HOW THE WATER SECTION IN THE HAT CYCLE CAN BE AN EFFICIENT POWER OUTPUT MODULATOR

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ABSTRACT

After a long period during which theoretical analyses were presented on evaporative gas turbine cycles, such as the HAT (Humid Air Turbine) and CHAT (Cascaded Humid Air Turbine), the first experimental plant with water recovery is currently under construction at the Lund Institute of Technology in Sweden. The pilot plant is due to start in evaporative mode in May 1998, and this represents the first step for the validation of the humid air turbine concept.

One of the main points of interest is the power modulation which should be possible controlling the evaporated water flow rate. If the whole compressed air flow rate is introduced into the evaporator the possibilities to vary its water content are scarce if the temperatures in the recuperator are not changed. A solution to this problem has been patented by Vattenfall AB, and consists in bypassing a fraction of the air entering the evaporator directly into the recuperator.

In this paper a detailed study of the different evaporation modes is presented from the point of view of both the first and second law analysis. The thermodynamic analysis will also be compared with the operational flexibility that the by-pass solution offers. Applications to some commercial turbines, which are most suited to use in HAT cycle mode, will also be presented.

NOMENCLATURE

- c_p specific heat at constant pressure
- m flow rate (kg/s)
- P pressure (Pa)
- St Stanton number
- T temperature (K)
- TIT turbine inlet temperature (K)
- β pressure ratio
- ε_h heat exchange efficiency
- Ω surface

Subscripts

- b blade
- c colant
- g hot gas
- out outlet
- 1 low pressure compressor
- 2 high pressure compressor

INTRODUCTION

Starting from the first proposed configuration (Rao and Joiner, 1990) the Humid Air Turbine cycle has shown very interesting characteristics for its performance, environmental impact and economic feasibility (Chiesa, et al., 1994, Day and Rao, 1993, Klara, et al., 1996, Lindgren, et al., 1992, Rosén, et al. 1994). In recent years, different solution have been proposed, either for increasing cycle performance (Stecco et al., 1993a-b) or for reducing some cycle drawbacks such as water consumption and low stack temperatures (Bidini et al., 1996, Desideri and Di Maria, 1997).

One possible advantage of the HAT cycle concerns the possibility of power modulation, at constant speed, by changing the injected steam flow rate. This was successfully implemented in conventional steam injected gas turbines (Brown and Cohn, 1981; Tuzson, 1992; Rice, 1993a-b) and in the Cheng cycle (Saad and Cheng, 1992). However, changing water flow rate in the HAT cycle is more complicated because of the water circuit configuration and its behavior.

In the conventional steam injected cycles and in the Cheng cycle, steam is generated in heat recovery boilers, which are surface heat exchangers, and then injected in the main flow, before or inside the combustion chamber. Steam flow rate depends on exhaust gas characteristics and on steam pressure, maximum temperature and pinch-point temperature difference. It is therefore quite easy to control the injected steam flow rate into the gas turbine.

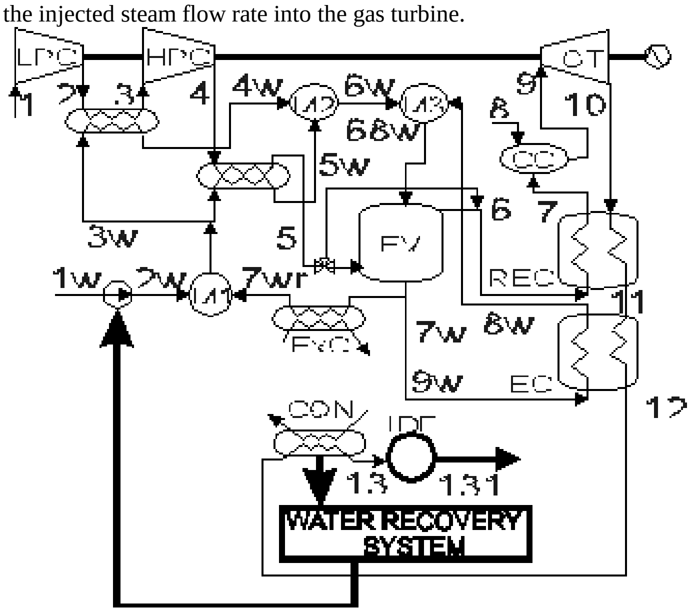

Figure 1: Schematic of the HAT cycle studied in this paper. Legend: LPC: low pressure compressor, HPC: high pressure compressor, EV: evaporator/saturator, EC: economizer, REC: recuperator, GT: turbine, CC: combustor, ExC: external cooler, CON: condenser, IDF: induced draft fan

In the HAT cycle, the introduction of water into the airflow is made in the evaporator that is a direct contact heat exchanger, where both heat and mass are transferred. Water content at the evaporator outlet only depends on air pressure and temperature that are the two properties, which determine saturation conditions for humid air. It is not possible to have a relative humidity higher than 1.0, but it could be possible to have it lower, then reducing water content in the airflow. To this aim, Vattenfall AB patented an evaporator inlet air bypass (Fig. 1), which has been studied in this paper.

