Int.J. Applied Thermodynamics,
Vol.4, (No.4), pp.165-171, December-2001
ISSN 1301-9724

Influence of the After Cooler on the Efficiency of the Humid Air Turbine[#]

Ireneusz SZCZYGIEŁ
Technical University of Silesia
Institute of Thermal Engineering
ul. Konarskiego 22, 44-100 Gliwice - Poland
E-mail: szczygie@polsl.gliwice.pl

Abstract

The published schemes of the Humid Air Turbine (HAT) contain usually the after cooler of the compressed air. The after cooler elongates the chain of processes and therefore introduces additional exergy losses. Calculations have been performed in order to prove that the after cooler in the proposed installations does not increase the efficiency of HAT.

Key words: humid air turbine, HAT, final cooler, HAT efficiency, gas turbine simulation

1. Introduction

The Humid Air Turbine (HAT) power plant analyzed here is equipped with a regenerative preheater of compressed air as well as with an open blade cooling system. The main feature of HAT turbine is the humidification of compressed air resulting in a lower air excess ratio in the combustion chamber that increases the plant efficiency. The liquid water is injected into the compressed air just after the second stage of the compressor. Injection can be realized in two ways:

- Single-point injection (Szargut and Cholewa 1997)
- Multi-point injection (Szargut and Szczygiel 1999)

As the results show, multi-point injection does not increase plant efficiency, but allows one to avoid the presence of droplets in the air stream. The droplets can have negative influence on the regenerator life.

As was mentioned, the turbine blades cooling system has been applied in the HAT plant considered herein. Compressed air is taken from the main stream and directed into the channels of the cooled blades. After flowing through the channels the cooling air mixes with the main stream of the flue gasses. Based on the results of previous calculations (Szargut, Skorek and Szczygiel 1998) it can be shown that open cooling system significantly decreases HAT plant efficiency. On the other hand, an open system is much simpler and cheaper than a closed cooling system. The efficiency drop due to the blade cooling is a function of the cooling air amount that depends on the temperature of flue gasses after the combustion chamber. The fraction of cooling air ratio can be estimated from the equations given in (Szargut 2000):

$$\alpha = \alpha_1 \frac{T_F - T_C}{T_F - T_1} \frac{T_T - T_{Tm}}{T_{T1} - T_{Tm}} \tag{1}$$

$$\beta = \beta_1 \frac{T_F - T_C}{T_F - T_1} \frac{T_T - T_{Tm}}{T_{T1} - T_{Tm}} \tag{2}$$

$$\gamma = \gamma_1 \frac{T_F - T_C}{T_F - T_1} \frac{T_T - T_{Tm}}{T_{T1} - T_{Tm}} \tag{3}$$

where α stands for the main air stream fraction supplied to the first stage of turbine, β the main air stream fraction delivered to the second stage of turbine and γ denotes the main air stream fraction which flows to the third stage. The equations were theoretically derived from the experimental data.

The operational schemes of the single- and multi-point water injection HAT turbine taken under consideration are shown in *Figures 1* and *2*.

[#] This paper was presented at the ECOS'01 Conference in Turkey July 4-6, 2001 and revised.

