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HUMID AIR GAS TURBINE CYCLE: A POSSIBLE OPTIMIZATION

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ABSTRACT

The humid air turbine (HAT), patented by Fluor Daniel, is an innovative cycle which allows to obtain an increase in efficiency and power production. The modification proposed by DEF allows to optimise the plant when natural gas is injected in the combustion chamber.
Assuming a TIT (Temperature Inlet Turbine) at 1273 K and the cooling of recirculating water in the refrigerators, we studied the effects of the relative humidity and the compression ratio on the cycle's performances.
The aim of this paper is to suggest the parameters which allow to obtain high efficiency with high specific power, the possibility to modulate power production without a decrease in efficiency and low water consumption.

FOREWORD

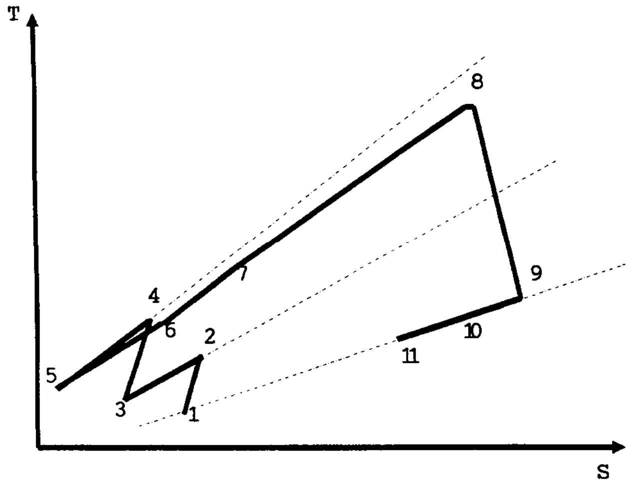
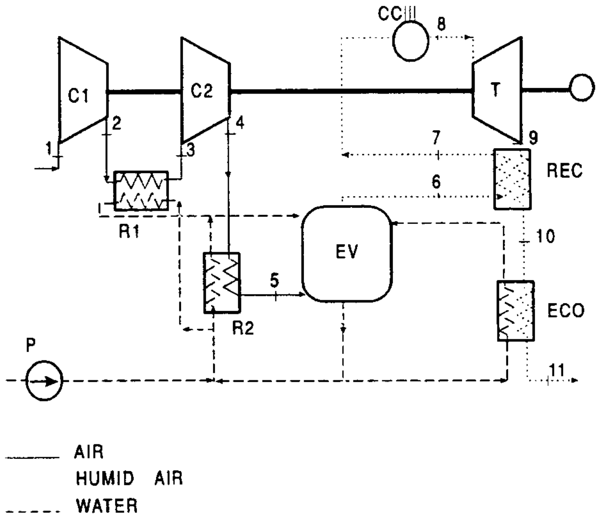


Fig.1: The HAT thermodynamic cycle



C1 = 1st compressor stage | *EV = evaporator*
C2 = 2nd compressor stage | *REC = recuperator*
R1 = intercooler | *CC = combustor*
R2 = aftercooler | *T = turbine*
P = water pump | *ECO = economizer*

Fig.2: HAT cycle: basic version

Fluor Daniel has recently patented a new gas turbine cycle with a mixing evaporator ahead of the combustion chamber thus feeding the turbine with large amounts of water. The system is complete with intercoolers and exhaust gas enthalpy recovery; the cycle has been named HAT (Humid Air Turbine) (Rao and Joiner, 1990) (Fig. 1-2). For details on this cycle please refer to Appendix 1, or to the references.
The Department of Energy Engineering of the University of Florence has been studying this new cycle, reviewing some potential modifications. The most promising one is that of

