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Humid Air Turbine as a Primary Link PRIVATE of a Conventional Gas Turbine Set

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Abstract.

The effectiveness of the primary link of a conventional gas turbine set has been expressed by means of the incremental energy efficiency, defined as the ratio of the attained increase of power to the increase of chemical energy consumption. The calculations performed for the humid air turbine (HAT) applied as a primary link indicate a high effectiveness of the utilization of the additionally consumed fuel. The compression ratio in both parts of the considered gas turbine set has been optimized in order to attain maximum energy efficiency.

Key words: humid air turbine, cascaded combustion, primary link

1. Introduction

Humid air turbine (HAT), operating with an injection of liquid water into compressed air, and equipped in regenerative preheating of the compressed air, can ensure a considerable increase of energy efficiency in comparison with conventional gas turbines.

Two main variants of HAT are considered. A prevailing number of authors consider the scheme with humidification of the compressed air by means of the saturation tower (Gallo 1996; Xiao et al., 1996). The scheme of HAT with a saturation tower is, however, rather complicated. Therefore, a scheme with immediate injection of water has been proposed (De Ruyck et al., 1996, 1997; Szargut and Cholewa 1997). This method can lead to a content of liquid droplets in the compressed air entering the regenerative preheater. In order to avoid the operational difficulties that can result from the content of liquid droplets, a staggered injection of liquid water has been proposed (Szargut and Szczygiel 1999). This leads to a slight decrease of the energy efficiency of the plant.

The actually applied high temperature values of working fluid entering the gas turbine require a blade cooling. An open cooling system is usually applied. The compressed air taken from the outlet of the compressor flows through the open channel inside the cooled blades and after that mixes with the working fluid. This cooling method leads to a considerable decrease of the energy efficiency of the HAT plant (Szargut et al., 2000). This effect which has been analyzed by Rufli (1990), depends on the consumption of cooling air. The consumption of cooling air can be reduced by lowering its temperature in the aftercooler of the compressed air. The prevailing part of considered schemes contains that aftercooler (Stecco et al., 1993; Lazareto and Segato 1999; Bram and de Ruyck 1996). However, the aftercooler elongates the chain of irreversible processes, and so introduces an additional exergy loss. The heat taken away from the compressed air in the aftercooler is again delivered in the regenerative preheater. Therefore, the aftercooler does not increase the energy efficiency of the HAT-plant (Szczygiel 2001). It is purposeful to apply the aftercooler only for lowering the temperature of the cooling air. Additional improvement can be attained by extracting the cooling air from between the stages of the compressor. This method, however, considerably complicates the scheme of the plant and gives only small positive effects.

Taking into account the above, the scheme of the HAT-plant considered in the present paper contains an immediate staggered injection of liquid water into the compressed air and an aftercooler used only for lowering the temperature of the cooling air that is taken from the outlet of the compressor. A linear dependence of the cooling air consumption on the temperature of the working fluid entering the gas turbine has been assumed (Szargut and Szczygiel 2002).

2. Indices of the Effectiveness of the Primary Gas Turbine

The application of the HAT-plant as a primary link of a conventional gas turbine power plant leads to a gas turbine complex with two stages of combustion. Systems of conventional gas turbines with two stages have been known for many years (Szargut 1973). The scheme of the HAT-power plant with two stages of combustion has been proposed and analyzed by Cohn (1999) and Nakhamkin et al. (1995, 1996, 1997). Nakhamkin has even received a US patent for that scheme (1994, 1995).

The aim of the present paper is the evaluation of the incremental effects of the additional consumption of expensive gaseous fuel.

Introduction of a gas turbine as a primary link of another thermal power plant increases the electric power of the plant but also the consumption of the chemical energy. The attained positive energy effect can be characterized by means of the incremental energy efficiency, defined as the ratio of the increase of electric power to the increase of the consumption of chemical energy at given consumption of chemical energy in the basic (initial) power plant (Szargut 1999). In the considered case of PGT that index can be expressed as follows:

$$\eta_{\Delta} = \frac{\Delta \dot{W}}{\Delta \dot{E}_{ch}} = \frac{\eta'_E \, \mu - \eta_E}{\mu - 1} \tag{1}$$

where:

η_E, η'_E - energy efficiency of a conventional gas turbine plant, and a plant with PGT,

μ - the ratio $\dot{E}'_{ch} / \dot{E}_{ch}$ of the consumption of chemical energy per time unit in the considered plants.