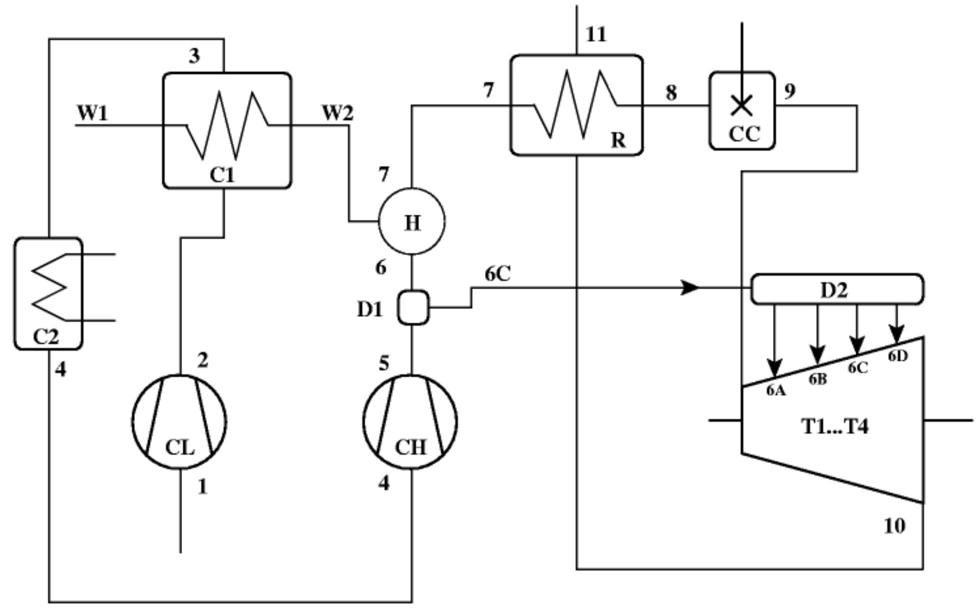

Figure 1. Scheme of the single-point water injection HAT power plant with cooling of turbine blades (CL, CH – low pressure and high pressure stage of the compressor, T1..T4 – stages of the turbine, C1 – interstage cooler and preheater of the injection water, C2 – external interstage cooler, CC – combustion chamber, H – humidifier of the compressed air, D – distributor of the cooling air, R – regenerative air preheater, 1...11 – streams of working fluid, W1, W2 – streams of the injection water).

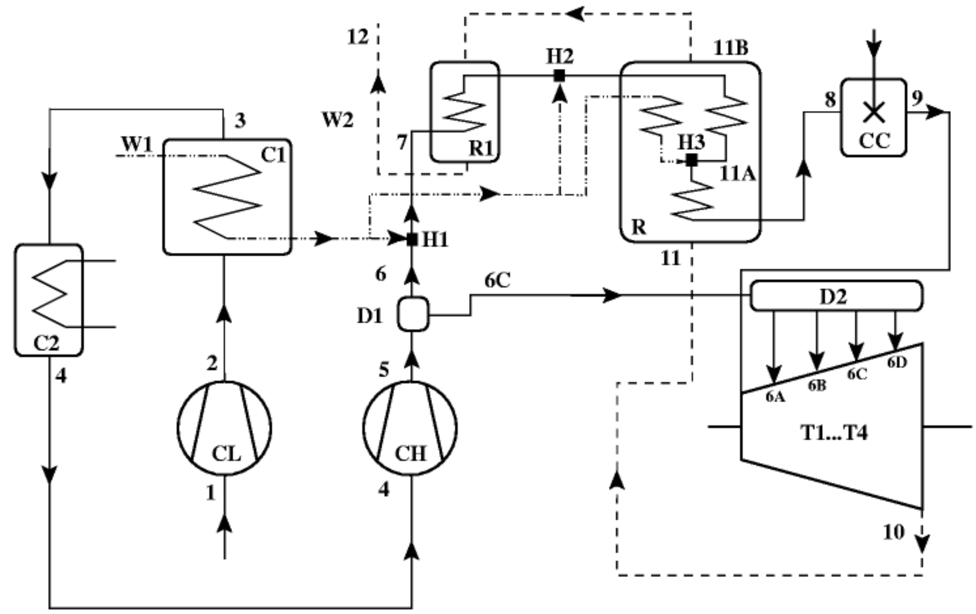

Figure 2. Scheme of the multi-point water injection HAT power plant with cooling of turbine blades (H1...H3 – stages of humidifier, other symbols as in Figure 1)

2. The Influence of the After Cooler on HAT Plant Efficiency

In the schemes proposed so far (Aargen et al. 1997, Krause, Tsatsaronis 1997, Jin et al. 1997, Chiesa et al. 1995), after the second stage of the compressor, a final cooler of compressed air has been installed (*Figure 3*). It makes it possible to decrease the temperature of combustion gasses flowing out from the system. So the enthalpy of the flowing out combustion gasses (stream 12 in the figures) becomes smaller, but, simultaneously the amount of heat rejected with cooling water in the after cooler becomes greater. The after cooler decreases the amount of blade cooling air, which has a positive influence on plant efficiency, but on the other hand elongates the chain of processes and so increases the exergy losses. After cooling in the