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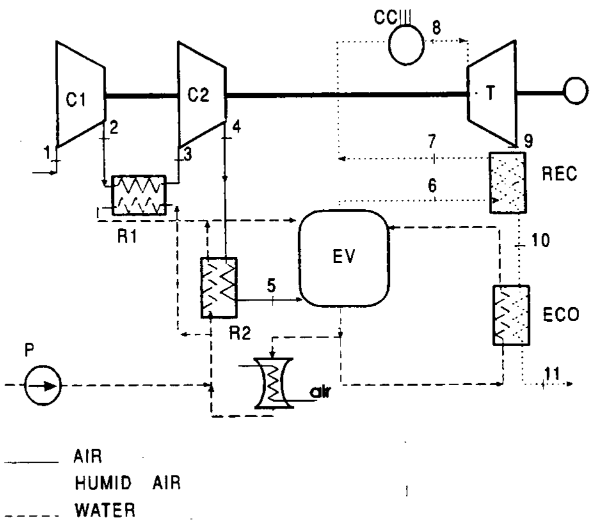


Fig.3: modified cycle:version II

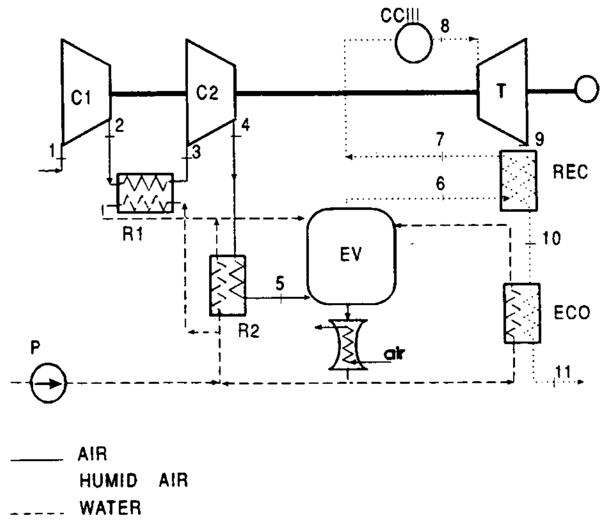


Fig.4: modified cycle:version III

cooling only the recirculating flow in the refrigerators with the possibly cooling the entire evaporator discharge when the water content is high (Fig. 3-4).
This solution, which differs from the Rao and Joiner proposal has some noteworthy thermodynamic and performance advantages. This article presents the how and why of these pluses.

NOMENCLATURE

- TIT = turbine inlet temperature (K)
- W = specific power output (MW/kg)
- WC = water consumption for producing 1 MW of electric power (kg/MW)
- x = water content (kg water/kg dry air)
- β = total compression ratio
- β_1 = compression ratio, 1st stage
- β_2 = compression ratio, 2nd stage
- ϕ = relative humidity (steam pressure/steam saturation pressure)
- η = thermodynamic efficiency.

DESCRIPTION OF THE CYCLE'S FEATURES

The results of the previous studies show that the most advantageous configuration is given by cooling only the recirculation flow; it has the best efficiency levels with high power output and low water consumption at low relative humidities. If ϕ increases it is worthwhile adopting the configuration with total refrigeration of the evaporator exit flow in order to decrease water consumption.

Since the switching from one type of system to the other is very easy, we can operate in the configuration II shown in Figure 3 with low ϕ levels, and in the Figure 4 configuration III with high ϕ to manage peak loads without increasing water consumption.
The water content of the mixture leaving the evaporator can be changed by varying flow trough the economizer; as flow decreases the heat available for evaporation and hence the water content also decrease.
The aim of this paper is to examine the parameters that would optimize the system, using the solution shown in Fig.3, and then to determine the β_2/β_1 ratio and relative humidity levels that would lead to the best results.

PERFORMANCE ANALYSIS

This analysis covers variation in cycle performance relating the β_2/β_1 ratio to different relative humidites at the evaporator outlet, with a set compression ratio in the first stage, namely β_1=2,3,4. The purpose of the analysis is to determine the parameters offering high cycle efficiency and specific power output combined with low water consumption per MW produced and high versatility of the system making it possible to handle high load variations without excessive efficiency reductions.

COMPRESSION RATIO β_1 = 4 (Fig.5-6)

Analyzing the characteristic efficiency curves, we can see, that with low second stage compression ratios, efficiency is very high; for β_2/β_1=0.5 efficiency ranges from 53% to 57% as humidity at the evaporator outlet varies between 20% and 50%.

