

POSSIBILITIES TO IMPLEMENT OXY-COMBUSTION IN STEAM BOILERS

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Rezumat. Arderea îmbogățită în oxigen reprezintă o tehnică modernă folosită în scopul îmbunătățirii performanțelor diferitelor sisteme termice. Ca aplicație importantă în generatoarele de abur, arderea îmbogățită în oxigen conduce la creșterea puterii cazanelor existente, respectiv la proiectarea unei noi generații de cazane modernizate. În lucrare sunt studiate ambele variante, și anume, impactul global al arderii îmbogățite în oxigen asupra proiectării cazanelor. Modelul prezentat tratează performanțele unui cazan existent (cazul de bază) în două ipoteze: ardere în prezența aerului atmosferic, respectiv ardere îmbogățită în oxigen, cu procente diferite de oxigen adăugat, analiza consecințelor îmbogățirii oxigenului și proiectarea termodinamică a unui nou cazan ce folosește arderea îmbogățită în oxigen.

1. INTRODUCTION

It is widely accepted that global climate change is due, in large part, to the emissions of greenhouse gases such as CO₂. Since fossil fuel will remain the primary energy source for some time, coal-fired power plants of the future will need to incorporate processes that can cost-effectively capture and sequester their CO₂ emissions. Fig. 1 shows the trends in CO₂ emissions for the coming decades, in terms of sources (coal, oil, and natural gas and total emissions).

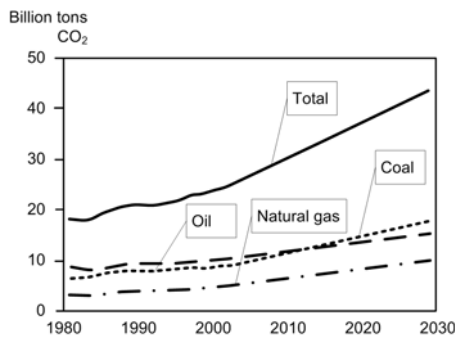


Fig. 1. Trends in CO₂ emissions.

If the present technologies remain unchanged, then the future of humankind is in jeopardy due to the cumulative effects of greenhouse gases. Fig. 2 presents

three scenarios for this future: one which involves present-day technology, one which supposes that a shift will be done in this direction by changing the technologies, and the third one (the most optimistic) which involves a technological progress so efficient that it will reduce and stabilize the greenhouse gases emissions to 550 ppm (the level corresponding to the year 1980).

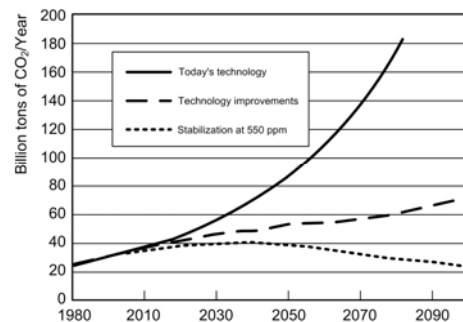


Fig. 2. CO₂ emissions scenarios.

At this moment, CO₂ capture and sequestration are the only methods that can control the carbon dioxide level in atmosphere. The leading CO₂ capture options for power plants include: postcombustion CO₂ scrubbing, oxy-combustion, and gasification with pre-combustion de-carbonization (See Fig. 3).

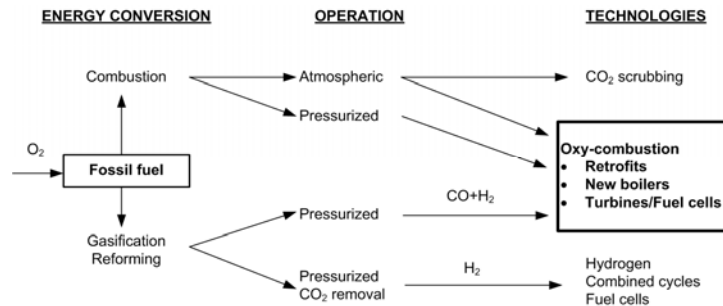


Fig. 3. CO₂ capture options.

In post-combustion CO₂ scrubbing, a sorbent is used to remove CO₂ from the flue gas. Chemical sorbents such as methyl ethyl amine (MEA) are used. In oxycombustion, the combustion air is replaced with a mixture of oxygen and recycled flue gas to produce a CO₂-rich flue gas. The re-circulated flue gas serves to moderate temperatures in the boiler and maintains heat transfer characteristics. In gasification with pre-combustion de-carbonization, coal is first gasified to produce syngas. This synthesis gas is then converted to a H₂/CO₂ mixture from which CO₂ is removed and hydrogen is used to generate power in a fuel cell and/or gas turbine system.