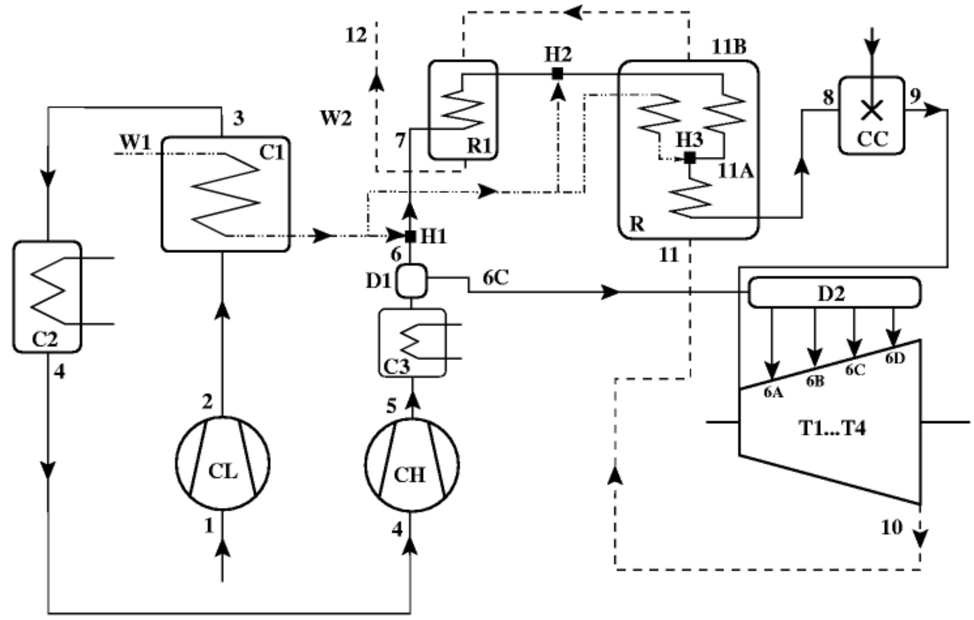

Figure 3. Scheme of the multi-point water injection HAT power plant with cooling of turbine blades with after cooler applied to the whole compressed air stream (C3 – after cooler, other symbol as in Figure 1)