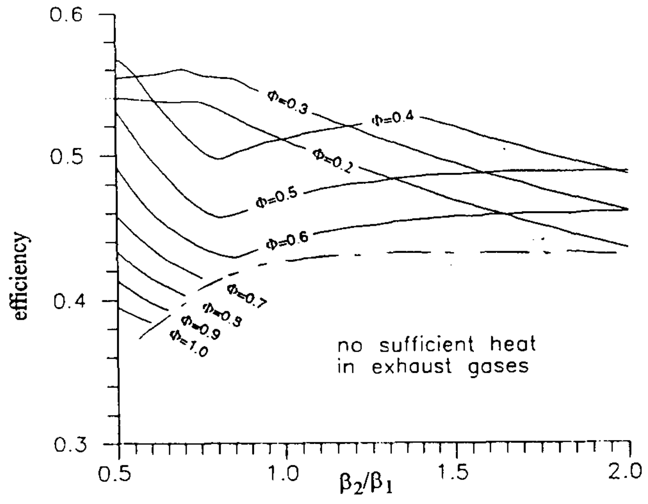


Fig. 5: efficiency for $\beta_1=4$ vs β_2/β_1

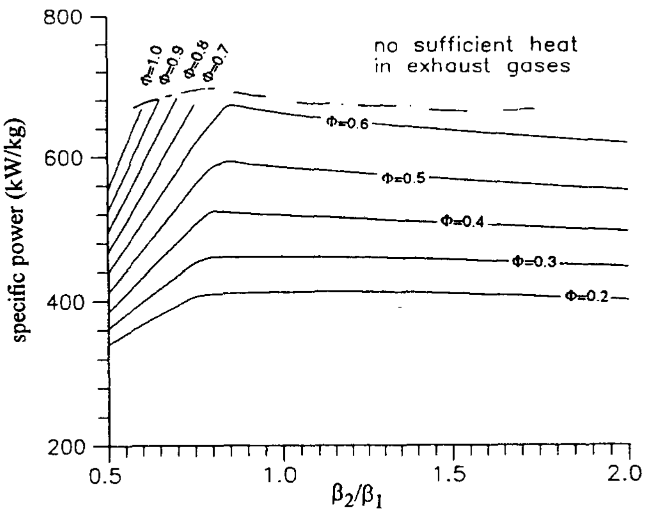


Fig. 6: specific power for $\beta_1=4$ vs β_2/β_1

By changing the mixture water content at the evaporator outlet power, output varies from 340 kW/kg to 410 kW/kg of evolving air, allowing easier modulation of power output without having to pay the price of markedly decreased efficiency.

The results of the WACY program (created by DEF to study cycle performance) show water consumption that ranges from 0.14 kg/s to 0.28 kg/s per MW produced with $\beta_2/\beta_1=0.5$. This corresponds to daily consumption between a minimum of 1210 m^3 and a maximum of 2420 m^3 for a 100 MW unit.

The water content obtained in the evaporator varies from 4.5% to 12% with ϕ between 0.2 and 0.5. This is a positive feature since there are no stability problems regarding the flame in the combustion chamber. The difference between the evolving flow in the turbine and the compressor is low enough to permit use of existing turbogas units without requiring extensive modifications.

Going on to higher second stage compression ratios we obtain better performance in terms of specific power output with equal water consumption. However, the possibility of making adjustments is lost since efficiency rapidly drops as the water content increases.

Therefore, for $\beta_1=4$ it is more advantageous to choose $\beta_2=2$ with a total compression ratio of 8.

Due to the mixture's high relative humidity, the system cannot operate as the stage second compression ratio increases since the exhaust gases are not hot enough to allow state change in the evaporator.

COMPRESSION RATIO $\beta_1=3$ (Figs. 7-8-9-10).

Maximum efficiency is obtained for $\beta_2/\beta_1=0.5$, reaching 58% with low water consumption per MW. On the other hand, there is low specific power output (less than 300 kW/kg) per kg/s of air evolving in the compressor, thus reducing the size of each unit. This is highly restrictive since gas turbine development for the future is currently directed toward large systems (greater than 100 MW) that cannot be achieved with low specific power output.

