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THERMODYNAMIC ANALYSIS OF A GAS TURBINE CYCLE FED BY DIFFERENT FUELS

TERMODYNAMICKÁ ANALÝZA OBĚHU SPALOVACÍ TURBINY PRO RŮZNÁ PALIVA

Abstract

The paper presents thermodynamic analysis of typical fuels in a theoretical gas turbine cycle. The fuels are burned at different temperatures and different working pressures. In a gas turbine system there are processes of air (and of gaseous fuel) compression, flue gas expansion and heat regeneration. They all run perfectly in the presented theoretical experiment. It means that the pressure changing processes work at the polytropic efficiency equals to one, the process intensities of combustion (as a chemical reaction) and heat exchange are set to one (i.e. the equilibrium is assumed), either. Presented results show that there is useful to distinguish between thermodynamic and the technological values of fuels.

Abstrakt

Příspěvek prezentuje termodynamickou analýzu typických paliv V teoretickém oběhu spalovací turbíny. Spalování paliv probíhá při různých teplotách a různých pracovních tlacích. V systému se spalovací turbínou jsou to procesy komprese vzduchu (a plynných paliv), expanze spalin a regenerace tepla. Všechny procesy probíhají V předloženém teoretickém experimentu perfektně. To znamená, že tlakové měnící se procesy fungují S polytropickou účinnosti rovnající se jedné, proces intenzity hoření (jako chemické reakce) a výměna tepla jsou také nastaveny na hodnotu jedna (rovnováha se předpokládá). Prezentované výsledky ukazují, že je užitečné rozlišovat mezi termodynamickými a technologické hodnotami paliv V obězích spalovacích turbín.

1 INTRODUCTION

Within a project section Thermodynamic Modeling, Analysis and Exergy Rating of Modern Power Technologies of the nation–wide Czech Research Program INTERVIRON results have been obtained that should be interesting for power engineers in their everyday practice and research. Origins of the applied method have been elaborated in the research team of Professor WOLFGANG FRATZSCHER, [1] one of the pioneers of the exergy method of thermodynamic analysis. Its basic assumptions are: introducing the so–called *intensities* as modeling dimensionless parameters (equal to zero — no process, equal to one — equilibrium reached, [2], in pressure changing processes the appropriate dimensionless parameter is identical to the polytropic efficiency, [3]), *extensities* (amount proportions) and *thermodynamic effectivity*, which is the generalized exergy efficiency. The last parameter allows rating complex systems of an arbitrary inner structure and determining the importance of a particular process for the whole system efficiency.

The following discussion is very important in understanding of the real (practical) influence of fuels characteristics onto a power system. The mostly used in balancing these systems fuel

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characteristics is its combustion heat (or the so–called combustion value). It is determined experimentally in an oxygen bomb calorimeter. The fuel is then burned in a pure oxygen and the reference temperature of 15°C (or 20°C) is taken into account. The particular fuel usability can be given also by the zero exergy, which could be determined applying the devalvation chemical reaction method by JAN SZARGUT.

2 GAS TURBINE CYCLE

Investigations will be led in a theoretical gas turbine scheme, the particular processes of which are set to be perfect ones. It means, the pressure changing processes run at polytropic efficiency of one (fuel and air compressions, expansion in a turbine), e.i. $\eta_{m,air}=1$, $\eta_{m,fue}=1$ and $\eta_{m,trb}=1$. Moreover, the equilibrium will be reached in the regenerative heat exchanger (appropriate process intensity $\iota_{hex}=1$), although it cannot be applied in all cases because the outlet temperature after the air compressor is higher than the turbine output temperature.

In Figure 1 the scheme of the gas turbine for gaseous fuels methane CH_4, hydrogen H_2 and carbon monoxide CO is presented, and in Fig. 2 the appropriate scheme for burning solid carbon C.