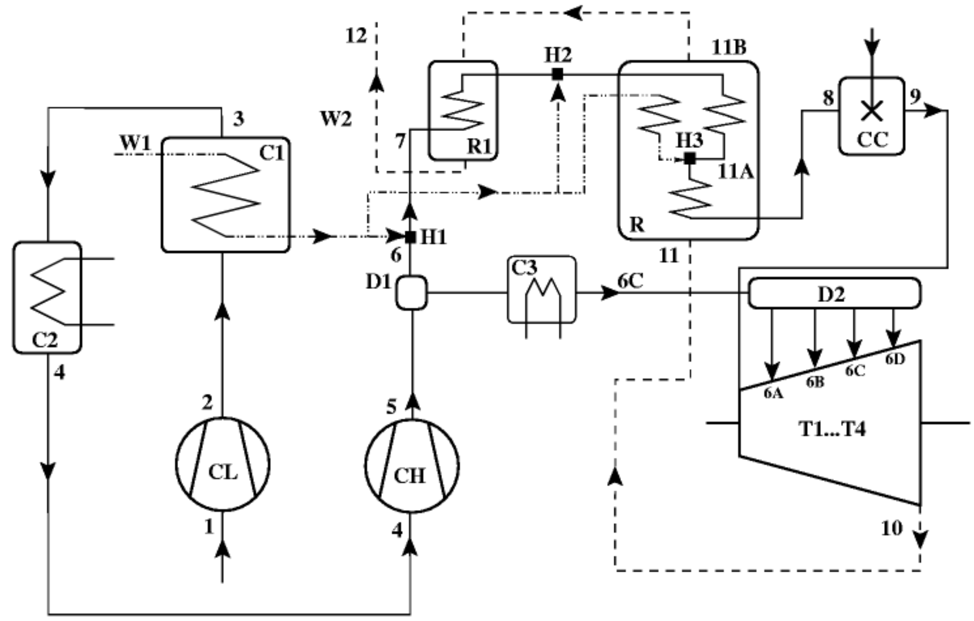

Figure 4. Scheme of the multi-point water injection HAT power plant with cooling of turbine blades with after cooler applied to the blades cooling air stream (C3 – after cooler, other symbol as in Figure 1)

final cooler, the compressed air is again heated in the regenerative preheater.. To avoid this negative effect, the final cooler should be applied only to the stream of blade cooling air (*Figure 4*). Such placing of the heat exchanger would affect the overall scheme by decreasing the amount of blade cooling air that significantly increases plant efficiency.

3. Assumed Data and Calculation Method

In the multicase calculations the following assumptions have been undertaken:

- Fuel: natural gas containing: 92% CH_4, 0.7% C_2H_6, 1.3% C_3H_8, 6% N_2;
- Temperature of combustion gasses before the turbine, T_T: 1200, 1300 and 1400°C;
- Temperature of environment and fuel: 10°C;
- Temperature difference between the combustion gasses and compressed air at the hot and cold ends of the air preheater 30K and 10K, respectively;
- Stage isentropic efficiency of the turbine and each compressor: 0.86 and 0.88

respectively;Mechanical efficiency of machines 0.98;

- Humidity of atmospheric air 80%;
- Humidity content in fuel 0.0242 kmol/kmol dry gas;
- Pressure losses in the air preheater at the air side 3% and at the combustion gas side 4.5%;
- Pressure losses in the compressor interstage air cooler 2.5%;
- Pressure losses in the external cooler 2.5%;
- Heat losses in the air preheater 1%;
- Heat losses in the combustion chamber 0.5%;
- Temperature of the injection water at the inlet to the interstage cooler 15^{o}C;
- Internal and mechanical efficiency of the injection pump 0.77 and 0.95, respectively;
- Efficiency of the electric generator 99%;
- Minimum temperature difference between air and water in the interstage cooler 10K;
- Air temperature before the second stage of compressor 30^{o}C;
- Air temperature after a final cooler T_1=35^{o}C;
- Reference temperature of combustion gasses before the turbine T_{T1}=1370^{o}C;
- Cooling air ratio supplied to the first stage of turbine (Eq. 1) α_l=0.13;
- Cooling air ratio supplied to the second stage of turbine (Eq. 2) β_l=0.06;
- Cooling air ratio supplied to the third stage of turbine (Eq. 3) γ_l=0.02;
- Minimal temperature of the gasses which require blade cooling T_{Tm}=800^{o}C;
- Temperature of air leaving blade cooling channels T_F=500^{o}C.

The internal efficiency of the entire turbine is greater than the efficiency of particular stages because of the recovered exergy of internal energy dissipation (hydraulic friction). The stage isentropic efficiency has been so assumed that the efficiency of the entire turbine equals that given by the producers.

Computational procedure should be iterative due to the fact that the final result (i.e. plant efficiency) is very sensitive to thermal parameters of system. To make the calculations objective, optimal values of crucial parameters should be undertaken. These values depend on many factors, so the only way to estimate them is to calculate them iteratively.

Computational procedure consists of several nested loops. In the first loop the amount of injection water is calculated. The criterion to be satisfied is temperature after the combustion chamber. In the second loop compression ratio is estimated necessary to achieve assumed temperature before the chamber. The last, the outermost loop is used to calculate the air excess ratio. This loop stops when the proper temperature is achieved. In the most nested point of procedure all thermodynamic parameters of the HAT plant are calculated. The block scheme of the whole computational procedure is shown in *Figure 5*.