CYCLE COMPONENTS

Figure 1 shows the HAT cycle configuration studied in this paper. This is the one described by Desideri and Di Maria (1997) with the addition of the evaporator bypass patented by Vattenfall AB. The ambient air is compressed in two compressors (LPC and HPC), which are inter (IC) and after (AC) cooled. The compressed cooled air enters the evaporator-saturator (EV), where air mass flow rate is increased by the water evaporation, then the recuperator (REC), where saturated air is heated, and then the combustion chamber (CC). After the blade-cooled turbine, the gas transfers heat in the recuperator and in the economizer (EC), which heats up part of the water that is recirculated from the EV. At the economizer outlet there is a condenser-dehumidifier (CON) to recover water and heat. Problems at the stack, due to the low flue gas temperature, are avoided by the introduction of an induced draft fan (IDF).

Cooling water flowing through the IC and the AC is mixed with the fraction coming out of the EC and injected at the top of the EV. Part of the water, which is not evaporated, is circulated into the EC while the rest flows through the IC and AC, after being cooled in the external cooler ExC. Evaporated water make up is introduced before the IC.

The air is humidified in a direct contact heat exchanger (EV) where the cold compressed air is heated and humidified by the hot water, increasing both its temperature and water content. In this process the air temperature is increased by mass and heat transfer between water and air. Furthermore, the water evaporates at rising pressure: i.e. at each step the water saturation pressure is at the mixture temperature, reducing the exergy loss during the heat transfer (Gallo, et al., 1995).

The basic configuration of the HAT cycle was modified with a bypass valve that allows reducing the compressed air fraction that enters the evaporator. The bleeded fraction is mixed with the humidified water at evaporator outlet before entering the recuperator. It is then possible to control the water content in the air within a wide range.

Table 1: Main simulation parameters

PARAMETER	VALUE
Pressure loss at compressor inlet	1500 (Pa)
Pressure loss at turbine outlet	1000 (Pa)
β_1/β_2	0.8
Polytropic efficiency of LP compressor	84.0 (%)
Polytropic efficiency of HP compressor	84.0 (%)
Polytropic efficiency of gas turbine	85.0 (%)
Adiabatic efficiency of pumps	88.0 (%)
Mechanical efficiency	99.0 (%)
Combustion efficiency	99.0 (%)
Electric generator efficiency	99.0 (%)
Polytropic efficiency of IDF	92.0 (%)
Minimum temperature difference IC inlet	20 (K)
Minimum temperature difference IC outlet	20 (K)
Minimum temperature difference AC inlet	20 (K)
Minimum temperature difference AC outlet	20 (K)
Temperature difference EV outlet	48 (K)
Pinch-point temperature difference EV	5 (K)
Minimum temperature difference REC inlet	30 (K)
Minimum temperature difference REC outlet	30 (K)
Minimum temperature difference EC inlet	20 (K)
Minimum temperature difference EC outlet	20 (K)
Pressure loss IC	2 (%)
Pressure loss AC	2 (%)
Pressure loss REC	1 (%)
Pressure loss combustion chamber	1 (%)
Pressure loss EC	1 (%)
Pressure loss condenser	1000 (Pa)
Condenser outlet temperature	323 (K)
IDF pressure ratio	1.0048

MAIN ASSUMPTION AND SIMULATION PARAMETERS

All the cycle components are simulated with validated codes developed for this purpose. It is also possible to set the ambient air conditions (P=101325 Pa, T=288 K and relative humidity=60 %) and the fuel composition among 16 different components. In the evaporation process the water and steam dew characteristics are utilized. The simulation code uses a blade cooling model developed by Facchini (1990) and the user can set 4 different points from which the cooling air is bled (i.e. 4, 5, 6, 7 of fig. 1)

The main flow splitting is simulated with no pressure and heat loss. The mixing process between the humid air and the bled dry cooling air is calculated by means of a mass and heat balance which neglects pressure and heat losses. Otherwise, if the blade cooling fluid is humid air, the user can chose to use humid air before or after the mixing point. All the other parameters for the calculation of the cycle are selected according to common values for commercial machines (Table 1).

BLADE COOLING MODEL

The high turbine inlet temperatures achieved in modern gas turbines are possible only because a large fraction of compressed air is used to cool the blades. Thermodynamic analysis has to take into account the effect of cooling on cycle performance.

