/steam turbine combined-cycle system.

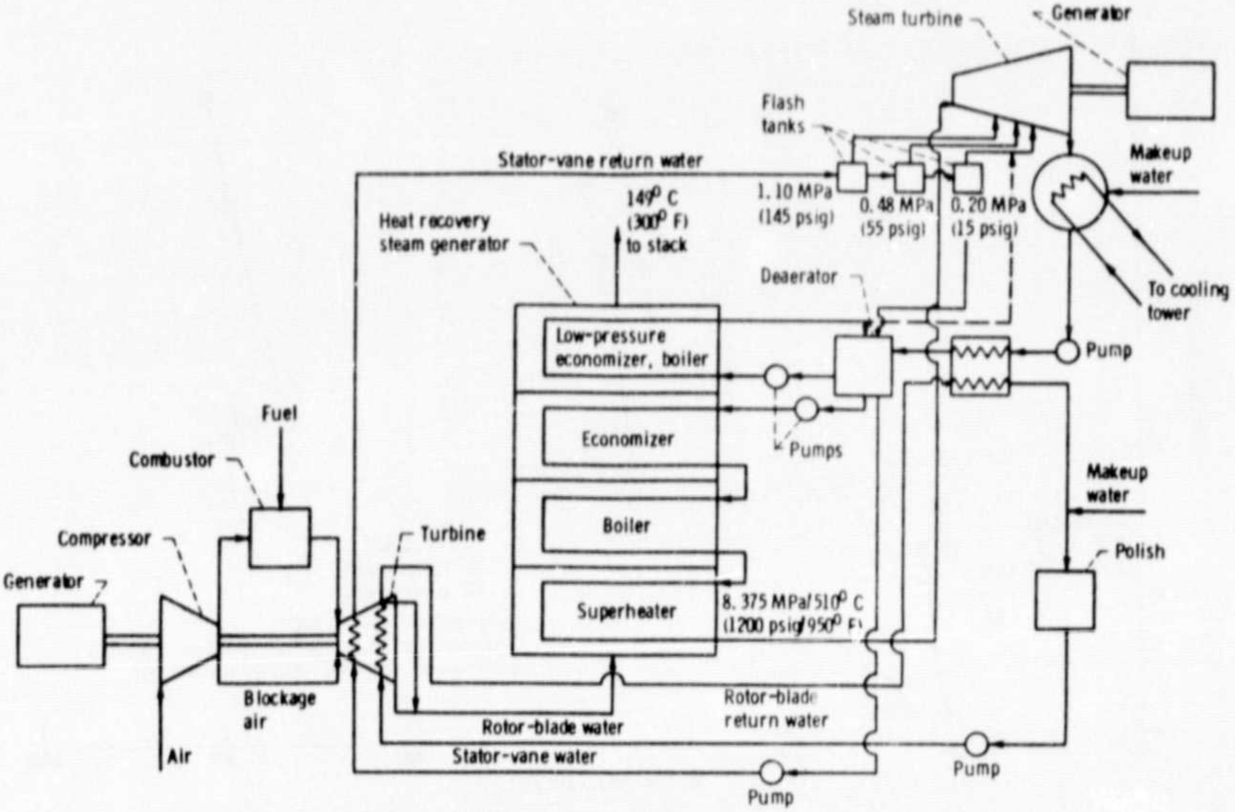
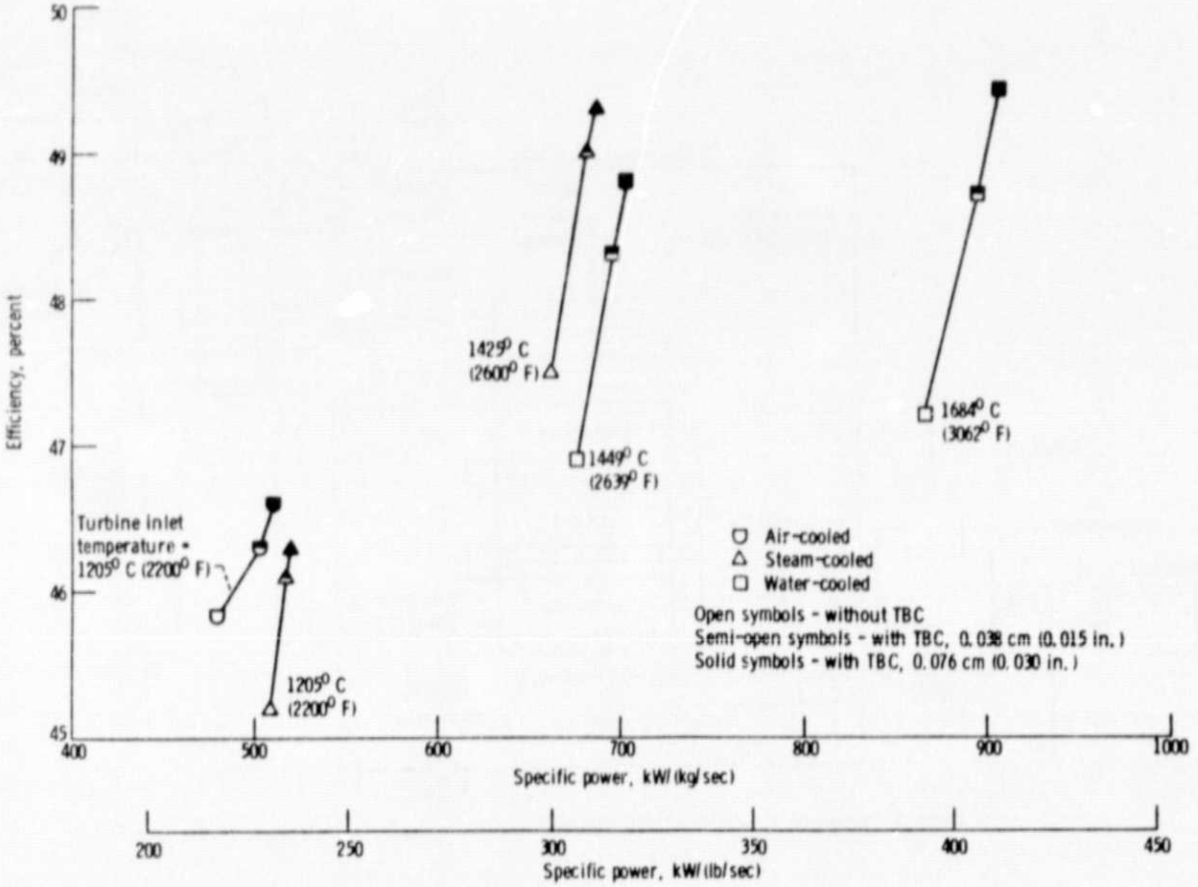
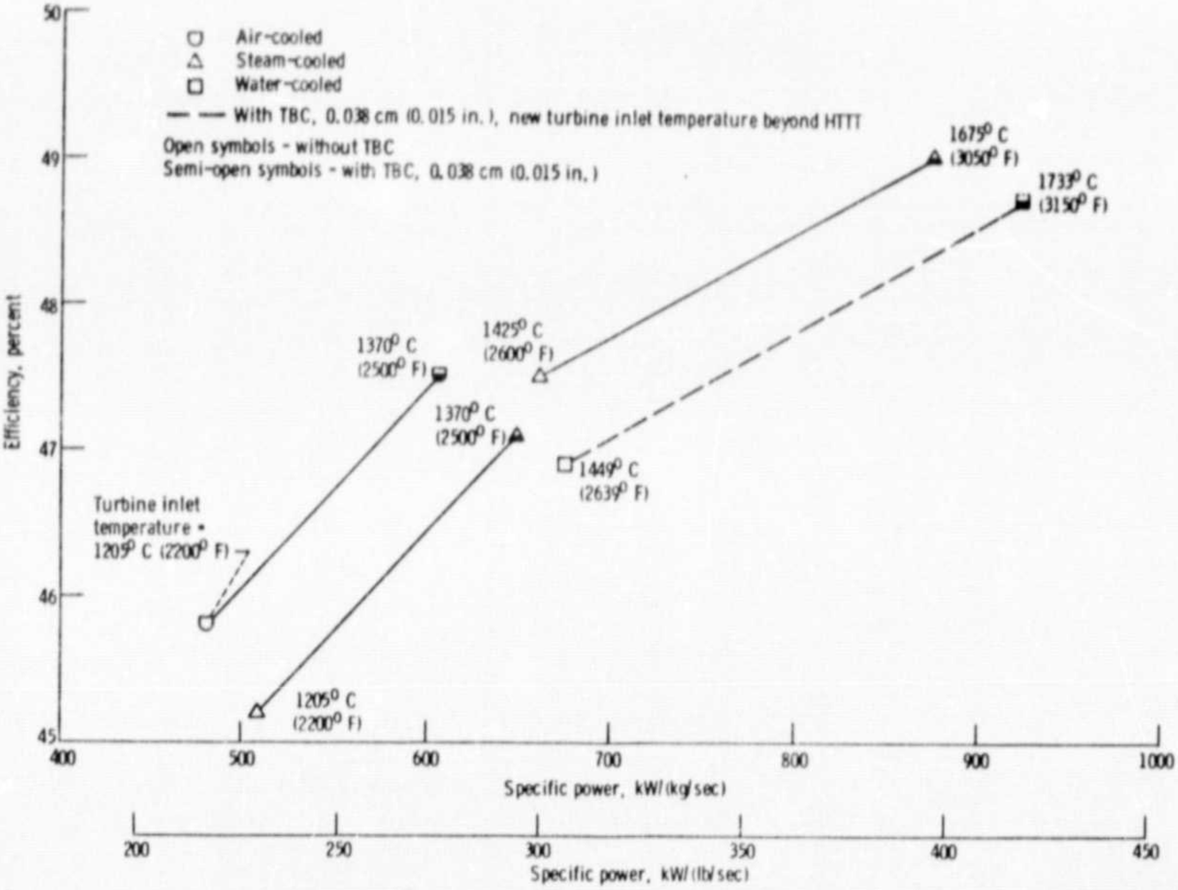


Figure 3. - Schematic for water-cooled gas turbine/steam turbine combined cycle system.



(a) Constant turbine inlet temperature, reduced cooling flow.

Figure 4. - Combined cycle performance gains with thermal barrier coatings.



(b) Constant cooling flow, increased turbine inlet temperature.

Figure 4. - Concluded.