As the analysis of results of computation shows, with the given temperature before the combustion chamber, and with the fixed temperature before the turbine, the optimal efficiency is achieved when the temperature difference (pinch) in the regenerator is minimal. So the air excess ratio is decreased as far as the

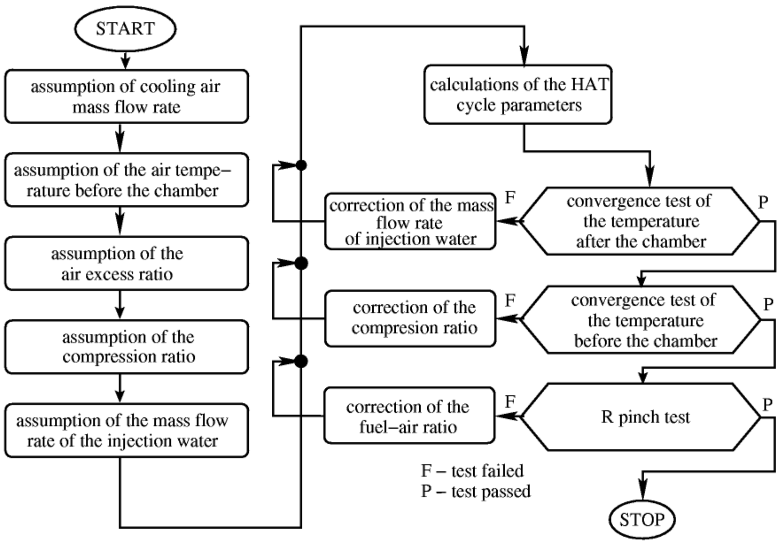

Figure 5. Block scheme of the calculations procedure. R pinch test denotes the test of minimal temperature difference in the regenerator (R in Figure 1)

pinch in the regenerator is achieved. The efficiency as a function of air temperature before the chamber has its optimum, so the full characteristic is calculated in order to place this optimal point.

4. Calculation Results

The calculations were performed in order to investigate:

- The influence of the after cooler presence in the system;
- The influence of the after cooler applied only on the cooling air stream;
- The influence of the temperature difference before and after the after cooler on the optimal efficiency of HAT plant;
- The influence of the after cooler application on the amount of cooling air.

All the calculations were performed for three values of the inlet temperature of working fluid to the turbine: 1200^{o}C, 1300^{o}C and 1400^{o}C for the system with single-point water injection, as well as multi-point water injection.

The results of calculations are presented in the TABLES I, II and III. Additionally, the linearized influences of the temperature drop in the after cooler on the various HAT parameters for the inlet temperature 1300^{o}C are presented in the *Figures 6, 7, 8* and *9*.

TABLE I. INFLUENCE OF THE AFTER COOLER APPLICATION ON THE HAT POWER PLANT EFFICIENCY [%]

temperature after the chamber T_T, ^{o}C	without after cooler		with after cooler		after cooler applied to the cooling air	
	single point injection	multi point injection	single point injection	multi point injection	single point injection	multi point injection
1200	48.19	47.67	45.70	45.35	48.38	47.88
1300	49.53	48.85	47.17	47.15	49.94	49.35
1400	50.78	49.82	48.59	48.57	51.38	50.69

TABLE II. INFLUENCE OF THE AFTER COOLER APPLICATION ON THE SELECTED THERMAL PARAMETERS OF THE SINGLE-POINT WATER INJECTION SYSTEM

temperature after the chamber T_T, ^{o}C	without after cooler			with after cooler			after cooler applied to the cooling air		
	humidity of the main air stream* kmol/kmol	amount of cooling air %	optimal air excess ratio	humidity of the main air stream* kmol/kmol	amount of cooling air %	optimal air excess ratio	humidity of the main air stream* kmol/kmol	amount of cooling air %	optimal air excess ratio
1200	0.197	15	2.75	0.10	11	3.13	0.170	11	2.80
1300	0.230	18	2.34	0.12	13	2.64	0.200	12	2.30
1400	0.277	22	1.96	0.14	15	2.26	0.220	14	2.01