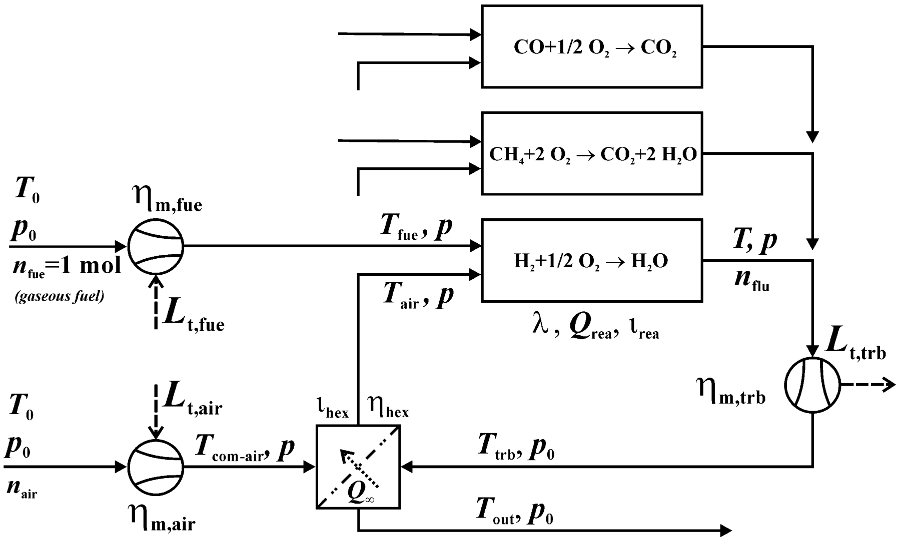

Fig. 1 Theoretical gas turbine scheme for burning gaseous fuels.

Chemical reactions of combustion are taken to be simple stoichiometric ones, with equilibrium at the formula right side (the side of reaction products). The reaction intensity has been assumed equal to one, either, i.e. $\iota_{rea}=1$. Thus, following chemical processes are considered:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \tag{01}$$
$$H_2 + 1/2O_2 \rightarrow H_2O \tag{02}$$
$$CO + 1/2O_2 \rightarrow CO_2 \tag{03}$$

and

$$C + O_2 \rightarrow CO_2 \tag{04}$$

In the general modeling method oriented onto thermodynamic analyzes additional dimensionless parameters have been introduced. They are giving an information about heat losses to the surroundings, e.g. for the technological heat exchange process it is η_{hex}. In the following investigations it has been set equal to 0 (no losses, if η_{hex} were equal to 1, there would be only losses).

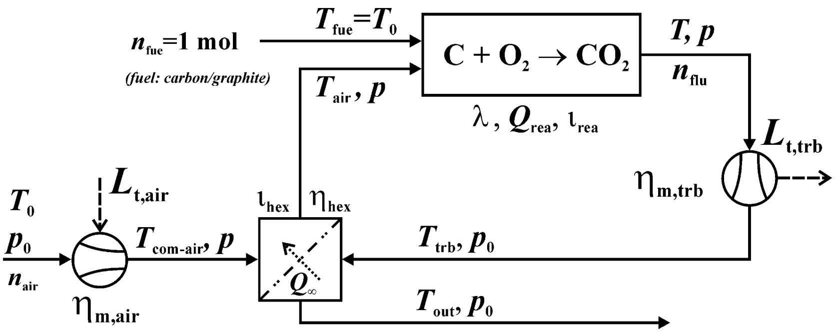

Fig. 2 Theoretical gas turbine scheme for burning the solid fuel carbon C.

3 COMPUTING PROCEDURE

The system presented in Figures 1 and 2 will be calculated in an iterative way. For the given values of process temperature T and process pressure p (at natural environment intensities T_0=283.15 K and p_0=1 bar) the numerical value of the air excess number λ will be searched, which fulfills the energy balance of the appropriate chemical reaction. The reaction heat is determined using methods of the chemical thermodynamics, i.e. it depends on the particular process temperature. Material data have been taken from [4] and zero exergy values from [5]. All parameters of the system for 1 mole of fuel could be calculated, and using resulting numerical values of particular technical (shaft) works special rating quotients have been formulated:

$$\eta_{0,\mathrm{ex}} = \frac{L_{\mathrm{t,trb}} - L_{\mathrm{t,air}} - L_{\mathrm{t,fuel}}}{\bar{e}^{0}_{\mu,\mathrm{fuel}}} \tag{5}$$

as a ratio of the net useful work and the particular fuel zero exergy at 283.15 K,

$$\eta_{0,\mathrm{th}} = \frac{L_{\mathrm{t,trb}} - L_{\mathrm{t,air}} - L_{\mathrm{t,fuel}}}{Q_{\mathrm{comb}}} \tag{6}$$

as a ratio of the net useful work and the combustion heat of the particular fuel,

$$\eta_{\mathrm{ex}} = \frac{L_{\mathrm{t,trb}} - L_{\mathrm{t,air}} - L_{\mathrm{t,fuel}}}{E_{\mathrm{Qrea}}} \tag{7}$$

as an exergy efficiency (for T_0=283.15 K) and finally

$$\eta_{\mathrm{th}} = \frac{L_{\mathrm{t,trb}} - L_{\mathrm{t,air}} - L_{\mathrm{t,fuel}}}{Q_{\mathrm{rea}}} \tag{8}$$

as a thermal efficiency. For the carbon C as a solid state fuel there is obviously $L_{\mathrm{t,fuel}}$=0, Q_{rea} is a reaction heat and Q_{comb} the particular combustion heat and E_{Qrea} its exergy.