The model used in the calculations presented in this work consists in a heat and mass balance between the coolant and hot gasses until the temperature is lower than the maximum allowable for the blades. The ratio between the coolant and hot gasses flow rate is calculated with the following equation:

$$\frac{m_c}{m_g} = \frac{c_{p,g}}{c_{p,c}} St \frac{\Omega_b}{\Omega_g} \frac{1}{\varepsilon_h} \left(\frac{T_g - T_b}{T_b - T_c} \right) \qquad (1)$$

Where the c_p ratios and temperatures define the thermodynamic characteristics of the two fluids and Ω_b/Ω_g (the blade to cross section vane ratio) defines the turbine geometry.

The heat exchange effectiveness is defined by:

$$\varepsilon_h = \frac{T_{out,c} - T_c}{T_b - T_c} \qquad (2)$$

Equation (2) characterizes the blade cooling technology and ranges from 0.25 for convection cooling to about 0.65 for film cooling. Once the coolant flow rate is calculated, it is mixed with the main flow and pressure loss of the mixing process are considered.

Table 2: Blade cooling model parameters

PARAMETER	VALUE
St_g	0.005
Ω_b/Ω_g	4
ε_h	0.45
T_b	1073 K
Row ΔT	150 K
Mixing pressure loss	1 %

THERMODYNAMIC ANALYSIS

The thermodynamic analysis was done by letting the bled air fraction vary for different TIT and β values. For each TIT and β efficiency and power output have been evaluated. The bleed point for blade cooling was set for all cases to 6 (Fig. 1), after the mixing with bypass air. At this point the cooling fluid has the highest thermal capacity, because of the high water content, and a low temperature due to mixing with the bypass air.

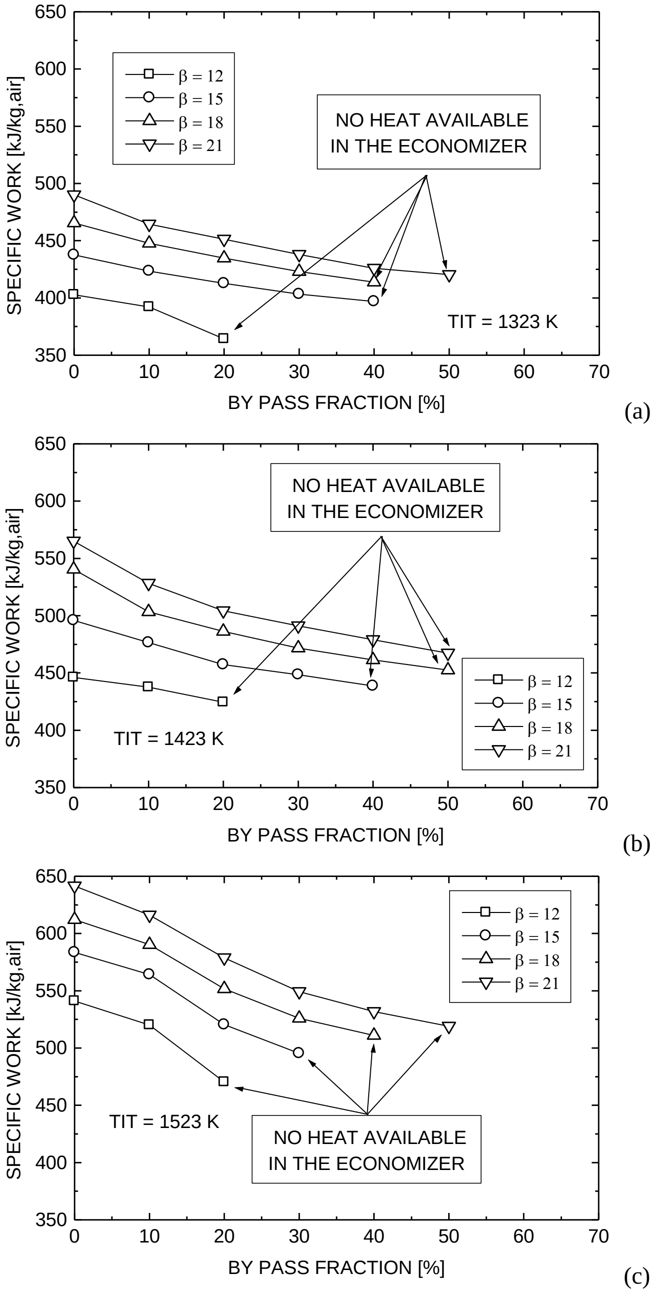

Figure 2: Specific work Vs. bypass flow rate fraction

It is possible to note that the cycle power output decreases as the air bypass fraction increases at each TIT and pressure ratio (Fig. 2).

