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ANALYSIS OF PARAMETERS AFFECTING THE EFFICIENCY OF A COMBUSTION CHAMBER

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Summary:

The efficiency of combustion chamber is one of the essential parameters during its design. It results from the equation of energy conservation in an open thermodynamic system, with heat supplied to such a system but without performing mechanical work. The article deals with the analysis of the partial losses and their effect on the efficiency. Practical experience shows that the heaviest effects on the efficiency of a combustion chamber are the loss by Q_{CH} chemical unburned matter, the Q_{DIS} loss by dissociation and loss caused by inhomogeneity of the flue gas temperature field behind the combustion chamber.

1. Introduction

The equation of energy conservation is as follows:

$$h_L(T_{2,M}) - h_L(T_0) + f\left[H_U(T_0) - Q_M - Q_{CH} - Q_{NOx} - Q_{DIS}\right] - Q_S = (1+f)\left[h_V(T_{3,M}) - h_V(T_0)\right] \tag{1}$$

where the air is denoted with the L index and V means the flue gases. The M index stands for medium values across the cross section, calculated on the basis of a mass flow. T_0 is reference temperature, f is fuel relation defined by the following equation

$$f = \frac{\dot{m}_B}{\dot{m}_L} , \tag{2}$$

where: $\dot{m}_B$ is the fuel mass flow rate. Other parameters in the No. (1) equation are: Q_M = loss caused by mechanical unburned matter, Q_{CH} = loss by chemical unburned matter, Q_{NOx} = heat loss caused by endothermic reactions during NO_X generation, Q_{DIS} = heat loss caused by dissociation, Q_S = heat loss by radiation into the surrounding – the latter being inconsiderable and therefore not considered in further calculations, $H_U(T_0)$=heating value at temperature T_0.

Formula for the calculation of efficiency of a combustion chamber results from the equation (1) (see [1])

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$$h_{BK} = 1 - \left(\frac{Q_M}{H_U} + \frac{Q_{CH}}{H_U} + \frac{Q_{NOx}}{H_U} + \frac{Q_{DIS}}{H_U} \right) - \left(\frac{(1+f)dh_K}{fH_U} + \frac{(1+f)dh_T}{fH_U} \right) =$$
$$= 1 - \left(\frac{Q_M}{H_U} + \frac{Q_{CH}}{H_U} + \frac{Q_{NOx}}{H_U} + \frac{Q_{DIS}}{H_U} + \frac{Q_K}{H_U} + \frac{Q_T}{H_U} \right) \tag{3}$$

where Q_K describes the effect of pressure losses in the combustion chamber and Q_T the impact of flue gas field homogeneity behind the chamber. The calculation of the Q_T losses consists in the calculation of energy losses that happen when mixing together the stream of flue gases of different temperatures (see [1]). Analysis of the Q_{NOx} and Q_{DIS} parameters is shown in [2].

From the No. (3) equation it can be seen that the efficiency of the proper combustion process, which consists of the reaction between the fuel and the combustion agent (oxidant) in the primary zone of the combustion chamber, is affected only by the Q_M and Q_{CH} quantities. Therefore, the combustion efficiency is expressed by the equation

$$h_S = 1 - \left(\frac{Q_M}{H_U} + \frac{Q_{CH}}{H_U} \right) \tag{4}$$

Fig.1 shows the Brayton cycle of the gas turbine with pressure losses. The increase of the entropy of the working fluid in balancing the non-uniform temperature field of the flue gases is denoted with Ds_3. As a result, the turbine specific work will decrease by the value dh_T. In other words, the consequence of the non-uniform field of parameters at the combustion chamber outlet is the decrease of enthalpy by dh_T.at the combustion chamber outlet.

Fig. 1 shows the pressure loss of the combustion chamber demonstrated by pressures p_2, p_3 (they are total pressures), which is usually defined as the coefficient

$$s_{BK} = \frac{p_3}{p_2} \tag{5}$$

or

$$x_{BK} = \frac{p_2 - p_3}{p_2} = \frac{Dp}{p_2}. \tag{6}$$

Additional compressor work denoted with dh_K in the compression process has to be exerted for the additional pressure increase at the compressor outlet. This work can be understood as energy loss in the combustion chamber as a result of its hydraulic loss.