Calculations have been proceeded for combustion temperatures 1200 K through 2000 K and pressures 5, 10, 15, 20 and 25 bar. For some cases — especially at higher pressures and lower combustion temperatures — the turbine outlet temperatures were lower than temperatures of the compressed air. Hence, there were no heat regeneration unit (heat exchanger), i.e. the intensity of the process has been set to zero (ι_{hex}=0). In Tabs.1–4 appropriate results are marked by asterisks. According to the practical characteristics of the gas turbine run, air excess numbers were high and their numerical values reached ca. 3…8.

4 NUMERICAL RESULTS

Some results have been presented in Tables 1 through 4 for solid fuel carbon C, methane CH_4, carbon monoxide CO as an important constituent of the so–called water gas (the product of fuel or biomass gasification), and hydrogen H_2. It is clear that even at idealized parameters of processes in a gas turbine scheme, the fuel exergy (the zero exergy) and the combustion heat could be utilized only partially, i.e. is could not be reached values of ratios according to Eqs.(5)–(8) close to the CARNOT efficiency, which is defined as

$$\eta_{\mathrm{th}}^{\mathrm{C}} = 1 - \frac{T_0}{T} \qquad (9)$$

and is obviously valid for a reversible case. It depends only on temperatures and its values are in the range between 0.7640 (for T=1200 K) and 0.8584 (for T=2000 K). The finite–time CARNOT efficiency (e.g. [6])

$$\eta_{\mathrm{th,finite-time}}^{\mathrm{C}} = 1 - \sqrt{\frac{T_0}{T}} \qquad (10)$$

numerical values are all inside the limits of 0.5142 and 0.6237.

Tab. 1 Results for coal (solid carbon) C as a gas turbine fuel.

T=	1200K	1300K	1400K	1500K	1600K	1700K	1800K	1900K	2000K
p=5 bar									
$\eta_{0\mathrm{ex}}$=	0.5706	0.5970	0.6195	0.6391	0.6561	0.6711	0.6844	0.6961	0.7067
$\eta_{0\mathrm{th}}$=	0.4738	0.4957	0.5144	0.5307	0.5448	0.5573	0.5683	0.5780	0.5868
η_{ex}=	0.8197	0.8369	0.8509	0.8626	0.8723	0.8806	0.8878	0.8939	0.8993
η_{th}=	0.6263	0.6546	0.6788	0.6997	0.7180	0.7340	0.7481	0.7607	0.7720
p=10 bar									
$\eta_{0\mathrm{ex}}$ =	0.4971	0.5292	0.5568	0.5807	0.6015	0.6199	0.6362	0.6507	0.6637
$\eta_{0\mathrm{th}}$=	0.4128	0.4394	0.4623	0.4822	0.4995	0.5147	0.5283	0.5403	0.5511
η_{ex}=	0.7140	0.7419	0.7648	0.7837	0.7998	0.8135	0.8253	0.8356	0.8446
η_{th}=	0.5456	0.5803	0.6101	0.6358	0.6582	0.6780	0.6955	0.7111	0.7250
p=15 bar									
$\eta_{0\mathrm{ex}}$ =	0.4904*	0.4905*	0.5137	0.5405	0.5640	0.5847	0.6030	0.6194	0.6340
$\eta_{0\mathrm{th}}$=	0.4072*	0.4073*	0.4265	0.4488	0.4683	0.4855	0.5007	0.5143	0.5264
η_{ex}=	0.7045*	0.6877*	0.7056	0.7296	0.7499	0.7672	0.7822	0.7953	0.8069
η_{th}=	0.5382*	0.5379*	0.5629	0.5918	0.6172	0.6394	0.6592	0.6768	0.6926
p=20 bar									
$\eta_{0\mathrm{ex}}$ =	0.5237*	0.5239*	0.5239*	0.5240*	0.5345	0.5570	0.5769	0.5947	0.6106
$\eta_{0\mathrm{th}}$=	0.4349*	0.4350*	0.4351*	0.4351*	0.4438	0.4625	0.4790	0.4938	0.5070
η_{ex}=	0.7523*	0.7344*	0.7197*	0.7073*	0.7106	0.7309	0.7484	0.7637	0.7771
η_{th}=	0.5748*	0.5745*	0.5741*	0.5738*	0.5849	0.6091	0.6307	0.6499	0.6671
p=25 bar									
$\eta_{0\mathrm{ex}}$ =	0.5478*	0.5479*	0.5480*	0.5481*	0.5481*	0.5481*	0.5551	0.5741	0.5911
$\eta_{0\mathrm{th}}$=	0.4548*	0.4550*	0.4551*	0.4551*	0.4551*	0.4551*	0.4609	0.4767	0.4908
η_{ex}=	0.7868*	0.7681*	0.7527*	0.7398*	0.7287*	0.7192*	0.7201	0.7372	0.7523
η_{th}=	0.6012*	0.6008*	0.6005*	0.6001*	0.5998*	0.5994*	0.6068	0.6273	0.6458