The optimum configuration appears to be $\beta_2/\beta_1=1$, where efficiency ranges between 54% and 56% with relative humidity levels between 20% and 40% at the evaporator outlet. The power output is between 360 kW/kg and 420 kW/kg permitting large system sizes and good power output regulation. As ϕ increases efficiency decreases rapidly and there is no advantage in working at higher values except in the case of otherwise supportable peak loads.

Water consumption varies from 0.15 to 0.27 kg/s per MW produced. The water content ranges from 5% to 11% eliminating problems in the combustion chamber. It is interesting to compare this system with the Cheng cycle (Saad and Cheng, 1992) which generally functions with water contents of up to 18%.

This configuration ($\beta_1=3$, $\beta_2=3$) appears more advantageous than the previous one ($\beta_1=4$, $\beta_2=2$) since the efficiency levels are comparable and power output levels even better.

COMPRESSION RATIO $\beta_1=2$ (Fig. 11-12)

In these cases the gain in efficiency of 1% is paid for by a sharp decrease in power output per unit of evolving mass of air. Using $\beta_2/\beta_1=4$ efficiency and power output are lower than with $\beta_1=4$, $\beta_2=2$, and therefore, this configuration is not favourable. According to this brief analysis, the best cycle would consist of two compression stages with a total ratio of 9 equally divided between the compressors. Some comments are needed to justify the efficiency curve pattern, a pattern that is similar in all the cases examined.

THERMODYNAMIC ANALYSIS OF THE RESULTS

Figures 13-16 show the graphs that help to understand the

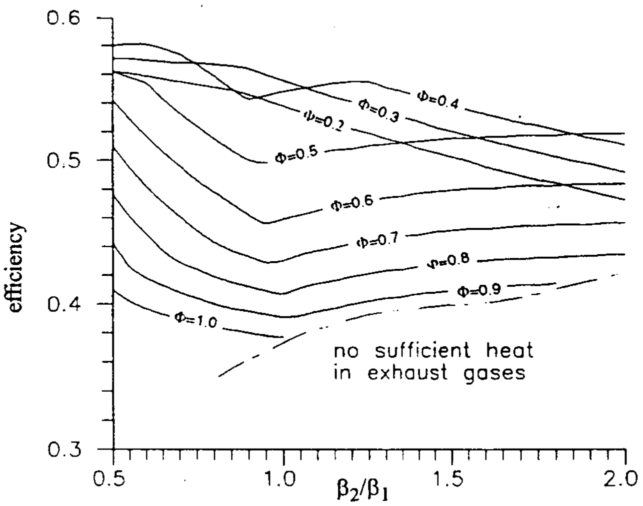


Fig. 7: efficiency for $\beta_1=3$ vs β_2/β_1

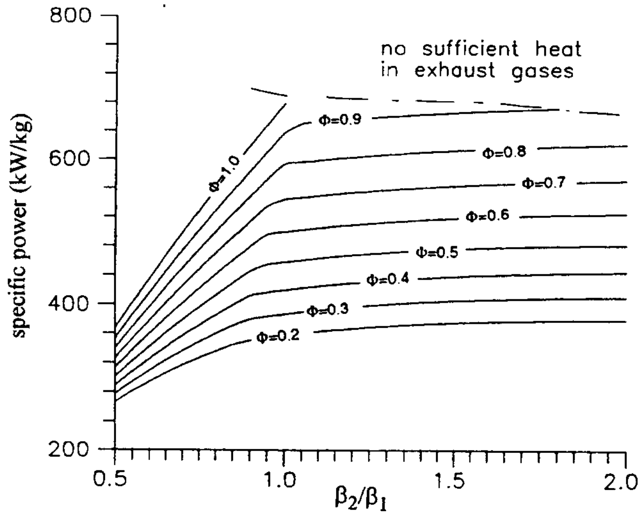
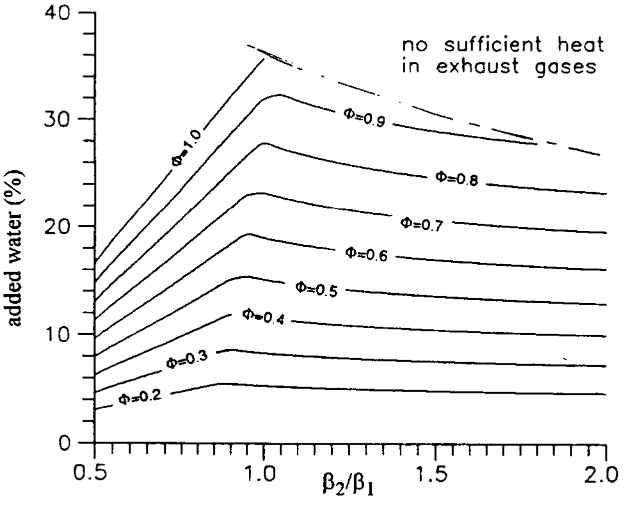