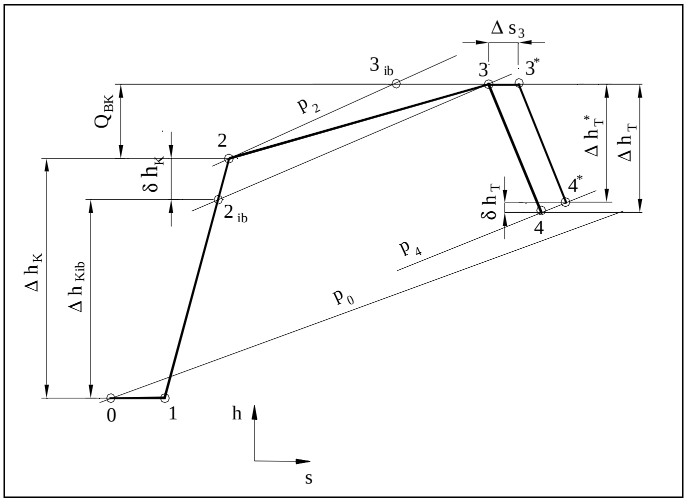

Fig.1: Brayton cycle with pressure losses and non-uniform temperature field at turbine inlet.

0 – suction system, 1 – compressor suction inlet, 2 – compressor discharge, 3 – turbine inlet casing, 4 – turbine outlet casing

2. Influence of pressure loss of combustion chamber on combustion chamber efficiency

The pressure loss of the combustion chamber is caused by friction, mixing of flows, supply of heat in the flame tube and the acceleration of air stream when it is streaming through the flame tube or nozzle openings. It follows from the equations (5), (6) that the following holds true:

$$x_{BK} = 1 - s_{BK}\,. \qquad (7)$$

For stationary turbines usually $x_{BK} = 0.02 - 0.04$, for aeronautical turbines $x_{BK} = 0.05 - 0.8$, for afterburning combustion chambers $x_{BK} = 0.07 - 0.12$.

The value dh_K can be calculated from the equation (see Fig.1)

$$dh_K = \Delta h_K - \Delta h_{Kib}\,, \qquad (8)$$

and it is true that

$$\Delta h_K = \frac{h_1}{h_K}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] \qquad (9)$$

$$\Delta h_{Kib} = \frac{h_1}{h_K}\left[\left(\frac{p_2}{p_1} s_{BK}\right)^{\frac{k-1}{k}} - 1\right]. \qquad (10)$$

By substituting (9) (10) into (8), we obtain

$$dh_k = \frac{h_1}{h_K}\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\left[1-\left(s_{Bk}\right)^{\frac{k-1}{k}}\right], \tag{11}$$

where h_1 is the air enthalpy at the compressor inlet and h_K is the isentropic efficiency of the compressor. The p_2/p_1 ratio is the specified compression ratio. The dependence between the increase of the specific compression work as a result of pressure loss dh_K of the combustion chamber and the pressure loss coefficient s_{BK} is shown in Fig.5.

Pressure losses of industrial combustion chambers are maintained within the range of $s_{BK} = 0.96 - 0.98$.