*) no regenerative heat exchange because of $T_{\mathrm{trb}}<T_{\mathrm{com-air}}$

Tab. 2 Results for methane CH_4 as a gas turbine fuel.

T=	1200K	1300K	1400K	1500K	1600K	1700K	1800K	1900K	2000K
p=5 bar									
η_{0ex}=	0.5408	0.5642	0.5841	0.6011	0.6157	0.6283	0.6393	0.6488	0.6570
η_{0th}=	0.5064	0.5284	0.5470	0.5629	0.5766	0.5884	0.5987	0.6076	0.6153
η_{ex}=	0.8028	0.8172	0.8285	0.8374	0.8444	0.8500	0.8543	0.8577	0.8604
η_{th}=	0.6133	0.6392	0.6610	0.6793	0.6950	0.7084	0.7199	0.7299	0.7386
p=10 bar									
η_{0ex}=	0.4762	0.5058	0.5311	0.5528	0.5717	0.5881	0.6025	0.6152	0.6263
η_{0th}=	0.4459	0.4737	0.4973	0.5177	0.5353	0.5507	0.5642	0.5761	0.5865
η_{ex}=	0.7068	0.7327	0.7534	0.7702	0.7841	0.7956	0.8052	0.8132	0.8200
η_{th}=	0.5401	0.5731	0.6010	0.6248	0.6453	0.6630	0.6785	0.6920	0.7040
p=15 bar									
η_{0ex}=	0.4706*	0.4706*	0.4933	0.5182	0.5399	0.5588	0.5755	0.5902	0.6032
η_{0th}=	0.4407*	0.4407*	0.4620	0.4853	0.5056	0.5233	0.5389	0.5527	0.5648
η_{ex}=	0.6986*	0.6817*	0.6998	0.7220	0.7405	0.7560	0.7691	0.7802	0.7898
η_{th}=	0.5337*	0.5332*	0.5583	0.5857	0.6094	0.6300	0.6481	0.6640	0.6780
p=20 bar									
η_{0ex}=	0.5027*	0.5028*	0.5029*	0.5030*	0.5143	0.5352	0.5536	0.5699	0.5843
η_{0th}=	0.4707*	0.4708*	0.4709*	0.4711*	0.4816	0.5012	0.5184	0.5337	0.5472
η_{ex}=	0.7462*	0.7282*	0.7134*	0.7008*	0.7054	0.7240	0.7398	0.7534	0.7651
η_{th}=	0.5702*	0.5696*	0.5691*	0.5685*	0.5806	0.6034	0.6234	0.6411	0.6568
p=25 bar									
η_{0ex}=	0.5259*	0.5260*	0.5261*	0.5263*	0.5265*	0.5266*	0.5349	0.5681	0.5681
η_{0th}=	0.4925*	0.4925*	0.4927*	0.4929*	0.4930*	0.4932*	0.5009	0.5320	0.5320
η_{ex}=	0.7806*	0.7618*	0.7463*	0.7332*	0.7221*	0.7124*	0.7149	0.7439	0.7439
η_{th}=	0.5964*	0.5959*	0.5954*	0.5948*	0.5943*	0.5937*	0.6024	0.6386	0.6386

*) no regenerative heat exchange because of $T_{trb} < T_{com-air}$

Tab. 3 Results for carbon monoxide CO as a gas turbine fuel.