The incremental energy efficiency is usually higher than that of the basic power plant and of the separately operating gas turbine plant.

The positive energy effect attained in the total energy system may also be presented by means of the cumulative economy of chemical energy per unit of the additional electricity production. It should take into account that additionally produced electricity partially replaces the production of the conventional power plant. In Polish conditions the power plants fed with coal meet the increase of the demand for electricity. It can be assumed that the efficiency of electricity transmission is equal for all the considered plants:

$$\begin{aligned}\varepsilon^* &= \frac{1}{\eta_{tr} \, \Delta \dot{W}} \left(\frac{\Delta \dot{E}_{ch\,el}}{\eta^*_{d\,g}} - \frac{\Delta \dot{E}_{ch\,g}}{\eta^*_{d\,g}} \right) \\ &= \frac{1}{\eta_{tr}} \left(\frac{1}{\eta^*_{d\,s} \, \eta_{E\,el}} - \frac{\Delta \dot{E}_{ch}}{\Delta \dot{W} \, \eta^*_{d\,g}} \right) \\ &= \frac{1}{\eta_{tr}} \left(\frac{1}{\eta^*_{d\,s} \, \eta_{E\,el}} - \frac{1}{\eta^*_{d\,g} \, \eta_{\Delta}} \right)\end{aligned} \tag{2}$$

where:

$\Delta \dot{E}_{ch\,el}, \Delta \dot{E}_{ch\,g}$ - increase of the consumption of chemical energy per time unit in the replaced power plant fed with coal and the gas turbine plant,

$\eta^*_{d\,g}, \eta^*_{d\,s}$ - cumulative efficiency of delivery of the gaseous and solid fuel,

$\eta_{E\,el}$ - energy efficiency of the replaced conventional power plant.

The additional production of electricity with increased efficiency and by means of ecologically more efficient fuel, reduces the emission of *i*th deleterious combustion products. The cumulative incremental index characterizing this effect per unit of the additionally produced electricity may be expressed as follows:

$$\begin{aligned}\rho^*_{\Delta\,i} &= \frac{1}{\eta_{tr}} \left(\frac{\rho_{s\,i}}{\eta^*_{d\,s} \, \eta_{E\,el}} - \frac{\Delta \dot{E}_{ch} \, \rho_{g\,i}}{\Delta \dot{W} \, \eta^*_{d\,g}} \right) \\ &= \frac{1}{\eta_{tr}} \left(\frac{\rho_{s\,i}}{\eta^*_{d\,s} \, \eta_{E\,el}} - \frac{\rho_{g\,i}}{\eta^*_{d\,g} \, \eta_{\Delta}} \right)\end{aligned} \tag{3}$$

Specific production cost of electricity in a replaced power plant fed with coal may be calculated as follows:

$$k_{el} = \frac{\sigma}{\tau_n} j_{el} + \frac{1}{\eta_{E\,el}} \left(k_{ch\,s} + k_{op\,s} \right) \tag{4}$$

where:

j_{el} - investment expenditures per power unit of a replaced power plant fed with coal,

$k_{ch\,s}$, $k_{op\,s}$ - specific purchase cost of the chemical energy of fuel, and the operational cost (without fuel) in the replaced power plant.

Similarly the specific cost of the additional electricity produced thanks to the PGT may be determined:

$$k_{el\,\Delta} = \frac{\sigma}{\tau_n} j_{pr} + \frac{1}{\eta_{\Delta}} \left(k_{ch\,g} + k_{op\,g} \right) \tag{5}$$

where:

j_{pr} - investment expenditures per power unit of a primary HAT,

$k_{op\ g}$ - specific operational cost (without fuel) of the primary HAT per unit of electricity.

The indices k_{el} and $k_{el\,\Delta}$ determine the economical profitability of PGT. The chemical energy of gaseous fuel is more expensive than that of solid fuel, but the high incremental energy efficiency of the additional production of electricity ensures an advantageous value of the calculated incremental specific cost of electricity.

Equations (4) and (5) do not take into account the cost of ecological losses due to the deleterious emission of the combustion products. The introduction of the indices of ecological losses leads to the specific social cost of chemical energy of fuel:

$$\chi_{ch} = k_{ch} + \sum_i \rho_i \zeta_i \qquad (6)$$

where:

ρ_i - emission of the *i*th deleterious combustion product per unit of chemical energy

ζ_i - index of ecological losses per unit of the *i*th deleterious combustion product.