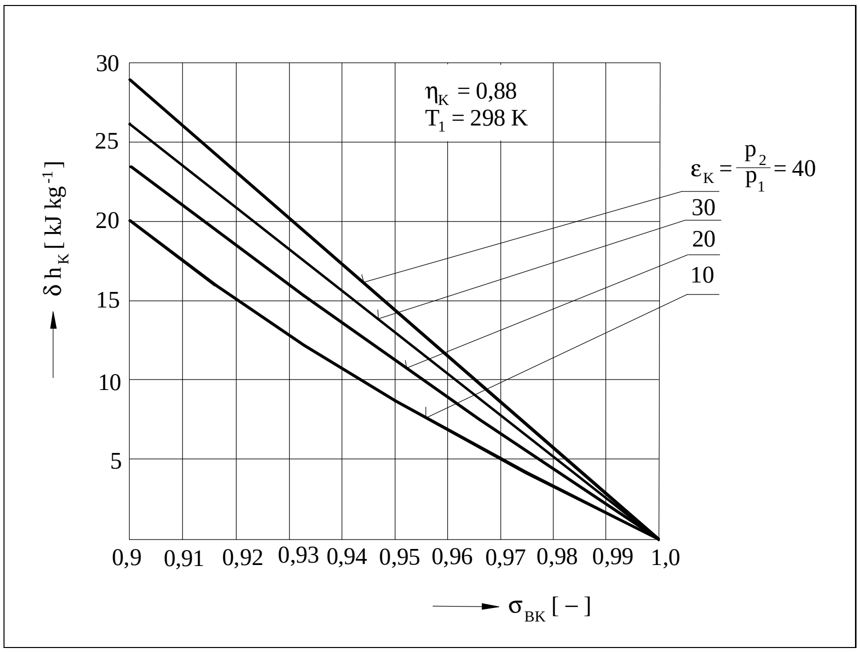

Fig.2: Increase of specific compression work as a result of the pressure loss of combustion chamber according to equation (11)

3. Influence of non-uniform temperature field at combustion chamber outlet on combustion chamber efficiency

The main parameters characterizing the gas turbine working cycle are the total temperature and enthalpy at the combustion chamber outlet. These parameters are used in thermodynamic calculations of the cycle as a whole also in the turbine calculation. It is assumed in all these calculations that the temperature T_3 and the enthalpy h_3 are uniformly distributed in the outlet cross-section and the flue gas composition corresponds to ideal combustion with the efficiency $h_S = 1{,}0$ with calculation excess air a_3 at the combustion chamber outlet.

Actually however, the distribution of temperature, enthalpy, pressures, velocities, fuel ratios is largely non-uniform. In such a case it is necessary to carry out the centering of these fields. Hereby, we will obtain a large number of parameters characterizing the non-uniformity of the aforementioned fields.

The process of balancing takes place in the flue gas expansion in the gas turbine and/or in the outlet nozzle or in the gas turbine outlet. The present measuring and evaluation methods applied to temperature fields, velocities and other quantities enable obtaining the distribution of these magnitudes across the cross-section with the maximum approach to the reality. The evaluation of sets of scalar magnitudes from the experiment by means of cubic splines enables obtaining a surface field with isosurfaces, which give a good idea of the non-uniformity of individual scalar magnitudes measured.

Such a temperature field gives the overall image of the distribution of temperatures across the surface and enables a good analysis of experimental results. The analysis of such a field consisting of isothermic surfaces enables studying the temperature field in each radial or peripheral direction and simultaneously evaluating the influence of individual design solutions on the temperature field. The analysis also permits to assign the size of the isothermic surface to each temperature and thus to assign to each temperature or another magnitude the proportional part in which it is located in the entire field. We will utilize it in the further analysis.

3.1 Enthalpy and exergy calculation

On the basis of the specifying nominal conditions for the given gas turbine cycle, the fuel ratio f_3 for an adiabatic combustion chamber with the ideal combustion is calculated in the outlet cross-section. Generally, it is true that $f_3 = f_1(t_3,t_2,t_B, \text{fuel})$. The temperature t_3 is calculated for this fuel ratio by an inversion function $t_3 = f^{-1}(f_3, t_2, t_B, \text{fuel})$. In foregoing expressions the denotations have the following meanings: 3- at combustion chamber outlet, 2- at combustion chamber inlet, B-fuel. It has to be pointed out that the temperature t_3, as well as other temperatures mentioned here are total temperatures. The temperature t_3 or T_3 is at the same time the thermodynamic temperature at the turbine inlet characterizing the inlet energy of flue gas stream. Actually, the temperature and pressure fields at the turbine inlet are non-uniform. It follows from the analysis that the influence of the non-uniformity of the pressure field on the flue gas stream enthalpy, entropy and exergy is negligibly small.