T=	1200K	1300K	1400K	1500K	1600K	1700K	1800K	1900K	2000K
p=5 bar									
$\eta_{0\mathrm{ex}}=$	0.5856	0.6092	0.6286	0.6445	0.6577	0.6685	0.6773	0.6845	0.6902
$\eta_{0\mathrm{th}}=$	0.5707	0.5937	0.6126	0.6281	0.6409	0.6514	0.6601	0.6670	0.6726
$\eta_{\mathrm{ex}}=$	0.8062	0.8205	0.8314	0.8398	0.8462	0.8510	0.8546	0.8572	0.8589
$\eta_{\mathrm{th}}=$	0.6160	0.6418	0.6633	0.6813	0.6964	0.7093	0.7202	0.7294	0.7373
p=10 bar									
$\eta_{0\mathrm{ex}}=$	0.5138	0.5449	0.5708	0.5925	0.6108	0.6262	0.6392	0.6502	0.6595
$\eta_{0\mathrm{th}}=$	0.5007	0.5310	0.5563	0.5774	0.5952	0.6102	0.6229	0.6337	0.6427
$\eta_{\mathrm{ex}}=$	0.7074	0.7339	0.7549	0.7720	0.7858	0.7972	0.8065	0.8143	0.8206
$\eta_{\mathrm{th}}=$	0.5405	0.5740	0.6023	0.6262	0.6468	0.6644	0.6797	0.6929	0.7045
p=15 bar									
$\eta_{0\mathrm{ex}}=$	0.5067*	0.5055*	0.5286	0.5542	0.5759	0.5944	0.6102	0.6237	0.6353
$\eta_{0\mathrm{th}}=$	0.4938*	0.4926*	0.5152	0.5401	0.5612	0.5792	0.5946	0.6078	0.6191
$\eta_{\mathrm{ex}}=$	0.6976*	0.6807*	0.6992	0.7220	0.7409	0.7567	0.7699	0.7811	0.7905
$\eta_{\mathrm{th}}=$	0.5330*	0.5325*	0.5578	0.5857	0.6098	0.6307	0.6488	0.6647	0.6786
p=20 bar									
$\eta_{0\mathrm{ex}}=$	0.5412*	0.5399*	0.5386*	0.5371*	0.5474	0.5682	0.5862	0.6017	0.6150
$\eta_{0\mathrm{th}}=$	0.5274*	0.5262*	0.5248*	0.5234*	0.5334	0.5538	0.5713	0.5863	0.5994
$\eta_{\mathrm{ex}}=$	0.7451*	0.7271*	0.7123*	0.6998*	0.7042	0.7234	0.7396	0.7535	0.7653
$\eta_{\mathrm{th}}=$	0.5693*	0.5688*	0.5682*	0.5677*	0.5796	0.6029	0.6233	0.6412	0.6570
p=25 bar									
$\eta_{0\mathrm{ex}}=$	0.5661*	0.5648*	0.5634*	0.5619*	0.5604*	0.5587*	0.5655	0.5826	0.5974
$\eta_{0\mathrm{th}}=$	0.5517*	0.5504*	0.5490*	0.5476*	0.5461*	0.5445*	0.5511	0.5677	0.5822
$\eta_{\mathrm{ex}}=$	0.7794*	0.7606*	0.7451*	0.7321*	0.7210*	0.7113*	0.7135	0.7295	0.7434
$\eta_{\mathrm{th}}=$	0.5955*	0.5950*	0.5944*	0.5939*	0.5934*	0.5928*	0.6012	0.6208	0.6381

*) no regenerative heat exchange because of $T_{\mathrm{trb}}<T_{\mathrm{com-air}}$

Tab. 4 Results for hydrogen H_2 as a gas turbine fuel.