Considering as a reference case the operation without bypass air, the reduction in power output ranges from 10 to 20% when the bypass fraction changes from 0 to 50%.

The larger the bypass flow rate the lower is the thermal capacity of the exhaust gasses stream. This means that, as the bypass fraction rises, the recoverable heat at the turbine outlet diminishes and a lower amount of heat is available at both the recuperator and the economizer.

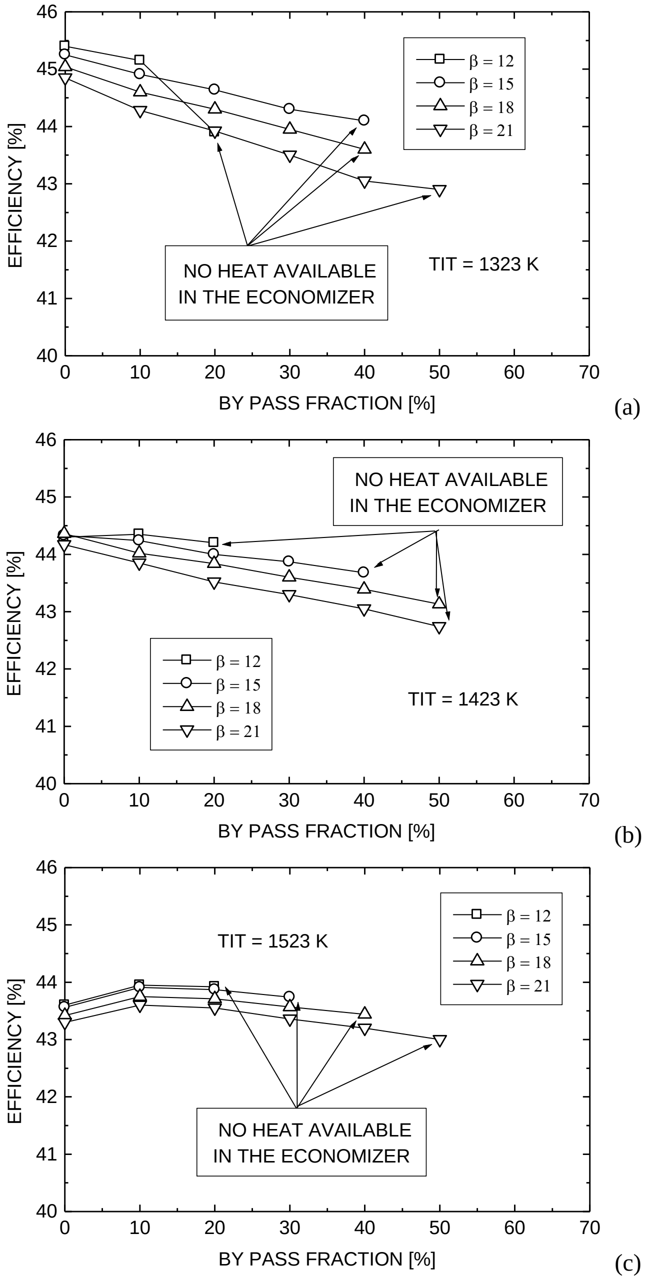

Figure 3: Efficiency Vs. bypass flow rate fraction

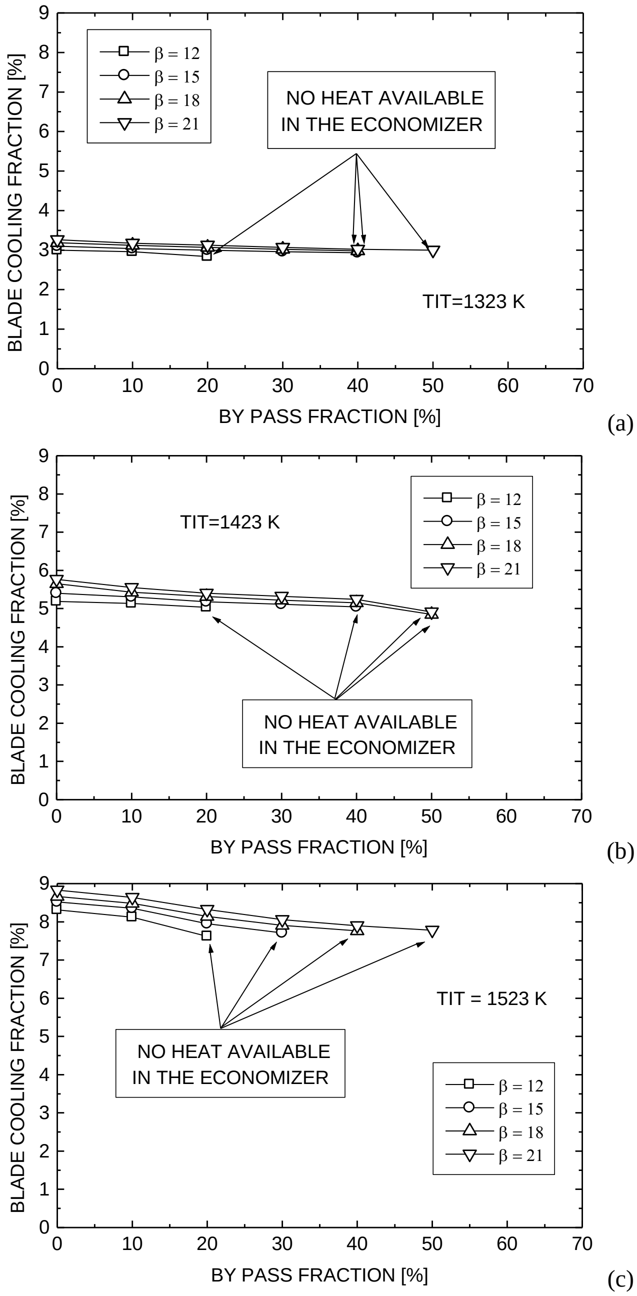

Figure 4: Blade cooling flow rate Vs. bypass flow rate fraction

If the bypass flow rate becomes too large, in some operating conditions the economizer will not recover any heat. This condition is indicated on each diagram and moves towards higher bypass fractions when pressure ratio rises. This is due to the higher amount of evaporated water that increases the main flow thermal capacity.

At TIT lower than 1523 K, the cycle efficiency decreases as the bypass fraction increases. When the TIT reaches 1523 K there is an increase in cycle efficiency for low bypass fractions. This means that the bypass technology can even be used for optimizing cycle efficiency. However, the best cycle performances are achieved at lower TIT (Fig. 3).

This can be explained by considering Figure 4. The increase in blade cooling flow rate is considerable when TIT is raised and this reduces the cycle efficiency. At constant bypass fraction the higher is the compression ratio the higher is the blade cooling flow rate (Fig. 4). This is true for each TIT and can be explained with the larger amount of heat recovered in the IC and AC that increases the temperature of the water coming out from the evaporator, after the humidification process.

