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EXTERNALLY FIRED EVAPORATIVE GAS TURBINE WITH A CONDENSING HEAT EXCHANGER

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ABSTRACT

The integration of externally fired gas turbines and evaporative gas turbines (e.g., the HAT cycle) can offer the features from both systems: using solid fuel without requiring particulate clean up to protect the gas turbine path, and having the potential to enhance the power output and increase the efficiency without including a bottoming steam turbine. Exhaust gases from externally fired evaporative gas turbines have a high moisture content. Using a condensing heat exchanger makes it possible to recover more exhaust heat, thereby providing more potential for improving the system performance. This paper presents a new system with integration of a condensing heat exchanger in an externally fired evaporative gas turbine. The first and second laws of thermodynamics have been used to analyze the system. This study extends the overall knowledge on the externally fired evaporative gas turbine system and provides an investigation of the system with a condensing heat exchanger.

NOMENCLATURE

B_f	Exergy of fuel
B_p	Exergy of process heat
E_{in}	Energy input
LHV	Lower heating value
M_f	Mass flow rate of fuel
Q_p	Process heat output
T_0	Environmental temperature
W_e	Power output
Δh	Enthalpy difference
Δs	Entropy difference
α	Power-to-heat ratio
ϕ_f	Exergy factor of fuel
ϕ_p	Exergy factor of process heat
η_e	Electrical efficiency
η_I	First -law efficiency
η_{II}	Second-law efficiency

INTRODUCTION

Energy resources and environmental impacts in both the near and long-term have become main issues for power generation. Solid fuels such as coal and biomass will play a more important role as the substitute for premium fuels and make a significant contribution to long-term energy stability. Currently, about 40 % of the electricity in Europe and worldwide is based on coal. Coal will continue to be a vital factor in the world economy, and its production can be expected to expand greatly over the next decades. In Sweden, biomass accounts for a significant portion of the primary energy supply, mainly for industrial processes and district heating. Biomass, as a clean domestic resource for energy production, is important not only for the security of the energy supply, but also for its potential benefits to the environment and to rural and regional development. Growing environmental challenges will create greater incentives to develop and deploy innovative technologies which use fuels in a cleaner, more efficient and more economic way.

The Swedish National Board for Industrial and Technical Development (NUTEK) has initiated a national program called *New Processes for Power and Heat Generation* (Rosén et al, 1994; and Eidensten et al, 1994b). One of the main goals of this program was to develop a technology base for improving thermal efficiency and environmental performance of future power plants. This was followed by another program called *Evaporative Gas Turbine (EvGT)*, where aim was to demonstrate the evaporative gas turbine technology. This national project, with a budget of about 14 million SEK, is being conducted in cooperation with Swedish universities and industrial companies, including electric utilities and engineering and manufacturing companies. The study presented here is an extension of these programs. It deals with a new system configuration, which integrates the evaporative cycle (sometimes called the HAT cycle) and externally fired gas turbine with a condensing heat exchanger (a heat recovery system by flue gas condensation). This system may have more potential to improve performance by recovering more

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exhaust heat than the system without a condensing heat exchanger. It will also be environmentally beneficial since air pollution is reduced by the exhaust gas cleaning process. This study is presented mainly from the thermodynamic viewpoints. Some technical discussion on the use of condensing beat exchangers in the externally fired evaporative gas turbine has also been addressed.

PREVIOUS WORK

Most of the studies on externally fired gas turbines have involved the externally fired combined cycles (EFCC) in which a bottoming steam turbine is involved. Recent studies can be found in the literature (LaHaye and Zabolotny, 1989, Sjödin and Svedberg, 1992, Eidensten et al, 1994a, Klara, 1994, LaHaye and Bary, 1994 and Yan et al, 1994a).