T=	1200K	1300K	1400K	1500K	1600K	1700K	1800K	1900K	2000K
$p=5$ *bar*									
η_{0ex}=	0.6469	0.6754	0.6994	0.7196	0.7368	0.7514	0.7638	0.7743	0.7833
η_{0th}=	0.5385	0.5622	0.5821	0.5990	0.6132	0.6254	0.6357	0.6445	0.6520
η_{ex}=	0.8056	0.8197	0.8306	0.8389	0.8452	0.8500	0.8536	0.8562	0.8579
η_{th}=	0.6155	0.6412	0.6626	0.6805	0.6956	0.7084	0.7193	0.7286	0.7365
$p=10$ *bar*									
η_{0ex} =	0.5678	0.6044	0.6354	0.6618	0.6846	0.7042	0.7212	0.7359	0.7487
η_{0th}=	0.4726	0.5031	0.5289	0.5509	0.5698	0.5861	0.6003	0.6126	0.6232
η_{ex}=	0.7071	0.7335	0.7546	0.7715	0.7853	0.7967	0.8060	0.8137	0.8201
η_{th}=	0.5403	0.5738	0.6019	0.6259	0.6463	0.6640	0.6792	0.6925	0.7031
$p=15$ *bar*									
η_{0ex} =	0.5599*	0.5606*	0.5886	0.6192	0.6456	0.6686	0.6886	0.7061	0.7214
η_{0th}=	0.4660*	0.4666*	0.4899	0.5154	0.5374	0.5565	0.5732	0.5877	0.6005
η_{ex}=	0.6972*	0.6804*	0.6990	0.7218	0.7407	0.7564	0.7696	0.7807	0.7902
η_{th}=	0.5327*	0.5322*	0.5576	0.5850	0.6096	0.6304	0.6485	0.6644	0.6783
$p=20$ *bar*									
η_{0ex} =	0.5980*	0.5988*	0.5995*	0.6000*	0.6137	0.6393	0.6616	0.6812	0.6985
η_{0th}=	0.4978*	0.4985*	0.4990*	0.4995*	0.5108	0.5321	0.5507	0.5670	0.5814
η_{ex}=	0.7447*	0.7268*	0.7119*	0.6995*	0.7041	0.7232	0.7394	0.7532	0.7651
η_{th}=	0.5690*	0.5685*	0.5680*	0.5674*	0.5795	0.6027	0.6231	0.6410	0.6567
$p=25$ *bar*									
η_{0ex} =	0.6256*	0.6264*	0.6272*	0.6277*	0.6282*	0.6285*	0.6383	0.6596	0.6785
η_{0th}=	0.5207*	0.5214*	0.5220*	0.5225*	0.5229*	0.5231*	0.5313	0.5491	0.5648
η_{ex}=	0.7790*	0.7603*	0.7448*	0.7318*	0.7206*	0.7110*	0.7133	0.7294	0.7432
η_{th}=	0.5952*	0.5947*	0.5941*	0.5936*	0.5931*	0.5926*	0.6011	0.6207	0.6380

*) no regenerative heat exchange because of $T_{trb}<T_{com-air}$

The most interesting ratio is the grade of the zero exergy utilization according to Equation (5), because the technical (shaft) work is equivalent to the exergy, [5] – [6]. There are, however, some irregularities, not only because of the heat regeneration. In Figure 3 the formulated rating quotients numerical values have been plotted against the combustion pressure. The example for methane has been taken at T=1200 K. Similar functions can be obtained for all discussed fuels (even for the solid one) and for all combustion temperatures, Tabs. 1 – 4. Usually the minimum value for the rating quotients are between p=10 bar and p=15 bar. It is because of the relative huge need for the technical (shaft) work to run air (and the gaseous fuel) compressors.

It should be strongly emphasized that the analysis is a pure thermodynamic one, and it can be economically motivated to run the gas turbine at pressures, for which the minimum values are reached.

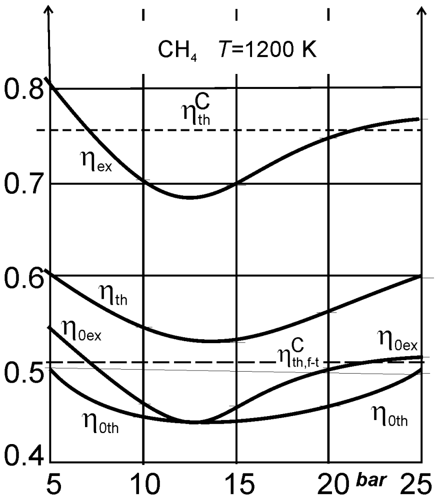

Fig. 3 Determined ratios as functions of the working pressure for methane driven turbine at process temperature of T=1200 K.

The above schemes of thermodynamic gas turbine cycles regarding the chemical reaction of combustion (according to the definition the combustion is a very rapid chemical oxidation reaction, the equilibrium of which lies at the side of products) can be used for some another investigation. E.g. for the solid coal (element carbon C) gas turbine the appropriate thermodynamic analysis has been presented in [7]. For the computations following dimensionless modeling parameters have been taken into account:

air compression	$\eta_{m,air}$	=0.95
intensity of heat exchange	i_{hex}	=0.75
thermal efficiency of heat exchanger	η_{hex}	=1 (no heat losses)
intensity of chemical reaction ($C + O_2 \rightarrow CO_2$)	ι_{rea}	=1
thermal efficiency of chemical reaction	η_{rea}	=1 (no heat losses)
expansion in a turbine	$\eta_{m,trb}$	=0.95

Thus, the whole system shown in Figure 2 can be univocally determined. The results are presented in Tables 5 and 6, once for the process pressure 10 bar, and another for 20 bar.