Fig. 8: specific power for $\beta_1=3$ vs β_2/β_1



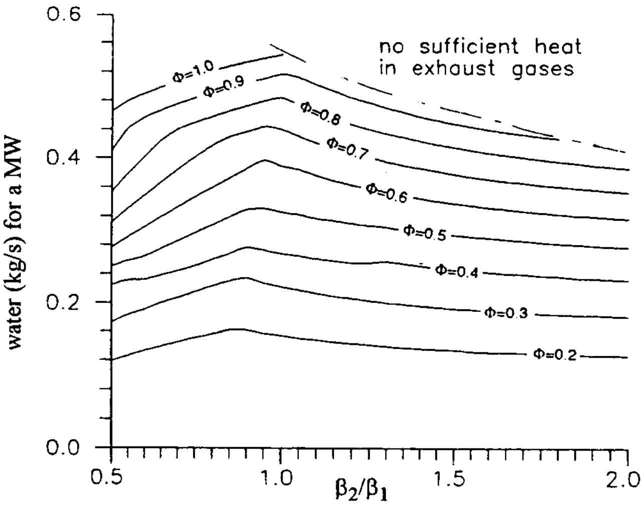


Fig. 10: water need for 1 MW for $\beta_1=3$ vs β_2/β_1

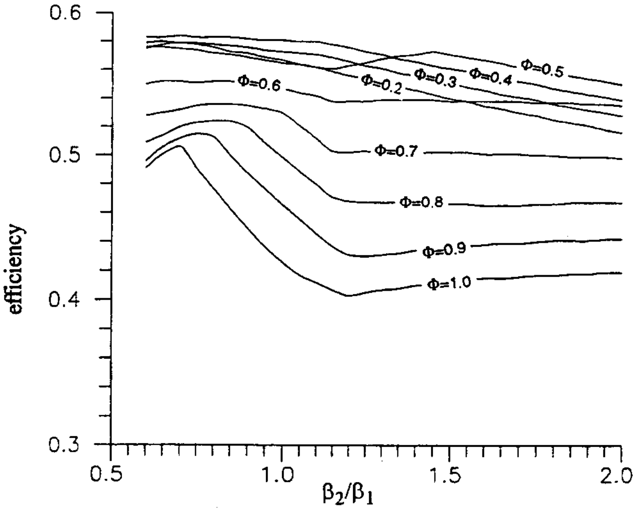
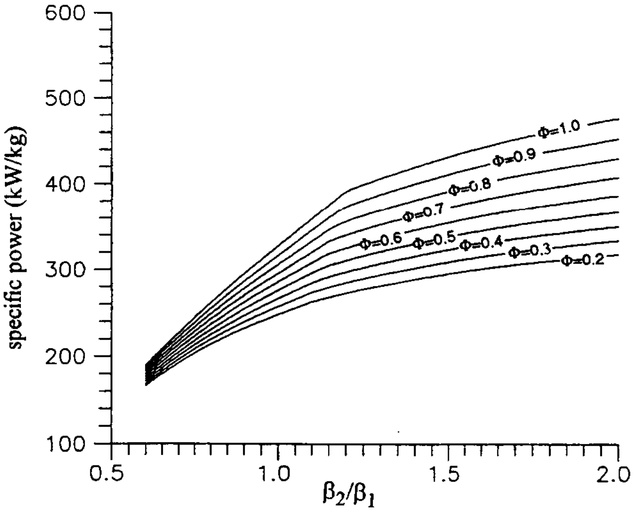