Recent work on the evaporative gas turbine cycle has concentrated on the directly fired gas turbine which operates only on premium fuels. The gas turbine with water injection was proposed by Gasparovic and Stapersma (1973) and was further studied by Mori et al (1983). Frutschi et al (1988) presented a cycle called the Intercooler/Recuperated Evaporation Cycle, which is a type of evaporative gas turbine cycle. The advanced gas turbine systems including the evaporative gas turbine were compared by Annerwall and Svedberg (1991). The heat recovery system coupled with the evaporative gas turbine cycle was investigated by Barthelemy and Lynn (1991). Fluor Daniel Inc. studied and patented the HAT cycle (Rao and Joiner, 1990, Day and Rao, 1992). They compared the HAT cycle with combined-cycle power plants integrated with coal gasification process and with natural gas-fired plants (Fluor Daniel, 1991, 1993). The potential to integrate the HAT cycle with other new approaches to power generation was discussed by Cohn (1993). These new approaches are the CASH (Compressed-Air Storage with Humidification) cycle, IGCASH (Integrated Gasification CASH), and CASHING (Compressed-Air Storage with Humidification, Integrated with Natural Gas). The configuration variations of the HAT cycle and their performance were also studied by Stecco et al (1993) and Chiesa et al (1994). Recently the CHAT (Cascaded Humidified Advanced Turbine) was proposed by Westinghouse Electric for a nominal 300 MW power plant (Nakhamkin et al, 1995). This CHAT power plant is designed around an industrial turbocompressor and expander train and separate combustion turbine train — mounted on independent shafts — which exchange working fluid through a series of heat exchanger, intercooler, reheat, air saturator, and recuperator stages.

The externally fired evaporative gas turbine system is the integration of the externally fired gas turbine and the evaporative cycle. Performance benefits available from compressor discharge water injection in an externally fired gas turbine were discussed by Parsons and Bechtel (1991). An externally fired gas turbine with evaporative water addition, in which the system is fueled by biogas, was discussed by De Ruyck et al (1991). A case study was performed on a small scale externally fired evaporative gas turbine fueled by solid biomass (Eidensten et al, 1994b). The overall thermodynamic performance on an externally-fired humid air turbine cycle was studied by Huang and Naumowicz (1994). The effect on performance of the inlet temperature and pressure ratio, amount of injected water, turbine efficiency and intercooling was examined by Yan et al (1994b). The interactive parameters among the three subsystems of the externally fired gas turbines (the gas turbine, solid fuel combustion, and heat recovery subsystems) were investigated by Yan et al (1995).

Generally, performance advancements in gas turbines will continue in the following areas (McDonald, 1994): (1) materials technology and the use of more sophisticated turbine cooling techniques, permitting higher firing temperatures with metallic stationary and rotating parts and a transition to ceramics as stoichiometric combustion conditions are approached, and (2) continued gains in compressor and turbine efficiencies by the use of advanced blading, close tolerances, and developing computational fluid dynamics (CFD) methodology, and (3) use of heat exchangers in some form or other (e.g. intercooler, recuperator, steam generator, reformer). Heat exchangers will play an important role across the full spectrum of gas turbine applications in coming decades. A wide range of advanced thermodynamic cycles with high efficiency potential are under investigation. This paper studies one of these advanced cycles, the externally fired evaporative gas turbine with a flue gas condensing heat exchanger.

CONDENSING HEAT EXCHANGER AND GAS CLEANING

Flue gases from any type of combustion contain heat in the form of sensible heat and latent heat of water vapor from the water and hydrogen content in the fuel. In an advanced heat recovery system, both the sensible and latent heat in the flue gas are recovered. Flue gas condensation, a heat recovery system with a condensing heat exchanger, is a technique for the simultaneous recovery of heat and capture of pollutants from a flue gas. When the flue gas is cooled to below the dew point, the water vapor starts to condense. During the cooling process, sensible heat in the flue gas and latent heat in the water vapor are recovered for another process.

Flue gas condensation is also of benefit for environmental reasons. Dust and noxious gases in the flue gases are captured in the condensed water and removed with it. A high degree of separation is obtained for dioxins, chlorine, mercury, dust, and to some extent, sulfur oxides. Of the nitrogen oxides formed during combustion, only nitrogen dioxide is slightly soluble in water. Therefore, only a minor part of the nitrogen oxides is removed.

Basically, there are two different approaches to flue gas condensation. The flue gases can be cooled either directly in contact with water or indirectly in a heat exchanger. Fig. 1 shows one configuration of an industrial flue gas condensation plant using solid biomass fuel (Sandberg, 1995) where a gas turbine is not involved in the system. The humidifier for combustion air is used to increase the dew point temperature of the flue gases. Thus, the heat from the flue gas can be recovered for the district heating at a higher temperature.