* after the last humidifier

TABLE III. INFLUENCE OF THE AFTER COOLER APPLICATION ON THE SELECTED THERMAL PARAMETERS OF THE MULTI-POINT WATER INJECTION SYSTEM

temperature after the chamber T_T, ^{o}C	without after cooler			with after cooler			after cooler applied to the cooling air		
	humidity of the main air stream* kmol/kmol	amount of cooling air %	optimal air excess ratio	humidity of the main air stream* kmol/kmol	amount of cooling air %	optimal air excess ratio	humidity of the main air stream* kmol/kmol	amount of cooling air %	optimal air excess ratio
1200	0.18	15	2.88	0.10	11	3.44	0.16	10	2.93
1300	0.21	18	2.35	0.11	13	2.75	0.18	13	2.47
1400	0.25	22	1.99	0.15	16	2.41	0.21	15	2.11

* after the last humidifier

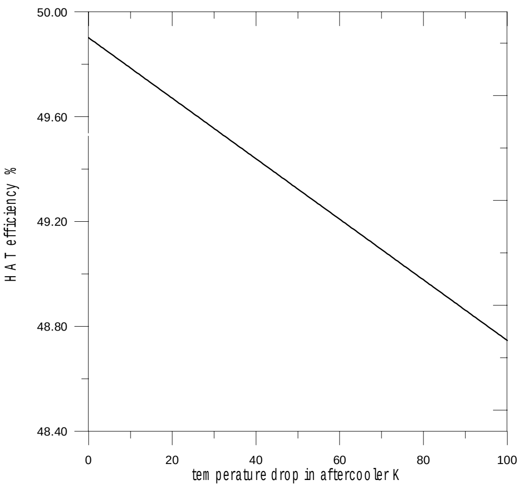

Figure 6. Influence of temperature drop in the after cooler on HAT turbine efficiency

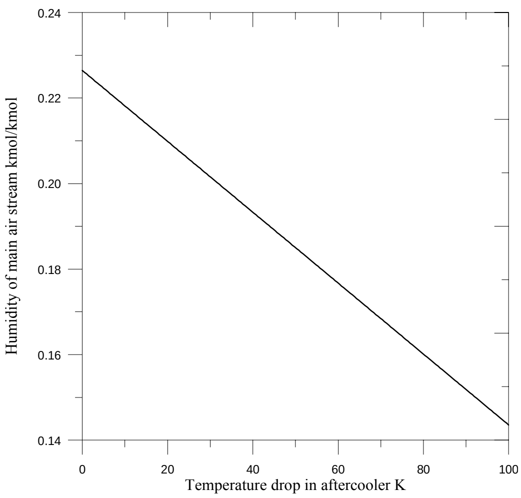

Figure 7. Influence of temperature drop in the after cooler on humidity of the main air stream

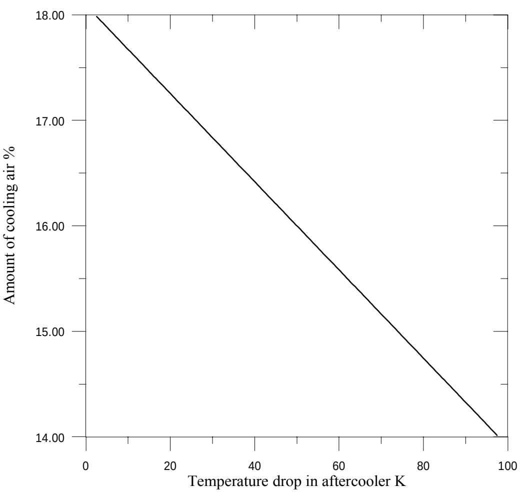

Figure 8. Influence of temperature drop in the after cooler on amount of the cooling air

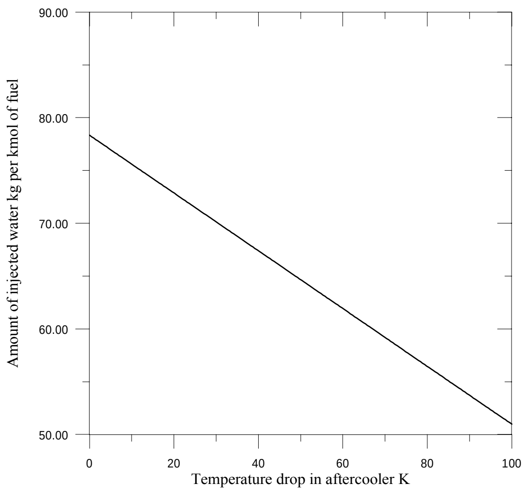

Figure. 9. Influence of temperature drop in the after cooler on the amount of injected water