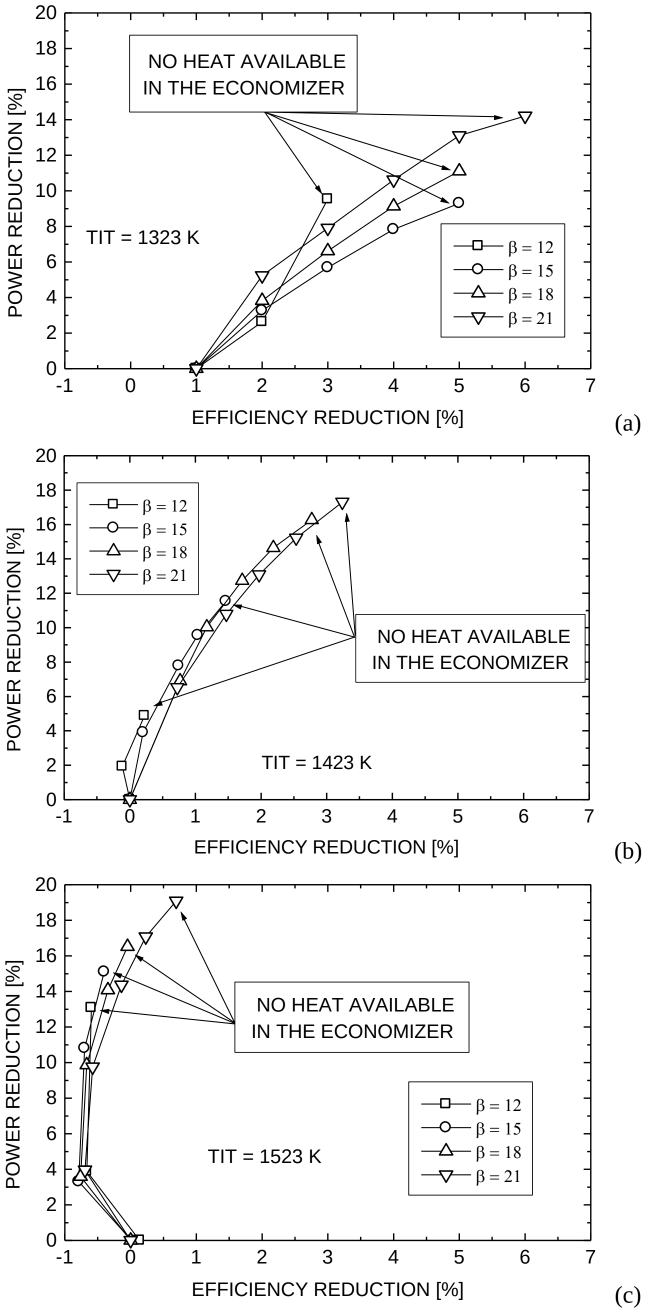

Figure 5: Power output reduction Vs. efficiency

The humid air temperature increase is higher than the increase of the of the coolant specific heat at constant pressure and so from Eq. (1) the coolant flow rate increases. Furthermore, at constant TIT and pressure ratio there is a slight decrease in coolant flow rate as the bypass fraction increases. This is due to the lower temperature of the by pass air, at AC outlet, that reduces the temperature of the coolant

At lower TIT, reductions in power output are accompanied by a reduction in efficiency (Fig. 5). On the contrary, when the TIT reaches 1525 K, the decrease in power output, that is also evident in Figure 1, is combined with and increase in efficiency. Figure 5 also shows the maximum variations in power output that can be obtained by increasing the bypass flow rate. The range of power output control is