Systems for advanced heat recovery from flue gases have been used for a long time in, for example, the pulp industry. There is growing use of flue gas waste heat recovery in other fields, such as in district heating plants, especially in those plants using the fuel with a high moisture content, such as domestic refuse, wood chips and peat, and also in plants using natural gas. Several processes for flue gas heat recovery have been developed in Sweden and other countries. The largest installation for flue gas condensation is in Uppsala (Eidensten et al, 1995). Annually

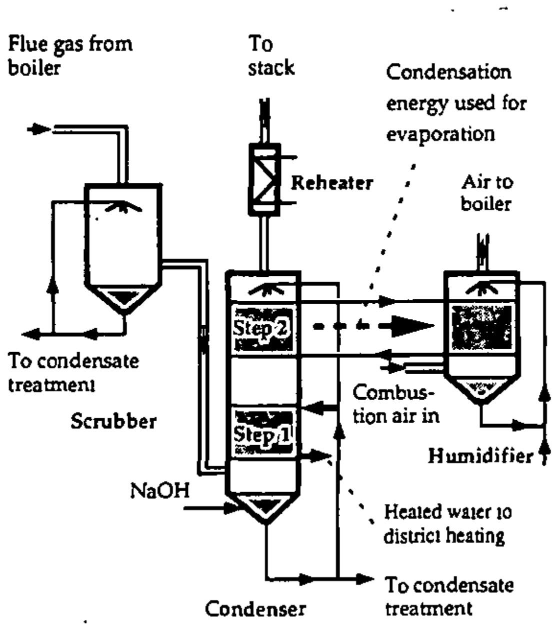

Fig. 1 Example of flue gas condensation plant

250,000 tons of municipal solid wastes are combusted, producing 120 MW of electricity and 190 MW of heat for the district heating system. The total energy output from the flue gas condensation is 22 MW, which is an increase of 30 % relative to the case without flue gas condensation. In addition, flue gas condensation reduces the emission of pollutants. Most of the hydrogen chloride, dioxins, particulates, and heavy metals, such as mercury, are removed.

A study of flue gas condensation installations in five Swedish plants showed good performance and maintenance results. Operating costs are low and profitability is high, with an average increase in energy recovery of 20-30 %. Emissions of pollutants are low in all plants (Brännland, 1992).

In this study, a condensing heat exchanger in an externally fired evaporative gas turbine is used with the following aims:

- To lower stack temperature and recover more heat from the exhaust gas.
- To reduce pollutant emission.
- To recover make-up water.

SIMULATION SYSTEM AND PERFORMANCE CRITERIA

The configuration of the system studied in this paper is shown in Fig. 2. The main assumptions and system inputs are listed in the Appendix. The power required for fans, pumps, etc. in the system has not been taken into account. The gas turbine is a cooled turbine with an inlet temperature of 1120°C and a pressure ratio of 13.6. These data are selected based on an industrial gas turbine. The solid biomass with 50% moisture content is fired in a biomass combustion system. In the district heating system, 70°C is assumed as supplied water temperature and 40°C as return water temperature.

From the viewpoints of energy conversion and transformation, the system is divided into three main subsystems:

- gas turbine subsystem (GT) to generate power;
- solid fuel combustion subsystem (SFC) to supply energy (the solid fuel is biomass in this paper);
- heat recovery subsystem (HR) to recover the exhaust heat from both the GT and SFC subsystems for humidifying and heating compressed air and also for preheating combustion air for the SFC. Condensing heat exchangers are the final step in recovering exhaust heat from the flue gas. The recovered heat in the condensing heat exchangers is used for district heating.

As shown in Fig. 2, the GT consists of a compressor, the turbine and the topping combustor. The topping combustor is optional. If the ceramic high-temperature heat exchangers are capable of raising the air temperature to the turbine inlet temperature, the topping combustor will not be necessary. The SFC includes the solid fuel combustor and the high-temperature heat exchanger. The HR is made up of an intercooler, aftercooler, economizers, humidification tower, recuperators, combustion air preheater, and condensing heat exchangers.

The GT is an essential subsystem to generate power. The SFC, together with the topping combustor of the GT, is the energy source for the whole system. It allows solid fuel to be used as the main energy input to the gas turbine by means of external firing. The key equipment of the biomass combustion system is the high-temperature heat exchanger. The HR makes use of the exhaust gas streams from the GT and the SFC as effectively as possible to improve the thermodynamic performance of the whole system. In the HR, several heat exchangers are coupled to each other. Heat supplied to the heat recovery subsystem can come from three sources:

- compressed air via the intercooler and aftercooler,
- exhaust gas from the gas turbine and
- flue gas from the solid fuel combustor.