Fig. 11: efficiency for $\beta_1=2$ vs β_2/β_1



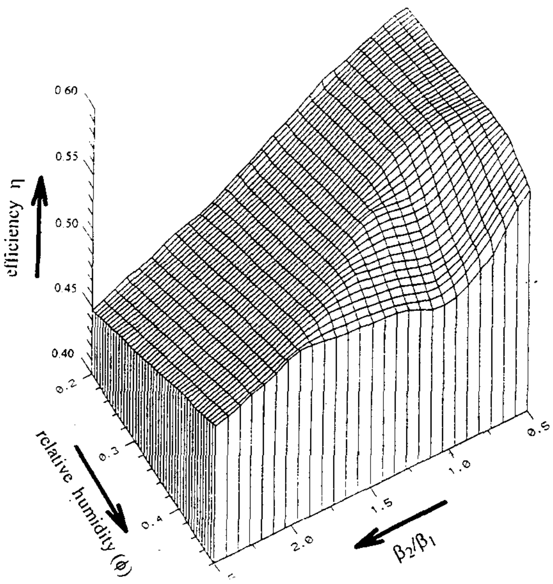


Fig. 13: three dimensional representation of efficiency, relative humidity and β_2/β_1

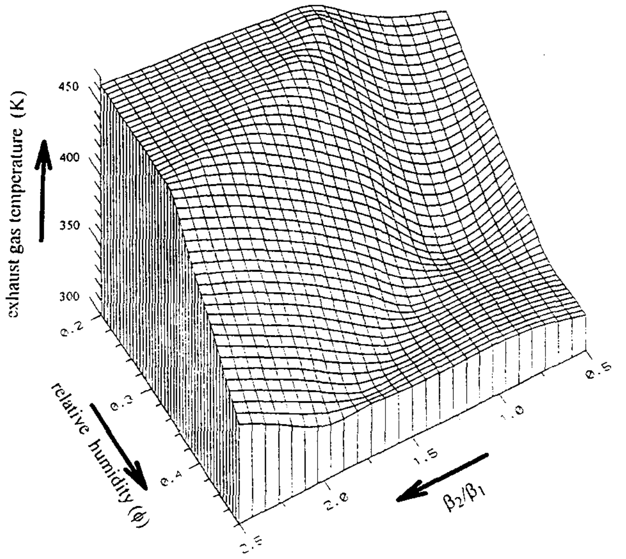


Fig. 15: three dimensional representation of exaust gas temperature, relative humidity and β_2/β_1

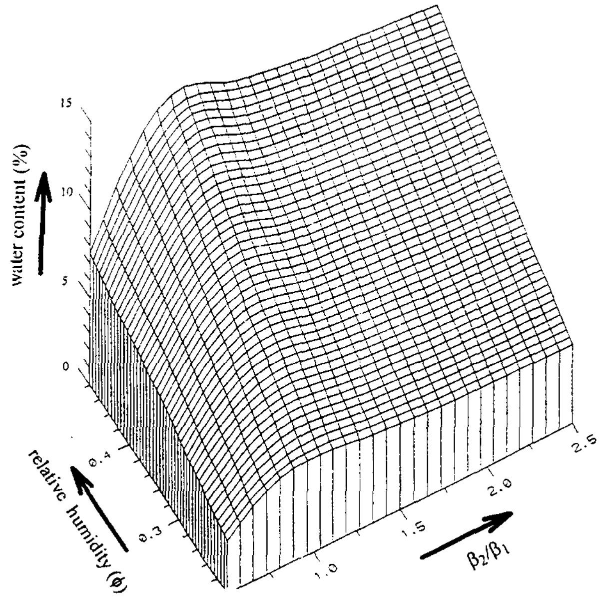


Fig. 14: three dimensional representation of water content, relative humidity and β_2/β_1