e considerable, because TIT is left unchanged. Moreover, such a high power reduction is achieved with a small penalty for the efficiency.

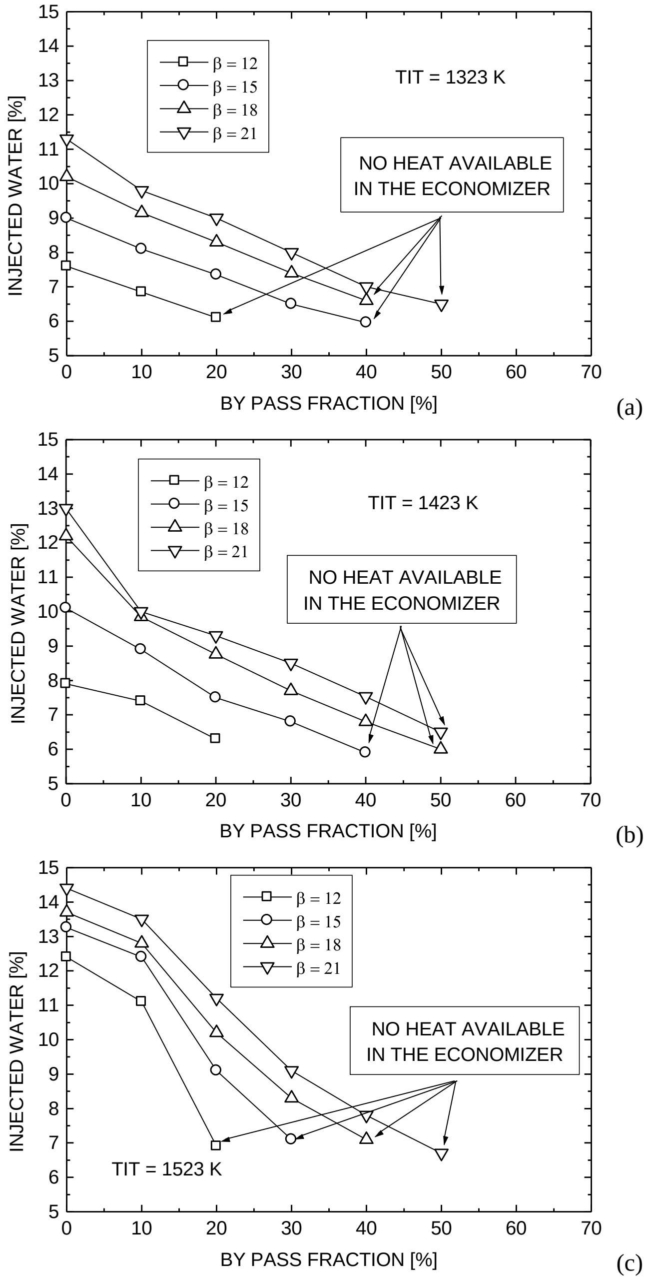

Figure 6: Water content in humid air Vs. bypass flow rate fraction

As it is common in the HAT cycle, the water content in humid air increases with both the TIT and the cycle pressure ratio, because the larger amount of recovered heat allows to achieve higher evaporator outlet temperatures. Using the bypass valve it is possible to reduce the evaporated water up to 50 % (Fig. 6), before reaching the operation limit of the economizer. However, increasing the bypass flow rate reduces the contribution of combustion water to the total water content, and the ratio between the recovered and injected water is lower accordingly (Fig. 7). At bypass fractions of 50% it is no more possible to recover the water from the exhaust gasses.