The entropy of the flue gas mixture is the function of temperatures T, T_o and pressures p, p_o and the function of the fuel ratio f and the absolute humidity of air d. Thereby, $T_o = 288.15$ K and $p_o = 101325$ Pa. The enthalpy of flue gas mixture depends on the temperatures T and T_o and on the composition of the mixture characterized by the fuel ratio f and the absolute humidity d. We will determine the corresponding fuel ratio f for each isothermic surface with the temperature T for the specified parameters at the combustion chamber inlet and the specified fuel. The exergy of flue gas mixture for an open thermodynamic system is the function of enthalpies and entropies for given temperatures and pressures for the reference temperature and pressure T_o,p_o according to equation

$$e = h - h_o - T\,(s - s_o). \tag{12}$$

The outlet cross-section of the combustion chamber is divided in d isothermic surfaces A_i and assigned to each surface is the temperature t_i , the proportional mass flow w_i and generally also the pressure p_i. Usually, however, we consider that $p_i = p_2 = \text{const}$. We will gradually mix the entire field according to the schematic diagram in Fig. 3.

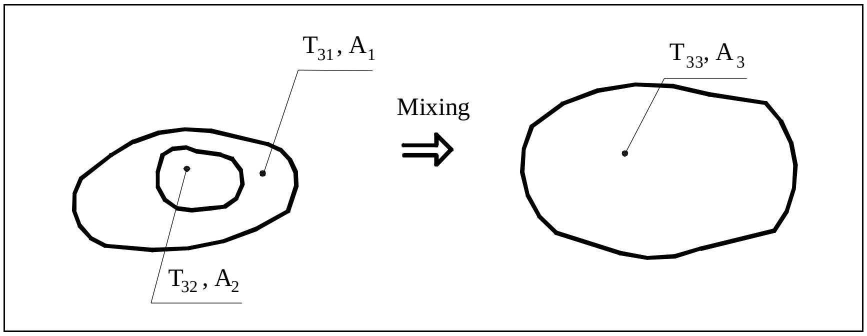

Fig.3: Schematic diagram of gradual mixing of the flue gas temperature field

The procedure of mixing two isothermic surfaces according to Fig. 3 is shown below and it is assumed that the inlet parameters are constant.

1) Relative temperature T_o and pressure p_o for the whole cross-section
2) $A_3 = A_1 + A_2$
3) We calculate the fuel ratios f_1, f_2 and their enthalpies h_1, h_2 for the specified fuel and inlet air temperature t_2 and the temperatures T_1, T_2 of isothermic surfaces
4) We calculate the enthalpy after mixing $h_3 = h_1 + h_2$
5) We calculate the fuel ratio for the mixture $f_3 = \frac{A_1}{A_3}(1+f_1)+\frac{A_2}{A_3}(1+f_2)-1$
6) We calculate the temperature after mixing $T_3 = f^{-1}(h_3, f_3)$
7) We calculate the exergy after mixing according to the equation (5.19)
   $e_3 = h_3 - h_o - T_o (s_3 - s_o)$
8) We calculate the exergy loss during mixing according to the equation

$$e_{ZTR} = (e_1 + e_2) - e_3 \tag{13}$$

The exergy loss during mixing can be expressed by the equation

$$\begin{aligned} e_{ZTR} &= (e_1 - e_2) - e_3 = [h_1 - h_{10} - T_o(s_1 - s_o)] + [h_2 - h_{20} - T_o(s_2 - s_{20})] - \\ &- [h_3 - h_{30} - T_o(s_3 - s_{30})]. \end{aligned} \tag{14}$$

After adjusting the equation (5.21) we obtain the relation

$$e_{ZTR} = T_o[s_3 - (s_1 + s_2)] = T_o \Delta s_{ZTR}. \tag{15}$$

The equation (15) represents the loss due to irreversibility. This is in harmony with the basic theorem that the difference of exergy between two thermodynamic states equals to maximal work that can be obtained from an irreversible action between these thermodynamic states at ambient parameters T_o, p_o. The difference between the maximal work a_{MAX} and the actual work equals to the expression

$$a_{MAX} - a = e_{ZTR} = (e_1 + e_2) - e_3 = T_o \Delta s_{ZTR}. \tag{16}$$

The loss caused by the non-uniformity of the temperature field at the combustion chamber outlet can be calculated from the following equation:

$$dh_T = \sum_{i=1}^{d} e_{ZTRi}, . \qquad (17)$$

where e_{ZTRi} stands for exergy losses with each partial mixing and d is the number of isothermic surfaces.