5. Conclusions

1. Application of the after cooler leads to lower efficiency of the HAT power plant. The efficiency drop is proportional to the temperature drop in the after cooler.
2. The cooling air amount is lower in comparison with the scheme without any after cooler. This fact makes the efficiency drop smaller.
3. Installation of the after cooler significantly decreases the amount of the injected water.
4. The optimal value of the air excess ratio decreases when the temperature drop increases.
5. The humidity of the main air stream is lower in comparison with HAT without any after cooler.

Acknowledgement

The investigations have been supported by the Polish Committee of Scientific Research.

Nomenclature

- T_1 temperature of cooling air before the turbine,
- T_C temperature of the compressed air after the second stage of compressor,
- T_F temperature of the cooling air leaving the blade channels,
- T_T temperature of working fluid in the turbine inlet,
- T_{T1} reference temperature of working fluid in the turbine inlet, 1370 ^{o}C.
- T_{Tm} minimal temperature of working fluid in the turbine inlet, which requires blade cooling.
- α, β, γ main stream fractions supplied for blades cooling in the first, second and third stages of the turbine,

$\alpha_1, \beta_1, \gamma_1$ reference stream fractions supplied for blades cooling in the first, second and third stages of the turbine with the reference temperature in the inlet,

References

Aargen N. D., Cavani A., Westermark M. O., 1997, "New Humidifier Concept in Evaporative Gas Turbine Cycles", Proceedings of TAIES'97, pp. 270-276, Beijing.

Chiesa P., Lozza G., Macchi E., Consonni S., 1995, "An Assessment of the Thermodynamic Performance of Mixed Gas-Steam Cycles: Part B-Water Injected and HAT Cycles. Trans". ASME J. of Engineering for Gas Turbine and Power, Vol. 17, pp. 409-508.

Jin H., Cai R., Lin R., Zhang S., 1997, "Influence of Temperature Difference at the Bottom of Saturator on the Performance of HAT Cycle", Proceedings of TAIES'97, pp. 73-77.

Krause A., Tsatsaronis G., 1997, "Thermodynamic and Exergoeconomic Evaluation of the Humid-Air Turbine Cycle", Proceedings of TAIES'97, pp. 111-117, Beijing.

Lazaretto A. and Segato F., 1999, "A Systematic Approach to the Definition of the HAT-Cycle Structure Using the Pinch Technology". Proceedings of ECOS'99, pp. 215-222, Tokyo.

Szargut J., Skorek J. and Szczygiel I., 2000, "Influence of cooling of the Blades in the Open System on the Efficiency of Humid Air Turbine". International Journal of Applied Thermodynamics, Vol. 3, pp. 21-26.

Szargut J. and Szczygiel I., 1999, "Humid Air Gas Turbine with Staggered Injection of Water" (in Polish) Proc. IV Conference "Research Problems of Thermal Energetics", Technical University of Warsaw, Prace Naukowe, Mechanika, No. 181, pp. 250-256.

Szargut J. and Cholewa W., 1997, ".Silownia turbogazowa na wilgotne powietrze", (in Polish), Gospodarka Paliwami i Energia, No. 4, pp. 19-22.

Szargut J., 2000, "Analiza termodynamiczna elektrowni turbogazowej na wilgotne powietrze i jej zastosowania jako czlonu czolowego innych elektrowni" (in Polish) report of Polish Scientific Committee, Grant No. 8T10B02713.

Xiao Y. H., Cai R. X., Lin R. M., 1996, "Modelling HAT-Cycle and Thermodynamic Evaluation". Proceedings, ECOS'96, pp. 211-216, Stockholm.