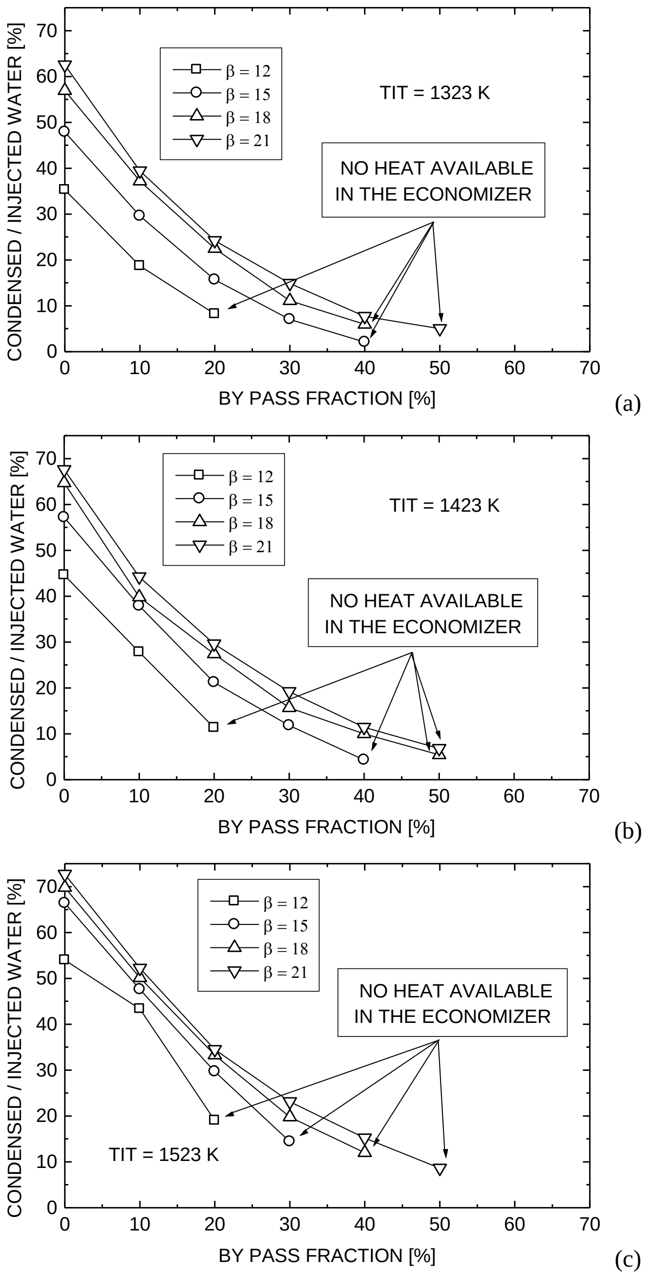

Figure 7: Ratio of condensed over injected water Vs. bypass flow rate fraction

It must be noted that the condenser becomes less and less important if the bypass fraction is kept low during most of the operation.

ECONOMIC ANALYSIS

One of the main questions that are generally asked about the HAT cycle concerns the economic feasibility. This paper presents a cost analysis which deals with the capital and operating costs of a HAT cycle, a gas turbine in simple cycle configuration and a combined cycle. A medium size gas turbine of 43 MW, such as the ABB GTX100, was considered, as the basic gas turbine for this analysis. Thus size is large enough to be combined with a steam cycle and reach the highest efficiency levels for the latest technology. At the same time a 40 MW gas turbine can be modified to operate in HAT cycle mode without requiring too large make-up water flow rates.

The additional costs to modify a gas turbine and make it operate as a HAT, are the introduction of the inter and after coolers, the evaporator, the recuperator and the economizer. The cycle configuration presented in this paper also comprises the water recovery system with the condenser and the IDF, which will also be considered in the cost analysis.

The cost of the cycle components were determined by the cost of similar equipment, which can be found on the market. The cost of similar components was assumed the same. For example the cost of the compressor’s coolers and the economizer, which are gas-water heat exchangers with comparable temperature and pressure levels, was considered the same.

The cost of both the gas turbine and the combined cycle was assumed as 500 $/kW, which is a standard level for medium size heavy duty gas turbines. However the change from simple cycle to HAT cycle requires a high swallowing capacity for the turbine, because the water content introduced between the compressor and the combustor can be as high as 15% of the inlet flow rate. Since most of the turbines have a design flow rate close to choking conditions, it is practically impossible to add such a large amount of water without resizing the turbine’s flow passages, and particularly the first nozzle.

One solution to this problem is that proposed by Nakhamkin et al (1995, 1997) for the CHAT cycle, where they couple a compressor of a smaller gas turbine with a turbine of a larger size. This can considerably reduce the development costs of the turbine for the HAT cycle. In this case the overall cost of the turbine is assumed as an average of the costs of two gas turbines of different size.