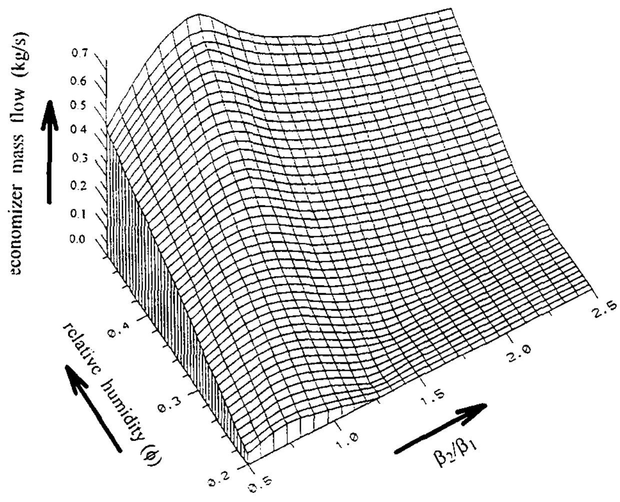


Fig. 16: three dimensional representation of economizer mass flow, relative humidity and β_2/β_1

efficiency curves. The x and z axis show the β_2/β_1 ratio, relative humidity ϕ, and show efficiency, mixture water content, exhaust gas temperature and water flow in the economizer, respectively.

One remarkable aspect is how efficiency decreases as the compression ratio rises, yielding the maximum values for low total compression ratios (always less than 10). This is easily explained: decreasing the total β, at the same maximum temperature means increasing the amount of heat that can be recovered from the exhaust gases for preheating the mixture before it enters the combustion chamber. Therefore, at low β values there is a greater recovery and hence more efficiency, while raising β increases power output while decreasing heat recovery.

Another important aspect shown on the graph is that efficiency for high compression ratios is greater when relative humidity is high, while the exact opposite holds true for low β values.

A review of this aspect is important because it will help understand the cycle's operation and its intrinsic limits.

At high β values, the turbine's exhaust gases have sufficient heat to allow the recuperator to operate at full capacity and heat the incoming mixture up to the turbine exhaust temperature and also to permit economizer operation. The economizer's heat share is set by the mass flow of water moving through it (the mass flow decreases as β rises since more heat is coming from the cooler and, moreover, the water content tends to decrease) and by the temperature difference between inlet and exhaust. The temperature of the exhaust gases is higher than the level set by the economizer heat exchanger and therefore a certain quantity of recoverable heat is discharged with the gases. As relativity humidity rises, the water content rises increasing economizer flow; the temperature of the exhaust gases is thus diminished, resulting in a greater recuperative effect and higher efficiency.

As relative humidity increases we can see a relative minimum for the efficiency curves when $\beta_2/\beta_1=1$. This minimum is somewhat difficult to understand immediately, but it reflects cycle operation and confirms the validity of the diagram adopted. When $\beta_2/\beta_1=1$ the water content is maximum and therefore the economizer's flow is also maximum since it must supply the heat required for evaporation.

Since the exhaust gas temperature is minimum as set by the temperature of the incoming water, part of the heat needed for evaporation is drawn from the "supply" for the recuperator which hence has less heat to supply to the mixture before it enters the combustion chamber. The temperature of the turbine discharge temperature (the maximum attainable in the event of complete operation), thus increasing the need for fuel and decreasing efficiency.

When exhaust gas temperature is high enough to allow full economizer and recuperator function, efficiency will once again decrease with β and rise with ϕ and the temperature of the exhaust gases, no longer restricted by the lower limit imposed by the temperature of the water entering the economizer, will once again rise freely.

To provide a better overview of the curves, the diagrams were draw in a limited field (between 0.3 and 0.5) of value for ϕ. The diversification of efficiency curves in relation to changes in the relative humidity of the mixture as it leaves the evaporator takes place at ϕ approximately equal to 0.36 as obtained from the program.

For $\phi<0.36$ the temperature of the exhaust gases is "free" without any horizontal areas showing the limit set by the evaporator discharge temperature. This would mean that the recuperator and economizer are functioning at full capacity and therefore the efficiency curve is generic, i.e. decreasing with β due to the effect of the greater distance of the isobars, and rising with ϕ due to the greater recovery archived on the exhaust gases.