Tab. 5 The coal (carbon) fed gas turbine — process pressure p=10 bar.

process temperature	=	1200K	1300K	1400K	1500K	1600K	1700K	1800K	1900K	2000K
thermal efficiency	η_{th}	0.403	0.434	0.460	0.482	0.500	0.516	0.530	0.542	0.553
exergy efficiency	η_{ex}	0.528	0.555	0.577	0.594	0.608	0.619	0.629	0.637	0.644
thermodynamic effektivity	ε_{Σ}	0.841	0.848	0.854	0.859	0.864	0.868	0.871	0.875	0.877
expansion in a turbine	ε_{trb}	0.934	0.936	0.937	0.937	0.938	0.939	0.940	0.940	0.941
	γ_{trb}	0.366	0.364	0.361	0.359	0.357	0.355	0.353	0.351	0.350
air compression	ε_{air}	0.917	0.917	0.917	0.917	0.917	0.939	0.917	0.917	0.917
	γ_{air}	0.192	0.176	0.163	0.151	0.141	0.132	0.124	0.117	0.111
regenerative heat exchange	ε_{hex}	0.972	0.958	0.947	0.938	0.931	0.926	0.921	0.917	0.914
	γ_{hex}	0.020	0.033	0.045	0.055	0.065	0.074	0.082	0.089	0.096
combustion reaction	ε_{rea}	0.720	0.737	0.752	0.764	0.776	0.786	0.795	0.803	0.810
	γ_{rea}	0.422	0.427	0.431	0.435	0.437	0.439	0.441	0.442	0.444

Tab 6 The coal (carbon) fed gas turbine — process pressure p=20 bar.

process temperature	=	1200K	1300K	1400K	1500K	1600K	1700K	1800K	1900K	2000K
thermal efficiency	η_{th}	0.395	0.417	0.433	0.444	0.460	0.483	0.502	0.520	0.535
exergy efficiency	η_{ex}	0.517	0.533	0.542	0.548	0.559	0.579	0.596	0.611	0.624
thermodynamic effektivity	ε_{Σ}	0.857	0.858	0.860	0.862	0.865	0.870	0.874	0.878	0.881
expansion in a turbine	$\varepsilon_{\mathrm{trb}}$ γ_{trb}	0.933 0.403	0.934 0.396	0.935 0.391	0.936 0.386	0.937 0.384	0.938 0.383	0.939 0.382	0.939 0.381	0.940 0.380
air compression	$\varepsilon_{\mathrm{air}}$ γ_{air}	0.921 0.258	0.921 0.234	0.921 0.215	0.921 0.198	0.921 0.185	0.921 0.173	0.921 0.163	0.921 0.155	0.921 0.146
regenerative heat exchange	$\varepsilon_{\mathrm{hex}}$ γ_{hex}	–*	–*	–*	–*	0.996 0.003	0.986 0.011	0.978 0.019	0.971 0.026	0.965 0.033
combustion reaction	$\varepsilon_{\mathrm{rea}}$ γ_{rea}	0.719 0.339	0.737 0.369	0.752 0.394	0.765 0.415	0.776 0.428	0.786 0.432	0.795 0.436	0.803 0.439	0.810 0.441

*) no regenerative heat exchange because of $T_{\mathrm{trb}} < T_{\mathrm{com-air}}$

The thermal system efficiency has been calculated according to Eq.(8) with $L_{\mathrm{t,fuel}}=0$ and the exergy one according to Eq.(7) with $L_{\mathrm{t,fuel}}=0$, either.

Additionally the concept of the thermodynamic effectivity of a process and of the complex system has been applied. In particular formulas have been used, which were already presented and discussed in [5] – [6] and [8]. The whole system effectivity of the solid fuel driven gas turbine cycle, which is shown in Figure 2, can be determined with help of the general formula

$$\varepsilon_{\Sigma} = \sum \gamma_{\mathrm{i}} \varepsilon_{\mathrm{i}} = \gamma_{\mathrm{trb}} \varepsilon_{\mathrm{trb}} + \gamma_{\mathrm{air}} \varepsilon_{\mathrm{air}} + \gamma_{\mathrm{hex}} \varepsilon_{\mathrm{hex}} + \gamma_{\mathrm{rea}} \varepsilon_{\mathrm{rea}}$$

whereby γ_{i} is the mathematical weights factor of the in *i*–th process diminishing exergies. ε_{i} is the particular *i*–th process thermodynamic effectivity quotient, [5]–[6], [8].