The following costs were assumed for the additional component of the HAT cycle:

Medium temperature heat exchangers (IC, AC, EC): 100 $/m^2
High temperature heat exchangers (REC): 180 $/m^2
Cooling towers with heat exchangers (ExC, Cond): 18000 $/MW
Evaporator-Saturator: 20000 $/MW
Piping, pumps and IDF: 5% of additional costs

Detailed capital costs of the additional items are shown in Table 3, for a total cost of 10,830,000 $.

Since the turbine and the combustors for the HAT cycle cannot be the same as in the base gas turbine, it is possible either to redesign completely both components or to use similar parts from a large turbine of the same manufacturer. The market for the HAT cycle may be promising, but the design of new components is very expensive for manufacturers. Therefore, the capital cost for the turbine to be used in HAT cycle configuration was assumed as the average of the cost of the base gas turbine and of a larger size gas turbine. The result of this operation gives 29,450,000 $.

Table 3: Additional costs for a HAT cycle

Component	Cost ($)
Intercooler	1,700,000
Aftercooler	1,300,000
Economizer	450,000
Recuperator	4,320,000
External Cooler + Condenser	700,000
Evaporator-saturator	2,360,000
Auxiliaries	540,000

The overall costs for a HAT cycle is then 40,870,000 $ that divided by the power output (74.8 MW) gives 546 $/kW. The additional costs represent 38.7 % of the gas turbine costs, but the influence on the cost per kW is lower than 10 %, with respect to the simple gas turbine and the combined cycle. It must be said that such low costs for the gas turbine and the combined cycle are determined by their large diffusion as power plants and the competition of several different manufacturers.

Table 4 shows the capital costs, the power output and the efficiency of the simple cycle gas turbine, a gas-steam combined cycle and the HAT cycle. The efficiency was corrected on the natural gas higher heating value basis, because the HAT cycle recovers water from condensation before the stack.

Table 4: Comparison of the capital costs

Cycle	Power output (MW)	Efficiency (% HHV)	Cost (M$)
Gas turbine	43	33.5	21.5
Combined cycle	62	49	31.0
HAT	75	45	40.9

For the operating and maintenance costs the following assumptions were made:

Cost of natural gas: 0.10 $/Stm^3
HHV of natural gas: 38100 kJ/Stm^3
Selling price for electricity: 0.05 $/kWh
Maintenance costs: Simple cycle: 2.7 mills$/kWh
Combined cycle 4.0 mills$/kWh
HAT cycle 4.0 mills$/kWh
Water treatment costs (HAT): 1.7 mills$/kWh

The cost of maintenance for the HAT cycle was considered equivalent to that of a combined cycle, because of the large number of heat exchangers and the presence of a complex water circuit.

Net yearly cash flow for the operating and maintenance cost are shown in table 5. It is interesting to note that the O&M costs represent the highest percentage of the capital costs for the combined cycle, whereas they are the lowest for the simple cycle.

From the above results the HAT cycle presents some disadvantages in comparison with the combined cycle. Part of the increment of capital costs can be reduced if additional components for the HAT cycle are built on a larger scale. This is particularly important for the recuperator and the evaporator that have the highest cost.

It must also be noted that the comparison was made with a gas turbine of the latest generation with a significant efficiency both in simple cycle and in combined cycle mode. The comparison with smaller gas turbines, having a lower efficiency and for which the combined cycle is not convenient (the steam section would be too small), would have raised the HAT cycle to a better rank.

Table 5: O&M costs

Cycle	Yearly O&M net cash flow (M$/year)	O&M/Capital costs (%)
Gas turbine	6.570	30.5
Combined cycle	12.620	40.7
HAT	13.954	34.1

CONCLUSIONS

Usually, when power plants operate on off design conditions, the efficiency is highly reduced. In particular gas turbine power plants are very interested in off design efficiency reduction. Many different solution have been proposed but it is always required an high increase in plant costs with no positive effects on design performances. Anyway, for keeping the efficiency high it is important to keep the TIT constant. In the steam injected cycle or the Cheng cycle this is possible by varying the injected water flow rate. This means a quite constant TIT value and pressure ratio. For the HAT cycle the evaporator by pass technology seems to be a good solution for keeping the cycle efficiency constant even when large reduction in power output are required. In some cases it is also possible to observe an increase in cycle efficiency.

The analysis of fixed and variable costs has shown that the HAT cycle has some advantages on the simple cycle gas turbine, but is still suffers from the low capital costs of the combined cycles, which are widely used and for which there is a significant offer of sizes, types and performance.

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