4.0 Thermal losses during NOx generation

Nitrogen oxides are matters considered toxic and therefore decreasing their concentration in flue gases is one of the most important elements in the development of combustion chambers for modern gas turbines. However, several thousands of gas turbines manufactured in the 50ies and 60ies of the 20th century and with very high NOx concentration in flue gases, sometimes more than 1000 mgm^{-3}, are currently in service. Such a high NOx concentration in the flue gases in these gas turbines may affect the combustion chamber efficiency. The reason for that is that in the generation of NO, which is the main NOx component, an endothermic reaction takes place [1] for which heat q_{NO} = 3040 KJ kg^{-1}_{NO} must be available. The nitrogen oxides consist of the NO and NO_2 components, with NO prevailing. The resulting emissions, however, are converted to NO_2 at standard ambient conditions, i.e. t_o = 273.15 K and p_o = 1.01325 bar. Both the NOx and CO emissions, as the main components of toxic matters in flue gases, are expressed either in mgm^{-3} or the so called EI_{NOx} or EI_{CO} emission indices.

The thermal loss during NOx generation is calculated from the equation

$$Q_{NOx} = EI_{NO} q_{NO}, \qquad (18)$$

but because it holds true that

$$EI_{NOx} = EI_{NO2},$$

EI_{NO} is expressed by the expression

$$EI_{NO} = EI_{NOx} \frac{M_{NO}}{M_{NO2}}. \qquad (19)$$

By substituting (19) into (18) we obtain

$$Q_{NOx} = EI_{NOx} \frac{M_{NO}}{M_{NO2}} q_{NO}. \qquad (20)$$

The ratio Q_{NOx} / H_u characterizes the efficiency decrease due to the endothermic reaction during NOx generation. By substituting q_{NO} = 3040 $kJkg^{-1}$, M_{NO} = 30.01 kg $kmol^{-1}$, M_{NO2} = 46.01 into the equation (20), we obtain the following equation for the efficiency decrease caused by NOx generation

$$\frac{Q_{NOx}}{H_u} = 196{,}6 \frac{EI_{NOx}}{H_u} \quad [\%]. \qquad (21)$$

If we consider NOx and CO harmful matters in the flue gases, the efficiency decrease of the combustion chamber caused by the impact of generation of harmful matters will be calculated from the equation

$$\frac{Q_{EMISE}}{H_u} = \frac{Q_{NOx}}{H_u} + \frac{Q_{CO}}{H_u} = 196{,}6 \quad \frac{EI_{NOx}}{H_u} + 1016{,}8 \quad \frac{EI_{CO}}{H_u}. \qquad (22)$$

The equation (22) will come out in per cent and it was considered that calorific value of CO is H_{uCO} = 10168 kJ kg^{-1}. The efficiency decrease caused by the NOx and CO generation for natural gas is shown in Fig. 4.

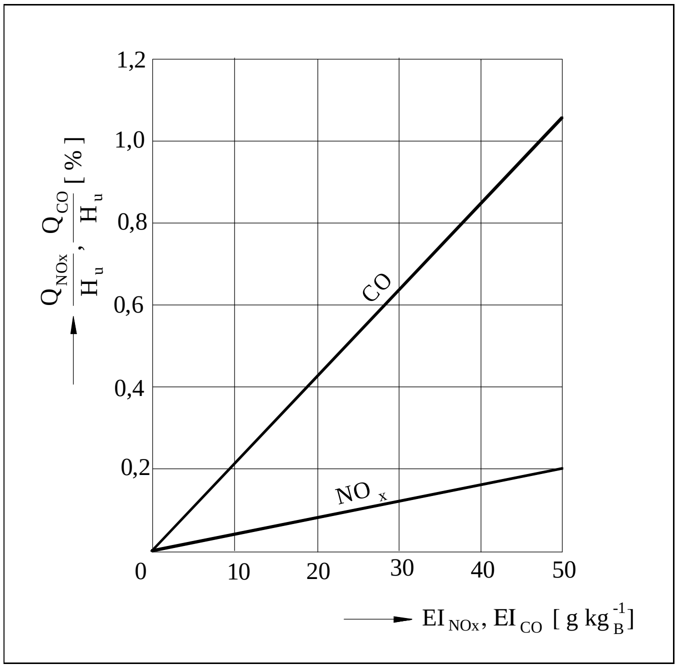

Fig.4: Efficiency decrease of combustion chamber due to NOx and CO generation. Fuel is natural gas according to Annex 1.