After introduction of the social cost of chemical energy into Eqs. (4) and (5), the social cost of electricity can be calculated:

$$\chi_{el} = \frac{\sigma}{\tau_n} j_{el} + \frac{1}{\eta_{E\,el}} (\chi_{ch\,s} + k_{op\,s}) \qquad (7)$$

$$\chi_{el\,\Delta} = \frac{\sigma}{\tau_n} j_{pr} + \frac{1}{\eta_{\Delta}} (\chi_{ch\,g} + k_{op\,g}) \qquad (8)$$

Example 1. The following indices of ecological losses can be accepted for gaseous combustion products: $\zeta_{SO_2} = \zeta_{NO_x}$ = 1.5 \$/kg, ζ_{CO_2} = 0.015 \$/kg, and for dust ζ_p = 0.3 \$/kg. The purchase price of chemical energy of hard coal is $k_{ch\,s}$ = 2 \$/GJ, the index of CO_2-emission is 98 kg/GJ, the index of SO_2 emission 0.03 kg/GJ (for S-content in fuel 0.65% and desulfurization efficiency 95%), index of NO_x emission 0.016 kg/GJ (after catalytic denox) and index of dust emission 0.15 kg/GJ. From Eq. (6) the resulting social cost of the chemical energy of hard coal:

$\chi_{ch\,s}$ = 3.55 \$/GJ.

Similarly for the high-methane natural gas the index of CO_2-emission is 55 kg/GJ, the assumed index of NO_x emission is 0.13 kg/GJ and the purchase cost of chemical energy is 4\$/GJ. The social cost of chemical energy is:

$\chi_{ch\,g}$ = 4.85 \$/GJ.

The emission of CO_2 considerably influences the social cost of chemical energy, despite a small assumed value of ecological losses resulting from the CO_2 emission.

3. Efficiency of the Complex Power Plant

Energy efficiency of the complex power plant has been determined for three values of the temperature of the working fluid entering the primary and basic gas turbine. When calculating the energy efficiency of the separately operating conventional gas turbine, it has been assumed that its compressor operates with intercooling, the initial rows of turbine blades are cooled with compressed air (open system) and the cooling air is taken from the outlet of the compressor.

It has been assumed that the demand for cooling air is proportional to the difference:

T_T - T_{Tm} of the actual temperature of working fluid before the turbine and the maximum temperature not requiring a blade cooling. It is inversely proportional to the temperature increase of cooling air within the cooling channel of the considered row of blades (Szargut and Szczygiel 2002):

$$\alpha = \alpha_1 \frac{T_F - T_c}{T_F - T_1} \frac{T_T - T_{Tm}}{T_{T1} - T_{Tm}},$$
$$\beta = \beta_1 \frac{T_F - T_c}{T_F - T_2} \frac{T_T - T_{Tm}}{T_{T1} - T_{Tm}}, \qquad (9)$$
$$\gamma = \gamma_1 \frac{T_F - T_c}{T_F - T_3} \frac{T_T - T_{Tm}}{T_{T1} - T_{Tm}}$$

where:

T_T - temperature of the working fluid before the turbine,

T_{T1} - temperature of the working fluid before the turbine that has been previously investigated,

T_{Tm} - highest temperature of the working fluid before the turbine not requiring a blade cooling,

α, β, γ - fraction of the compressed air used for cooling of the first, second and third stage of the turbine,

$\alpha_1, \beta_1, \gamma_1$ - previously determined values of α, β, γ at $T_T = T_{T1}$,

T_F - temperature of cooling air at the outlet of cooling channels,

T_c - temperature of cooling air at the outlet of air compressor,

T_1, T_2, T_3 - temperature of cooling air delivered to the first, second and third stages of the turbine.

The accessible data about the consumption of cooling air are not sufficient and controversial. Lazaretto and Segato (1999) cite the fraction 14% of compressed air at temperature 1300°C before the turbine. Rufli (1990) presents a dependence of