5 CONCLUSIONS

The investigation of a theoretical gas turbine cycle has been made as one of numerous tests of the worked out new methodology for thermodynamic modeling, analysis and rating of power engineering and process and chemical engineering systems. The main conclusion is that the really utilized fuel energy potential is much lower than it can be judged from its heat of combustion or the zero exergy. Even the CARNOT factor cannot show the circumstance. That is why the results obtained can be called the technological values of selected fuels, because the thermodynamic ones are given by the combustion heat Q_{comb}, the so–called gross calorific value, or the zero exergy. The combustion heat is experimentally determined in a calorimeter as a combustion process in a pure oxygen, usually at p=0.1013 MPa=1 atm, and the reference temperature of 298 K. Besides Q_{comb} the so–called net calorific value (or lower heating value) is usually given to characterize the energy potential of fuels. It is always

$$Q_{\mathrm{comb}} > Q_{\mathrm{comb}}^{\mathrm{net}} \tag{11}$$

or exactly

$$Q_{\mathrm{comb}} = Q_{\mathrm{comb}}^{\mathrm{net}} + 2500 \cdot M_{\mathrm{H2O}} \tag{12}$$

where M_{H2O} is the mass of water steam in the flue gas after burning the test fuel sample and 2500 kJ/kg the condensation heat of H_2O.

Thus, the thermodynamic and technological value of the fuel can be distinguished. The presented investigations show that in analyzes of combustion processes lower practical heats should be expected as heats given by traditionally thermodynamic quantities.

The thermodynamic analysis of the solid carbon driven gas turbine presented above shows some interesting quantitative results of the dependence of the whole complex system efficiency (the thermodynamic effectivity) on a particular process efficiency. How it can be seen from the Tables 5 and 6, the most important (decisive) process is combustion, but its influence on the total efficiency of energy conversions in the gas turbine cycle is comparable with the influence of the gas expansion (γ_{rea} vs. γ_{trb}).

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REFERENCES

1. KOZACZKA J.: *Modellierung eines Kohlevergasungssystems und seine thermodynamische Auswertung*, Final Report: manuscript, Sektion Verfahrenstechnik (Wissenschaftsbereich Technische Thermodynamik und Energiewirtschaft, Leiter: W. FRATZSCHER), Technische Hochschule „Carl Schorlemmer” Leuna–Merseburg, Merseburg 1981
2. KOZACZKA J., KOLAT P.: *Thermodynamic Analysis and Rating of Modern Power Technologies* článek č. 1536, Transactions of the VŠB – Technical University of Ostrava, Mechanical Series No. 3, 2007, vol.LIII, article No. 1536, p.95–107, ISBN 978–80–248–1633–3, ISSN 1210–0471
3. KOZACZKA J., KOLAT P.: *Modeling of Working Pressure Changing Processes Oriented onto Thermodynamic Analyzes*, Transactions of the VŠB – Technical University of Ostrava, Mechanical Series No. 3, 2007, vol. LIII, article No. 1537, p.109–123, ISBN 978–80–248–1633–3, ISSN 1210–0471
4. SZARAWARA J.: *Termodynamika chemiczna stosowana*, wyd. 3 zmienione, Wydawnictwa Naukowo–Techniczne, Warszawa 1997, ISBN 83–204–2121–7
5. KOZACZKA J.: *Thermodynamic Analysis of Energy Conversion Processes*, Problems of Mechanical Engineering and Robotics Nr 8, Faculty of Mechanical Engineering and Robotics, AGH – University of Science and Technology, Kraków 2002, ISBN 83–913400–0–7
6. KOZACZKA J.: *Analiza termodynamiczna*, Wydawnictwo AGH, SU 1110, Kraków 1988
7. KOZACZKA J.: *The Use of Coal in a Theoretical Gas Turbine*, Mechanics (Cracow) No. 1, 2007, vol. 26, p.16–20
8. KOZACZKA J., KOLAT P.: *Exergy in Energy Conversion Processes*, Transactions of the VŠB – Technical University of Ostrava, Mechanical Series No. 2, 2010, vol. LVI