Fig.5 shows the dependence of the efficiency decrease of the combustion chamber due to NOx and CO emissions generation for various fuels.

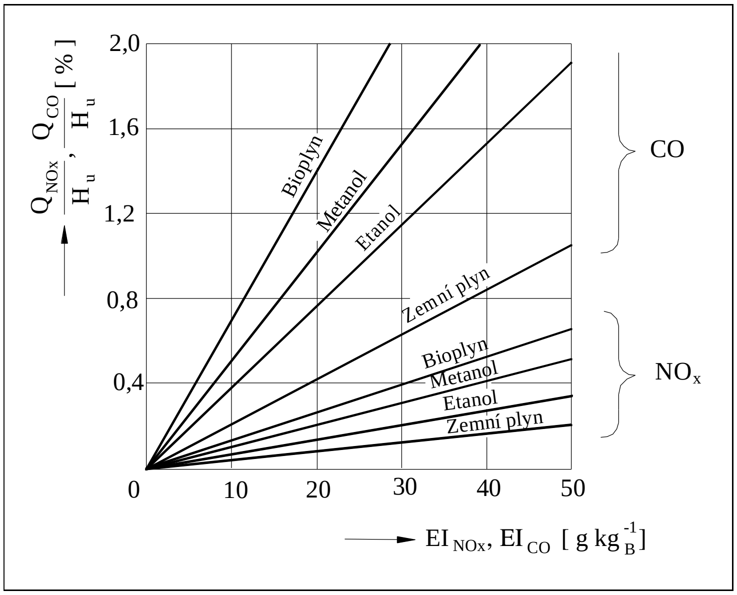

Fig.5.: Decrease of combustion chamber efficiency due to generation of NOx and CO emissions for different fuels.

5.0 Thermal losses in dissociation and their influence on combustion chamber efficiency

The calculation of the combustion process with dissociation is shown in [2]. Steady-state temperatures in natural gas burning are given in Fig.6.

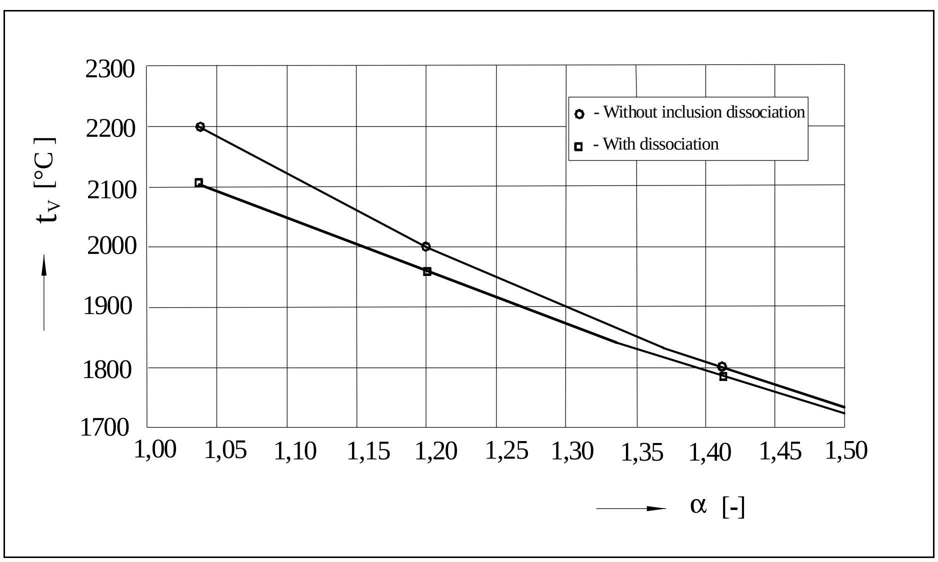

Fig.6: Steady-state temperatures in burning natural gas
Natural gas temperature: $t_p = 25^oC$, $t_2 = 350^oC$