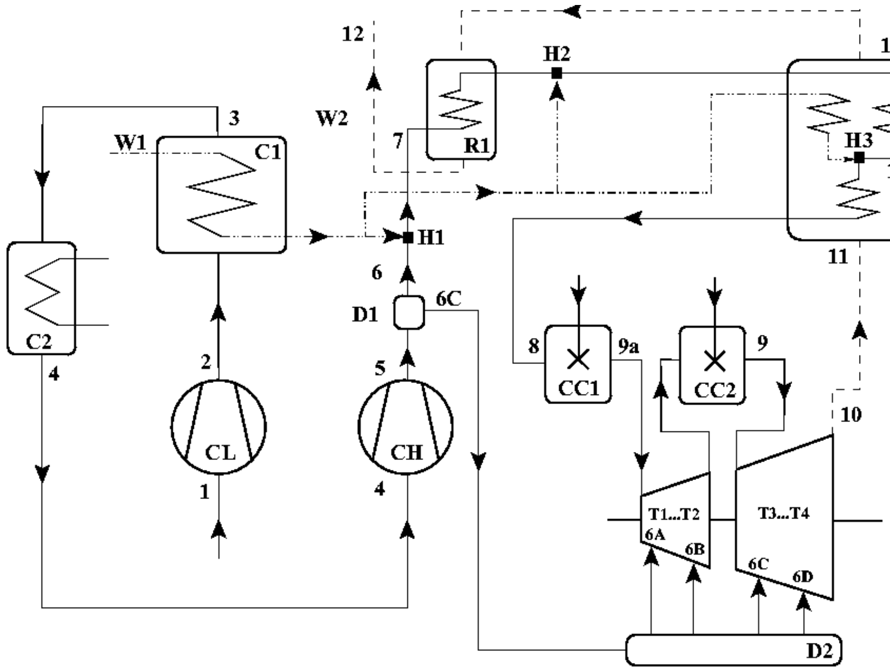

Figure 1. Scheme of the complex power plant with primary HAT

CL, CH — low pressure and high pressure compressor, C1 — intercooler, C2 —external cooler, R1, R — preheater of the compressed air, CC1, CC2 — combustion chambers, $T_1 \ldots T_4$ — stages of the turbine, D_1, D_2 — distributor of the cooling air, H_1, H_2, H_3 — humidifier, W_1, W_2 — water streams, 1...8 — streams of air, 9..12 — streams of working fluid.

relative consumption of cooling air on the temperature before the turbine. The high temperature part of the diagram of Rufli indicates that a maximum temperature before the turbine not requiring a blade cooling amounts to about 800°C, and at 1400°C the fraction of compressed air used for cooling is about 20%. The values cited by Szargut et al. (2000) have been used in the present paper. The scheme of the complex system is presented in *Figure 1*. When calculating the efficiency of this system it has been assumed that both turbines have a blade cooling and HAT operates with staggered injection of liquid water (Szargut and Szczygiel 1999; Szargut et al., 2000). The calculation scheme presented by Szargut et al. (2000) has been supplemented with a loop taking into account the partition of the total compression ratio in two parts the compression ratio, its partition ratio and the combustion air ratio λ have been optimized in order to attain a maximum efficiency.

Example 2. The following data have been accepted for exemplary calculations:

—fuel: natural gas containing 92% CH_4, 0.7% C_2H_6, 1.3% C_3H_8, 6% N_2,

—temperature of working fluid before the turbine: 1200, 1300, 1400°C,

—ambient and fuel temperature 20°C,

—temperature difference between the combustion gases and compressed air at the hot and cold end of the compressed air preheater 30 K and 10 K respectively, adiabatic internal efficiency of the stages of the turbine and parts of the compressor 0.84 and 0.88 respectively,

—mechanical efficiency of the engines 0.98,

—relative humidity of the ambient air 80%,

—humidity content in fuel 0.0242 kmol/kmol dry gas,

—pressure losses in each combustion chamber 3.5%,

—pressure losses in the compressed air preheater at the air side 3% and at the combustion gases side 4.5%,

—pressure losses in the intercooler of air 2.5% and in the external cooler 2.5%,

—heat losses in the air preheater 1% and in each combustion chamber 0.5%,

—water temperature at the inlet to the intercooler of air 15°C,

—internal and mechanical efficiency of the injection pump 0.77 and 0.95,

—minimal temperature difference of air and water in the intercooler 10 K,

—temperature of air before the second stage of the compressor 30°C,

—efficiency of the electric generator 99%,

—temperature of compressed air after the final cooler of compressed air 35°C,

—fraction of the compressed air delivered to the first, second and third stages of the turbine $\alpha_1 = 0.13$, $\beta_1 = 0.06$, $\gamma_1 = 0.02.$, (in the system without intercooler and temperature before the turbine T_T = 1370°C),

—maximum temperature of the working fluid before the turbine not requiring any blade cooling T_{Tm} = 800°C,

—temperature of cooling air at the outlet of cooling channels T_F = 500°C,

—efficiency of electric generator 99%,

—energy efficiency of the replaced power plant fed with coal $\eta_{Eel} = 0.38$,

—cumulative efficiency of coal and natural gas delivery $\eta^*_{d\,s} = 0.93$, $\eta^*_{d\,g} = 0.98$,

—efficiency of electricity transformation and transmission $\eta_{tr} = 0.9$.