Recovered heat is used for four purposes:

- increasing the temperature of process water for the humidification of compressed air in the humidification tower,
- preheating combustion air in the air preheaters for the solid fuel combustor,
- increasing the humid air temperature in the recuperators, and
- increasing the process water temperature in the condensing heat exchangers for district heating.

This paper intends to investigate the heat recovery subsystem, and especially the condensing heat exchangers. The system is a cogeneration system producing power and heat. Thus, it is important to define the performance criteria used to evaluate the system. On the basis of thermodynamics (Huang, 1985, Yan et al, 1994a), system performance can be defined according to energy input and output. Two useful products of a cogeneration system are electrical energy (W_e) and thermal energy or process heat (Q_p). The energy input into the system is supplied by the biomass fuel. The energy input (E_{in}) can be calculated from the mass flow rate (M_f) and heating value (LHV, lower heating value is used in this paper) of the biomass fuel.

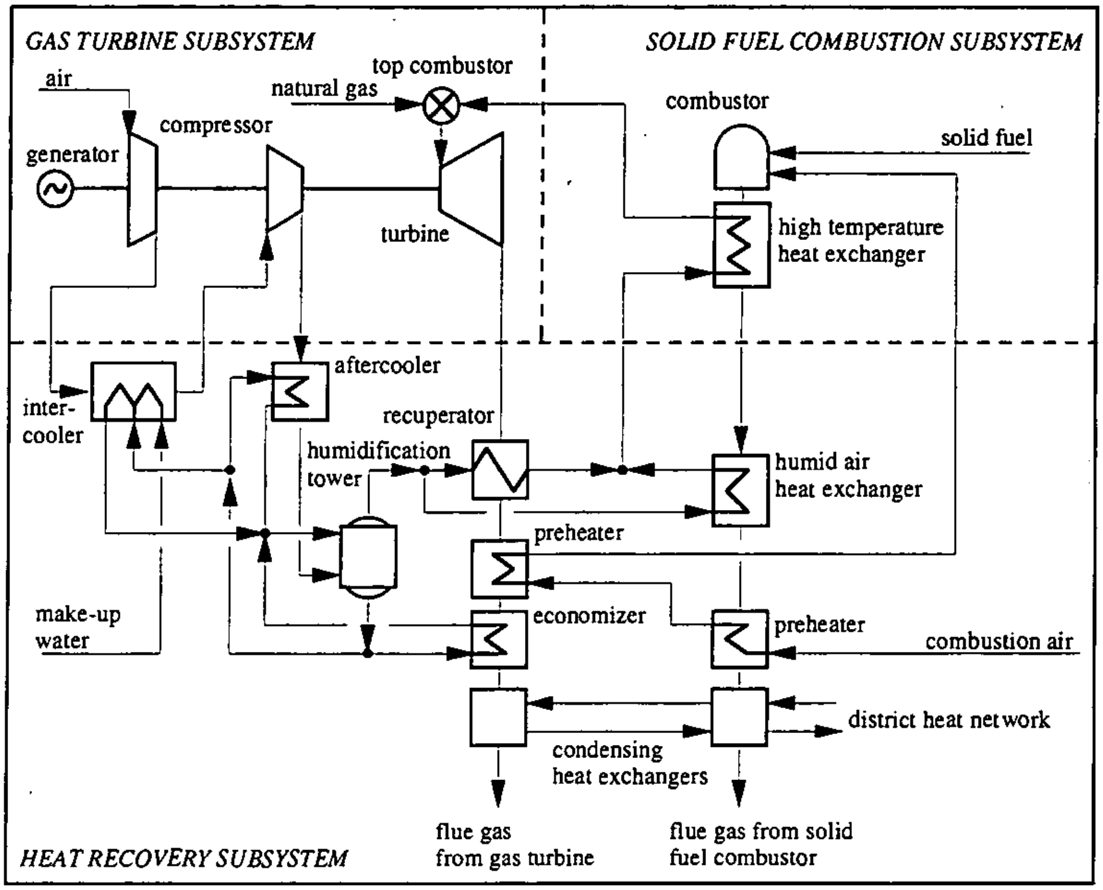

Fig. 2 The conceptual system of the biomass externally fired evaporative gas turbine with condensing heat exchangers

The first-law efficiency (fuel utilization efficiency) of the system can be defined as:

$$\eta_I = \frac{\text{Energy Output}}{\text{Energy Input}} = \frac{W_e + Q_p}{E_{in}} = \frac{W_e + Q_p}{M_f(LHV)} \quad (1)$$

This efficiency makes no distinction between electrical energy and thermal energy.