For $\phi>0.36$ we can see that the exhaust temperature is restricted. In this case the maximum quantity of heat is recovered from the exhaust gases, but the recuperator can only take that amount of heat which is in excess of the economizer's requirements. The rest must be provided by the fuel, and hence efficiency decreases. The efficiency curve normalizes when the exhaust temperature tends to rise.

Noting this regular feature and observing that, even by pushing to higher ϕ values, the minimum always corresponds to the maximum water content, we can assume that there is a compression ratio for each value of ϕ that will result in a "normal" efficiency curve. When ϕ increases even at low β values, the recuperator is not able to operate at full capacity and efficiency is lower. Observing the exhaust temperature we can see that it is bound to the temperature of the water entering the economizer in all cases and is only "free" for the low ϕ values reviewed above.

For low relative humidity values and high compression ratios, the exhaust temperature becomes nearly constant. There is a thermodynamic justification for this phenomenon as well. As long as a certain flow is needed in the economizer for evaporation, the temperature of the exhaust gases is lower than the temperature leaving the recuperator; it increases as the flow decreases, a situation which occurs when the compression ratio increases.

When flow in the economizer is cancelled out (i.e. when all the heat required for evaporation comes from the refrigerators) the exhaust gas temperature is equal to the temperature at the recuperator outlet which is limited by the temperature of the mixture leaving the evaporator.

With the above operating hypotheses, this temperature rises gradually as the compression ratio increases and, therefore, even the temperature of the exhaust gases follows that pattern.

CONCLUSIONS

On the basis of the above assumptions and considerations, the optimum configuration for the HAT cycle in the version here proposed is the one shown in Fig. 3, i.e. with a dry cooling tower on the recirculating flow in the refrigerators. The optimum operating condition that permits high efficiency, considerable power output and limited water consumption is the following:

$\beta_1 = 3$
$\beta_2 = 3$
$\phi = 0.2 - 0.4$

which permits the following features:

$\eta = 54\% - 56\%$
$x = 5\% - 11\%$
$W = 340 - 420$ kW/kg
$WC = 0/15 - 0.27$ kg/s

and regulation is simplified because only the recirculating flow in the economizer has to be adjusted. The choice of the minimum point for high ϕ values is justified by the fact that considerable power output is obtained with high efficiency; the important factor is not to go beyond $\phi=0.5$: after that point efficiency falls below 50%.

This HAT cycle is no longer anhydrous. Using a 100 MW unit and an average consumption of 0.2 kg/s per MW, daily water consumption comes to 1730 m^3 which corresponds to water consumption for a town with a population of about 11.500 (using 150 litres per person). However, this is not excessive and the cycle appears very promising.

APPENDIX 1

The HAT cycle is an intercooled turbogas system with a mixing evaporator ahead of the combustion chamber and exhaust gas heat recovery down to very low temperature levels (Bettagli 1992)

The most important advantages of this cycle are the following:

- increased efficiency due to the heat recovery, intercooled compression and variable temperature evaporation;
- increase in power output due to the greater evolving mass flow in the turbine and to the intercooled compression;
- reduction in NO_X production in combustion chamber thanks to the steam which homogenizes the temperature;
- possibility of an easy power output modulation.

APPENDIX 2

The simulations carried out to optimize the cycle do not take pressure losses in the gas end of the exchangers (about 2% in the water - gas exchangers and 3% in gas - gas exchangers on both ends) evidently leads to different results. Reviewing the example with ϕ=0.3 we obtained the following comparative results.

	without pressure losses	**with pressure losses**
η	55.6	54.0
W (kW/kg)	386.6	361.5

Note that with a 6.5% power output decrease caused by the lower drop in available turbine pressure, efficiency decreased by 2.9%. The lower decrease in efficiency as compared to power output is due to the regenerative effect of the cycle which uses the higher temperature of the exhaust gases to increase the heat recovered in the recuperator and thereby decreases fuel consumption in the combustion chamber.
The energy needed to treat the water for the evaporator is not taken into account because it is negligible compared to power output.

ACKNOWLEDGMENTS

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