TABLE I contains the expected indices of energy and ecological effects.

4. Exemplary Economic Calculations

Results obtained in the above section make it possible to calculate the indices characterizing the expected economical effects of the introduction of HAT as a primary link of the conventional gas turbine.

Example 3. The following data have been assumed for economical calculations:

—investment expenditures per power unit of the replaced power plant fed with hard coal j_{el} = 1200 \$/kW, — investment expenditures per power unit of primary HAT j_{pr} = 725 \$/kW,

—annual time of utilization of the nominal power τ_n = 6300 h/a,

—annual rate of fixed costs $\sigma = 0.12$,

—specific operational cost (without fuel) of the replaced power plant $k_{op\,s}$ = 1.25 \$/GJ,

—specific operational cost (without fuel) of the primary HAT $k_{op\,g}$ = 0.5 \$/GJ,

TABLE II presents the expected economical indices.

TABLE I. ENERGY AND ECOLOGICAL INDICES OF THE PRIMARY HAT

Temperature before the turbine °C	Energy efficiency		Ratio μ	Incremental efficiency η_Δ	Cumul. economy of chem. energy ε^*	Cumul. decrease of emission $\rho^*_{\Delta\,CO_2}$ kg/GJ
	η_E	η'_E				
1200	0.354	0.489	1.665	0.692	1.51	218
1300	0.366	0.502	1.640	0.715	1.56	220
1400	0.376	0.512	1.649	0.722	1.57	221

TABLE II. SPECIFIC PRODUCTION COST AND SOCIAL COST OF ELECTRICITY

Temperature before the gas turbine °C	Replaced power plant fed with coal		Primary HAT	
	k_{el} \$/kWh	χ_{el} \$/kWh	$k_{el\,\Delta}$ \$/kWh	$\chi_{el\,\Delta}$ \$/kWh
1200	0.054	0.068	0.0372	0.0416
1300	0.054	0.068	0.0365	0.0407

1400	0.054	0.068	0.0362	0.0405

5. Conclusions

Introduction of HAT as a primary link of a conventional gas turbine power plant ensures a very high utilization efficiency of the additional gaseous fuel. This effect has been confirmed by means of the incremental energy efficiency of the primary HAT turbine.

The specific cost of electricity additionally produced in the primary HAT link is lower than in the replaced coal-fired plant, despite the higher specific price of chemical energy of natural gas when compared with coal. The primary HAT link also ensures high ecological effects, especially a considerable reduction of CO_2 emission.

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Nomenclature

$\dot{E}$ energy flow rate (W)
j investment expenditures per unit of the electric power ($/W)
k_{ch}, k_{el} specific cost of chemical energy and electricity ($/J)
T temperature (^oC or K)
$\dot{W}$ electric power (W)

Greek letters

δ symbol of a loss
Δ symbol of an increase
η efficiency
η_Δ incremental energy efficiency
ζ_i index of ecological losses per unit of the *i*th component of combustion products, ($/kg)
ε^* cumulative economy of chemical energy per unit of the additional electricity production compared with the replaced power plant fed with coal
μ ratio of the consumption rate of chemical energy after and before the introduction of primary HAT
ρ_{gi}, ρ_{si} emission of the *i*th deleterious component of combustion products per unit of chemical energy of solid and gaseous fuel, (kg/J)
$\rho^*_{\Delta i}$ cumulative incremental decrease of the emission of *i*th deleterious component of combustion products, (kg/J)
σ annual rate of fixed costs, (1/a)
τ_n annual utilization time of the nominal power, (h/a)
χ specific social cost of chemical energy, taking into account the environmental losses, ($/J)

Subscripts

ch chemical
d delivery of fuel (extraction, processing and transportation)
E related to energy
el electricity or replaced power plant fed with coal
g gaseous fuel
i number of the deleterious component of combustion gases
me electro-mechanical
n nominal value
op operational
s solid fuel
tr transmission of electricity
Δ incremental
0 environment
1,2,… order number of state

Superscripts

* cumulative quantity after introduction of the primary HAT

Abbreviations

HAT humid air turbine
PGT primary gas turbine

References

Bram, S. and De Ruyck, J., 1997, "Exergy Analysis Tools Applied to Evaporative Cycle Design", *Proceedings of ECOS'96*, Alvforts et al., ed., Royal Institute of Technology, Stockholm, 1996, pp. 217-223.