The electrical efficiency is defined as:

$$\eta_e = \frac{\text{Power Output}}{\text{Energy Input}} = \frac{W_e}{E_{in}} = \frac{W_e}{M_f(LHV)} \quad (2)$$

Another commonly used performance criterion, the power-to-heat ratio, is defined as:

$$\alpha = \frac{W_e}{Q_p} \quad (3)$$

The first-law based approach to cycle analysis, especially when applied to cogeneration systems, has inherent limitations because the energy-balance makes no distinction between heat and work and contains no provision for quantifying the "quality" of the heat. Thus, when two final products, heat and power are produced by a cogeneration system, the first-law analysis does not provide sufficient information for improving the system. The second law of thermodynamics, applied in the form of entropy and availability balances, makes the distinction between heat and work. The second-law efficiency (exergetic efficiency) is defined as:

$$\eta_{II} = \frac{\text{Exergy Output}}{\text{Exergy Input}} = \frac{W_e + B_p}{B_f} \quad (4)$$

where B_p is the exergy content of process heat, and B_f is the exergy content of the fuel. An exergy factor ϕ is introduced as the exergy fraction of energy:

(2)

$$\phi_p = \frac{B_p}{Q_p} \quad (5)$$

$$\phi_f = \frac{B_f}{E_{in}} \quad (6)$$

The second-law efficiency can be rearranged as:

$$\eta_{II} = \frac{\eta_I}{\phi_f}\left(\frac{\alpha + \phi_p}{\alpha + 1}\right) \quad (7)$$

The exergy factor of process heat is always less than unity. In this paper, the process heat is supplied to the district heating system. The exergy factor of process heat can be calculated by:

$$\phi_p = 1 - T_0 \frac{\Delta s}{\Delta h} \quad (8)$$

where Δs and Δh are entropy difference and enthalpy difference between return and supply water respectively in the district heating network. When the return water and the supply water temperatures are assumed to be 45°C and 70°C respectively and the environmental temperature is 15°C, the exergy factor of process heat (ϕ_p) is 0.129. This means that the exergy content of process heat is 12.9 % in this case.

Since there is little difference between energy content and exergy content of fuel (Moran, 1989), the exergy factor of fuel is assumed to be equal to one. In addition, the power-to-heat is usually less than one. The second-law efficiency is significantly less than the first-law efficiency. This important point reveals that the second-law analysis is very valuable for cogeneration systems.

Based on equation (7), Fig. 3 is plotted to show the relationship between the first-law efficiency and the second-law efficiency with variables of power-to-heat ratio. It shows that the second-law efficiency is always much lower than the first-law efficiency in cogeneration plants.

RESULTS AND DISCUSSIONS

The externally fired evaporative gas turbine cycle integrated with the condensing heat exchanger is simulated by using the process simulator, ASPEN PLUS®. The cycle efficiency is based on lower heating value. The simulation results are presented in the following.

Dew Point Temperature and Water-to-Air Ratio

In the evaporative gas turbine (HAT) cycle, water is added into a compressed air in the humidification tower. The amount of latent heat in the flue gases is higher than in the gas turbine without water injection. In the biomass externally fired gas turbine, most of the water vapor in the biomass combustion flue gases comes from the water content in the wet biomass fuel.

The dew point temperature of combustion gases is one of the important parameters which should be considered when using condensing heat exchangers. Figure 4 shows the dew point temperatures of flue gas from both the gas turbine and biomass combustor. Water-to-air ratio means the ratio of water added into the gas turbine to the inlet air in the compressor. The dew point temperature of the gas turbine combustion gas is increased with increasing water-to-air ratio. When the water-to-air ratio is between 0.15 to 0.35, the dew point temperature of flue gas from the gas turbine is between 62 and 75°C. The dew point temperature of flue gas from the biomass combustor, however, does not depend on the water-to-air ratio, since changes in the water-to-air ratio do not affect the vapor content of the combustion gases from the biomass combustor. The dew point temperature of biomass combustion gases is 66°C. The dew point temperatures of the flue gases from both biomass combustor and gas turbine are higher than those of flue gases in normal gas turbine systems.