It is obvious that the difference between the dissociation and non-dissociation steady-state flue gas temperature at the combustion chamber outlet increases with the decreasing value of the excess air coefficient a , and only values related to $a\langle 1{,}6.$ are of practical significance. The dissociation loss of the combustion chamber can be calculated from the equation

$$q_{DIS} = (h_V)_{ND} - (h_V)_{DIS}, \qquad (23)$$

where

$(h_V)_{ND}$ [Jkg_V^{-1}]- flue gas enthalpy at non-dissociation steady-state temperature and fuel ratio $f = f_{stech}/a$

$(h_V)_{DIS}$ [Jkg_V^{-1}]- the same at dissociation steady-state temperature

The dissociation thermal loss output related to a kilogram of fuel is then

$$Q_{DIS} = \frac{1+f}{f} q_{DIS} = \frac{1+f}{f}\left[(h_V)_{ND} - (h_V)_{DIS}\right] \qquad (24)$$

6. Analysis of particular losses on combustion chamber efficiency

An analysis of the combustion chamber efficiency h_{BK} was carried out for a gas turbine with the following parameters:

$t_2 = 170^oC$- inlet air temperature

$p_2 = 4.5\ x10^5$ Pa- inlet air pressure

$t_3 = 723^oC$- outlet flue gas temperature

$t_B = 25^oC$- fuel temperature

$d = 0.006$- specific air humidity

The fuel is natural gas with following parameters:

$$\left.\begin{array}{l} X_{H2O} = 1.363268 \\ X_{CO2} = 3.105086 \\ X_{SO2} = 0 \\ X_{O2} = 3.468354 \end{array}\right\} \text{componentsgenerated and consumed during stoichiometric combustion related to 1kg of fuel}$$

$H_U = 49172\ kJ\ kg_B^{-1}$ – calorific value of fuel

$f_{stech} = 0.058768$ - stoichiometric fuel ratio

The excess air coefficient air in the primary zone $a_{PRIM} = 1.3.$

The combustion chamber pressure loss was determined by measurement and is (total states)

$$x_{BK} = \frac{p_2 - p_3}{p_2} = \frac{Dp_{2-3}}{p_2} = 0.035.$$

Hence, it is true (see (7)) $s_{BK} = 1 - x_{BK} = 0.965$

Compressor isentropic efficiency is $h_K = 0.85$

The flue gas analysis at gas turbine outlet documented the following values:

CH_4 concentration: K(CH4) = 0.12 $mg\, m_V^{-3}$

C_2H_6 concentration: K(C2H6) = 0.02 $mg\, m_V^{-3}$

CO concentration: K(CO) = 105 $mg\, m_V^{-3}$

NOx concentration: K(NOx) = 301 $mg\, m_V^{-3}$

Soot concentration: K(C) = 25 $mg\, m_V^{-3}$

The measured concentrations are related to normal conditions determined as

p_o =1.01325 x 10^5 Pa, t_o = 15°C.

Furthermore, the following calorific value compounds are specified as follows:

H_U (CH_4) = 50 011 x 10^3 J $kg^{-1}{}_{CH4}$

H_U (C_2H_6) = 47483 x 10^3 J $kg^{-1}{}_{C2H6}$

H_U (CO) = 10168 x 10^3 J $kg^{-1}{}_{CO}$

H_U (C) = 35 000 x 10^3 J kg^{-1} c

The flue gas temperature field at the combustion chamber outlet is given according to Table 1.

Table 1: Temperature field at combustion chamber outlet

	$w_i = \frac{A_i}{A}$ [-]	t_{3i} [°C]		$w_i = \frac{A_i}{A}$ [-]	t_{3i} [°C]
1	0.08	520	12	0.09	770
2	0.05	549	13	0.05	783
3	0.15	670	14	0.04	790
4	0.05	679	15	0.01	795
5	0.07	685	16	0.04	799
6	0.02	695	17	0.05	805
7	0.02	700	18	0.02	810
8	0.03	705	19	0.01	820
9	0.02	710	20	0.04	825
10	0.03	720	21	0.03	829
11	0.02	725	22	0.08	855