Cohn, A., 1999, "Humid Air Turbines: CHAT and CASH Cycles". *Proceedings of International Conference on Humidified Gas Turbines*, Brussels.

De Ruyck, J., Bram, S., Allard, G., 1996, "Humid Air Cycle Development Based on Exergy Analysis and Composite Curve Theory", ASME, Paper 96-CTP-39.

De Ruyck, J., Bram, S., Allard, G., 1997, "REVAP Cycle: a New Evaporative Cycle without Saturation Tower", *Journal of Engineering for Gas Turbines and Power*, 1997, October, Vol.119, pp.893-897.

Gallo, W.L.R., 1996, "A Comparison between the HAT Cycle (Humid Air Turbine) and Other Gas Turbine Based Cycles: Efficiency, Specific Power and Water Consumption", *Proceedings of ECOS'96*, Alvforts et al., ed., Royal Institute of Technology, Stockholm, pp. 203-210.

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*, Ishida, M. et al., ed., Tokyo Institute of Technology, Tokyo, pp. 215-222.

Nakhamkin M., 1994, "Cascaded Advanced High Efficiency Multi-Shaft Reheat Turbine with Intercooling and Recuperation". US Patent Number 5,347,806.

Nakhamkin, M., 1995, "Method of Generating Power with High Efficiency Multi-Shaft Reheat Turbine with Intercooling and Recuperation". US Patent Number 5,386,688.

Nakhamkin, M., Swensen, E.C., Wilson, J.M., Gaul, G. and Polsky, M., 1995, "The Cascaded Humidified Advanced Turbine (CHAT)", ASME Paper 95-CTP-5.

Nakhamkin, M., Swensen, E.C., Wilson, J.M., Gaul, G. and Polsky, M., 1996, "The Cascaded Humidified Advanced Turbine (CHAT)", *Journal of Engineering of Gas Turbines and Power*, Vol.118, pp. 565-571.

Nakhamkin, M., Swensen, E.C., Touchton, G., Cohn, A. and Polsky, M., 1997, "CHAT Technology: an Alternative Approach to Achieve Advanced Turbine Systems Efficiencies with Present Combustion Turbine Technology", ASME Paper 97-GT-142

Rufli, P., 1990, " Systematische Berechnungen über kombinierte Gas—Dampf Kraftwerke", Dissertation ETH Zürich, Nr 9178.

Stecco, S.S., Desideri, U., Facchini, B., Betagli, N., 1993, "The Humid Air Cycle: Some Thermodynamic Considerations", ASME Paper 93-GT-77.

Szargut, J., *Theory of Thermal Processes* (in Polish), PWN, 1973, Warszawa.

Szargut, J. and Cholewa, W., 1997, "Humid air turbine power plant" (in Polish) *Gospodarka Paliwami i Energi_*, No 4, pp. 19-22.

Szargut, J., 1999, "Energy and Ecological Effects of the Primary Gas-Turbine Supplementing a Coal-Fired Power Plant". *International Journal of Applied Thermodynamics*, Vol.2, pp.1-4.

Szargut, J., Skorek, J., Szczygiel, I., 2000, "Influence of Blade Cooling on the Efficiency of Humid Air Turbine". *International Journal of Applied Thermodynamics*, Vol.3 No 1, pp.21-26.

Szargut, J., Szczygiel, I., 1999, "Humid air turbine with staggered injection of water" (in Polish) Politechnika Warszawska, *Prace Naukowe, Mechanika* No 181, Warszawa, pp.250-256.

Szargut, J. and Szczygiel, I., 2002, "Thermodynamic and Economic Analysis of the Humid Air Turbine Cooperating with Other Power Plants"(in Polish), Report of the project 8T-10B-055-18 sponsored by the Polish Committee of Scientific Research, Silesian University of Technology, Institute of Thermal Technology, Gliwice.

Szczygiel, I., 2001, "Influence of the Aftercooler of the Compressed Air on the Energy Efficiency of the HAT". *Proceedings of ECOS'01*, Öztürk, A. et al., ed., Istanbul Technical University, Istanbul, pp.289-294.

Xiao, Y.H., Cai, R.X., Lin, R.M., 1996, "Modelling HAT Cycle and Thermodynamic Evaluation", *Proceedings of ECOS'96*, Alvforts et al., ed., Royal Institute of Technology, Stockholm, pp. 211-216.