In this paper, the flue gas temperature is varied between 100°C to 50°C. This means that sometimes only part of the water vapor in the flue gas is condensed. The discussion in a later section shows that the dew point temperature is very important for the performance of the system.

Heat Recovery by the Condensing Heat Exchanger

As shown in Fig. 2, power output is the main product of the system. In the heat recovery system, the heat from the flue gas is first recovered in the recuperator, humid air heat exchanger, air preheater and economizer. Using condensing heat exchangers makes it possible to lower the stack temperature to recover more exhaust heat for district heating.

Fig. 5 shows the simulation results on heat recovery when condensing heat exchangers are used. The heat recovered in the condensing heat exchangers is highly dependent on the stack temperature. At the same time, Fig. 6 presents the specific power output vs. water-to-air ratio. Power output increases with increasing water-to-air ratio. However, it does not depend on the stack temperature, because the condensing heat exchangers are downstream of the economizer and preheater.

In Figure 5, the heat output does not cover all the stack temperature ranges in the cases when water-to-air ratio equals 0.25 or 0.30. There is a point where heat output is equal to 0, which means no heat is produced from the system. Thus, no condensing heat exchanger exists in the system. In fact, if the stack temperature is higher than this point, the process heat required for air humidification cannot be supplied by exhaust heat from the system. Therefore, no heat could be recovered by using condensing heat exchangers. The curves have a transition point which changes the characteristics of the process. This point is located at the interval between dew point temperatures of the flue gases from the gas turbine and biomass combustor as shown in Fig. 4. When the stack temperature is lower than this point, latent heat is recovered in the condensing heat exchangers. At

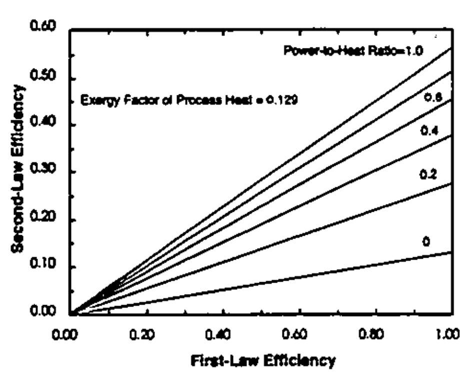

Fig. 3 Second-law efficiency vs. first-law efficiency

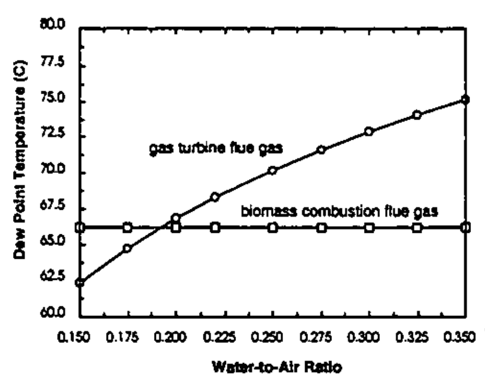

Fig. 4 Dew point temperature vs. water-to-air ratio

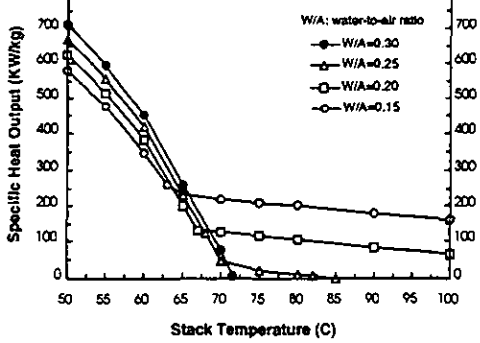

Fig. 5 Recovered heat in condensing heat exchangers vs. stack temperature

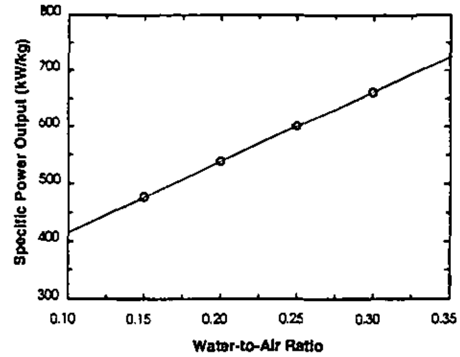

Fig. 6 Power output vs. water-to-air ratio

temperatures higher than the transition point, only sensible heat is recovered*

First-law and Second-law Efficiencies

Based on the definitions of the criteria given above, the total performance of the system is shown in Figures 7 and 8.