The individual compounds affecting the combustion chamber efficiency were calculated for the specified parameters according to the equation (3). Below are shown the calculation results:

$$Q_M = 57.3 \cdot 10^3\ Jkg_B^{-1}; Q_{CH} = 523 \cdot 10^3\ Jkg_B^{-1}; Q_{NOx} = 3.87 \cdot 10^3\ Jkg_B^{-1}; Q_{DIS} = 200.9 \cdot 10^3\ Jkg_B^{-1};$$

$$Q_K = 0.8 \cdot 10^3\ Jkg_B^{-1}; Q_T = 138 \cdot 10^3\ Jkg_B^{-1}. \tag{25}$$

By substituting (25) into (3), we obtain the combustion chamber efficiency

$$h_{BK} = 0{,}981 \tag{26}$$

and by substituting Q_M a Q_{CH} into (4), we obtain the combustion efficiency

$$h_S = 0{,}988 \tag{27}$$

7. Conclusion

The above described method enables the calculation of the combustion chamber efficiency or the combustion efficiency on the basis of measurements of concentration of unburned hydrocarbons, soot, NOx and CO emissions in flue gases and on the basis of measurements of the combustion chamber pressure losses ant the non-uniformity of the temperature field at the combustion chamber outlet. The calculation enables to determine also the effect of the dissociation at primary zone temperatures higher than 1500°C. The results show that the strongly non-uniform flue gas temperature field at the combustion chamber outlet can reduce the gas turbine output by up to 0.5 % as compared with the output computed from the mean enthalpy values. The correction between the combustion efficiency h_S and the operating parameters $p_L, T_L, \dot{m}_L$, or the geometrical parameters D, L, V_{pl} is important for practical purposes. The index L denotes the air, D,L,V_{pl} denote the flame tube diameter, the flame tube length and the flame tube volume accordingly. Methods [3],[4] were developed, which enable to determine the combustion efficiency h_S as the function of the load parameter

$$k_V = \frac{\dot{m}_L}{p_L^{n_1} T_L^{n_2} Vpl}, \tag{28}$$

where n_1, n_2 are empirical values ascertained by experiments. For details see [2].

An example of the dependence of the combustion efficiency on the output parameter is shown in Fig.7, where the parameter is a_3, which is the excess air at the combustion chamber outlet.

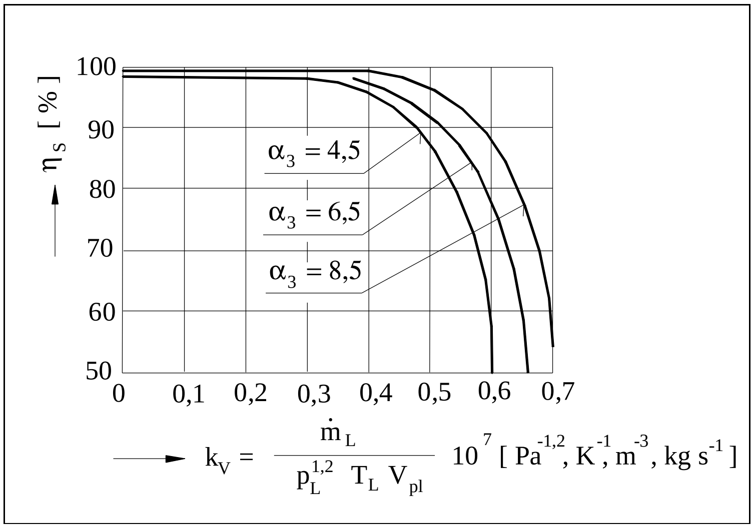

Fig.7: Dependence of combustion efficiency on the load parameter. Fuel: natural gas.

5. References

Zeldovic,Ja.B.,Sadovnikov,P.Ja.,Kameneckij,F.:"Okislenije azota pri gorenii", AN SSSR, 1947

Veselý, S : „Spalovací komory spalovacích turbin." (*Combustion chambers of gas turbines)* ISBN 80-239-6657-X,Galant,Brno , 2007.

Lefebre,A.H.: "Gas Turbine Combustion", Taylor & Francis, 1999

Mellor,A.M.: "Design of Modern Turbine Combustors", Academic Press, London, 1990