Fig. 7 gives the first-law efficiency of the system vs. stack temperature. The first-law efficiency increases with decreasing stack temperature. When the stack temperature is lower than the transition point, first-law efficiency is greatly increased with decreasing stack temperature. From a first-law viewpoint, flue gas condensation using condensing heat exchangers greatly improves system performance due to the latent heat recovery. For example, for the case with a water-to-air ratio of 0.20, when the stack temperature is at 50°C, the first-law efficiency is about 90 % compared to 47 % at a stack temperature of 100°C.

Since the first law does not take into account the "quality" of work and heat, Fig. 8 represents the second-law efficiency vs. stack temperature. Here, in order to compare the improvement due to flue gas condensation, the electrical efficiency is also plotted against the stack temperature. This diagram gives a better view of the improvement provided by using condensing heat exchangers. The second-law efficiency increases when a condensing heat exchanger is used in the system, especially when the stack temperature is lower than the transition point. The improvement in the second law efficiency is about 5 percentage points when the stack temperature is lowered from 100°C to 50°C. With a water-to-air ratio of 0.20, for example, the second-law efficiency is increased from 42% to 48% as the stack temperature decreases from 100°C to 50°C. This can be compared to the first-law efficiency which increases from 47% to 90%.

When the water-to-air ratio is higher, condensing heat exchangers are more important for system improvement in externally fired evaporative gas turbine systems. Fig. 8 shows that, if latent heat can be recovered by flue gas condensation, the higher the water-to-air ratio, the higher the second-law efficiency.

With water-to-air ratios of 0.25 and 0.30, there is a point where the second-law efficiency and electrical efficiency are the same. This point corresponds to the case where there is only power production from the system, as discussed above.

The results show the benefits of using a condensing heat exchanger in the evaporative gas turbines. The first is the possibility to lower stack temperature, for example, 50°C, which is impossible to do in an evaporative gas turbine without a condensing heat exchanger due to the limitation of stack temperature. Thus more heat can be recovered by this approach. The second is the opportunity to achieve overall reduction of pollution emission because of the simultaneous gas cleaning with a condensing heat exchanger.

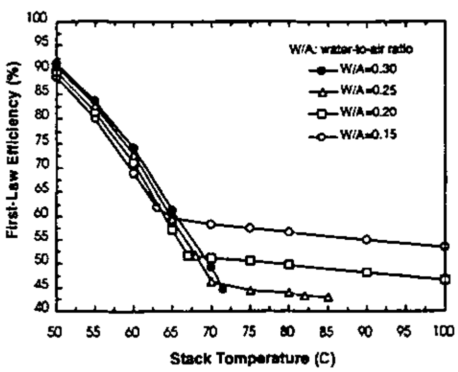

Fig. 7 First-law efficiency vs. stack temperature

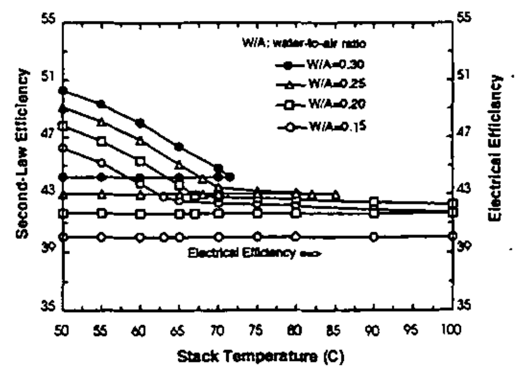

Fig. 8 Second-law efficiency and electrical efficiency vs. stack temperature

*In this paper, the heat exchanger is still called a condensing heat exchanger for simplicity, although, in this case, no condensation is occurring.

Other Considerations

Water recovery is one of the important issues for evaporative gas turbine cycles. In applications where makeup water is scarce or expensive, it may be economically justifiable to recover moisture from the stack gas. Flue gas condensation is one possibility for recovering moisture from flue gases. However, the recovered water would need to be treated prior to reuse in the cycle. How to treat the water requires further study.

The pollutants make the condensate corrosive. Therefore, resistant materials are required in the construction of the flue gas condensation system. Stainless and acid-resistant steels are used, as well as other, more sophisticated materials, such as Corten, Hastelloy, glass fiber reinforced polyester, graphite, Teflon, and enamel.

CONCLUSIONS

The following conclusions can be drawn from this study:

- Using a condensing heat exchanger with an externally fired evaporative gas turbine can improve system performance by recovering both sensible and latent heat from the flue gases since it allows a lower stack temperature than the system without condensing heat exchanger.
- The dew point temperature of the flue gas is one of the key parameters to be considered for a system with flue gas condensation. In a biomass externally fired evaporative gas turbine, the dew point temperatures of the flue gases from both gas turbine and biomass combustion are higher than those in the gas turbine without water injection. Thus, using flue gas condensation technology in this system is of importance for system improvement.
- The first-law efficiency can be greatly improved when a condensing heat exchanger is used in an externally fired evaporative gas turbine cycle. The second-law efficiency provides a rational tool for system analysis. Simulation results show that the improvement is about 5 percentage points based on second-law efficiency when introducing a condensing heat exchanger to decrease the stack temperature from 100°C to 50°C.
- A condensing heat exchanger provides the possibility of recovering make-up water in an evaporative gas turbine. The recovered water would need treatment prior to reuse in the cycle.

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APPENDIX: SYSTEM INPUTS FOR THE EXTERNALLY FIRED EVAPORATIVE GAS TURBINE WITH A CONDENSING HEAT EXCHANGER

Inputs for gas turbine subsystem

COMPRESSOR	
Inlet pressure loss	0.01
Pressure ratio	13.6
TOPPING COMBUSTOR	
Combustion efficiency	0.99
Combustor pressure loss (%)	3
Outlet Temperature (°C)	1120
Isotropic efficiency	0.88
TURBINE	
Isotropic efficiency	0.90
Turbine diffuser pressure loss (%)	1
GT mechanical efficiency	0.98
Generator efficiency	0.99

Inputs for heat recovery subsystem

INTERCOOLER	
Cold side pressure loss (%)	2
Hot side pressure loss (%)	1
Pinch or approach point temp. diff. (°C)	10
AFTERCOOLER	
Cold side pressure lossm (%)	2
Hot side pressure loss (%)	1
Pinch or approach point temp. diff. (°C)	10
HUMIDIFICATION TOWER	
Pressure loss (%)	4
Minimum Temp. difference of outlet water to wet-bulb point (°C)	10
Outlet air state	saturated
RECUPERATOR	
Pressure loss cold side (%)	3
Pressure loss hot side (%)	3
Approach or pinch point temp. diff. (°C)	30
COMBUSTION AIR PREHAETER	
Pressure loss cold side (%)	3
Pressure loss hot side (%)	3
Maximum air temperature (°C)	350
Pinch or approach point temp. diff. (°C)	30
Inlet air temperature (°C)	15
Relative humidity (%)	60
Pressure (bar)	1.0132
ECONOMIZERS	
Pressure loss cold side (%)	2
Pressure loss hot side (%)	1
Approach or pinch point temp. diff. (°C)	10
STACK	
Minimum stack temperature (°C)	100
Stack pressure (bar)	1.0132

Inputs for the biomass combustion subsystems

BIOMASS COMBUSTOR	
Excess air ratio	1.30
HIGH TEMPERATURE HEAT EXCHANGER	
outlet humid air temperature (°C)	900
Pressure loss hot side (%)	3
Pressure loss cold side (%)	3
Approach or pinch point temp. diff. (°C)	50

Inputs for all streams to the total system

NATURAL GAS		
Composition by mole fraction		
CH4	0.90982	
C2H6	0.04730	
C3H8	0.01732	
C4H10	0.01456	
N2	0.00598	
CO2	0.00502	
Temperature (°C)		10
Pressure (bar)		12
Lower Heating Value (MJ/kg)		48.4
SOLID BIOMASS		
Composition		
Moisture content (w)	50%	
Other elements (mass fraction)		
C	0.50	
O	0.44	
H	0.06	
Lower heating value (MJ/kg)		8.25
DISTRICT HEATING SYSTEM		
Supplied water temperature (°C)		70
Returned water temperature (°C)		40