Simbeck [1] compared the above three CO₂ capture technologies when applied to an existing 292 MW pulverized coal-fired power plant. He concluded that oxy-combustion and gasification are more economically favorable than post-combustion CO₂ scrubbing. Nsakala et al. [2] also examined the technical and economic feasibility of CO₂ capture on an existing US coal-fired power plant, by evaluating the 450 MW Conesville, OH plant. This study also found that oxycombustion with flue gas recycle is more economically favorable than CO₂ scrubbing. Oxy-combustion can also be applied as a retrofit option for existing coal-fired power plants, but this option would, however, be limited by the modifications that can be applied to a specific plant. In each of the aforementioned studies, the oxygen source was assumed to be a cryogenic Air Separation Unit (ASU). Cryogenic air separation is a highly energy- and capital-intensive process. For oxy-combustion to be economically attractive, the cost of oxygen needs to be reduced substantially from current levels. However, cryogenic air separation is a mature technology and only limited improvements can be expected in the coming years. Oxygen Transport Membranes (OTMs) have been studied in great depth over the past decade and have been conceived as potentially a step-change technology in oxygen production. In the recent years, a large effort has been directed towards the development of air separation ceramic membranes, a theoretically simple method, which poses important technological chal-

lenges [3]. However, significant technical challenges remain, preventing this technology from becoming a commercially viable solution for economical oxygen production, at least in the near future. Hence there is a need for alternate, low cost oxygen production technologies. OEC has and will continue to become more and more attractive for an increased number of industrial applications, due to the major advantages related to burning of low value fuels, and to reducing emissions. In the future, oxy-combustion is regarded as one of the most economic solutions for use in advanced energy cycles with reduced environmental impact, especially in the case of very low (or absent) CO₂ emissions.

Supercritical steam boilers have been developed because of their good performance and of the high values of the superheated steam parameters. Such steam generators operate usually at pressures up to 350 bar. Other advantages that result in using supercritical boilers are: an increased thermal efficiency of the entire power plant, a better circulation by the absence of the water-steam separation in the furnace pipe headers, and the disappearance of the transition zone, which arises major difficulties in the operation [4]. Several important applications of the oxygen-enriched combustion in steam generators include the design of a new generation of advanced boilers and the gain in power resulting from the use of oxygen-enriched air in existing boilers. This effort tackles the first option, namely, the impact of oxygen enrichment on the boiler design. The proposed model consists of the combustion calculus for an existing boiler (the base case), the analysis of the consequences of oxygen enrichment, and the evaluation of the heat transfer surface areas necessary for a new boiler that uses oxygen-enriched combustion.

2. THE BASE CASE

The existing boiler representing the Base Case is a SCRPP (Supercritical, Radiant, Reheat and Pressurized) boiler. The data sheet of this boiler is summarized in Table 1.

Table 1

The Base Case

| HEAT TRANSFER SURFACES | |
|---|---|
| Furnace | Volume □ 3270 m ³ |
| Superheater | Multistage, bank of staggered pipes |
| Reheater | Single stage, bank of staggered pipes |
| Economizer 1 | Single stage, bank of staggered pipes with spiral fins Heat transfer surface area □ 16000 m ² |
| OPERATING CONDITIONS | |
| Steam mass flow rate (ṁ ^{''}) – primary: 1350 t/h | Steam mass flow rate (ṁ ^{''}) – reheat: 1260 t/h |
| Feedwater temperature: 260.5 °C | Air temperature leaving Air Heater: 318 °C |
| Superheater inlet/outlet temperatures: 374 / 540 °C | Reheater inlet/outlet temperatures: 305 / 538 °C |
| Gases temperature leaving Economizer: 376.7 °C | Superheater outlet pressure: 245 bar |
| Excess air: 7% | Gas temperature from Air Heater: 133.3 °C |
| FUEL | |
| Natural gas: 86.5% CH ₄ , 7.9% C ₂ H ₆ , 2.2% C ₃ H ₈ , 0.3% C ₄ H ₁₀ , 0.5% CO ₂ , 2.6% N ₂ | |

The heat exchange areas have been calculated for the Base Case and for the oxygen enrichment cases, and the differences between the different designs are analyzed.

The circulation scheme of the boiler is presented in Figure 4 and corresponds to the once-through scheme where the superheater stages are located in the downflow convection pass and the reheater is situated in the horizontal section, being directly exposed to the flame radiation.

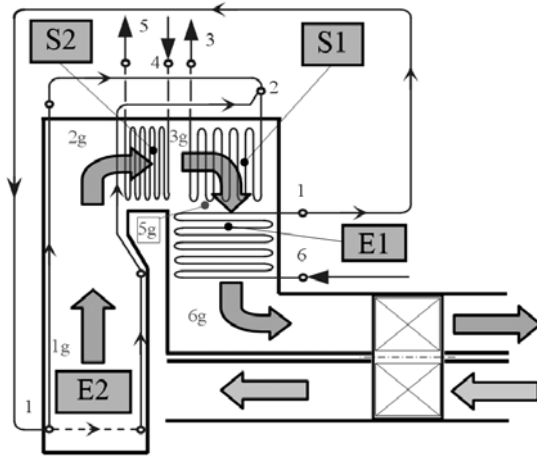


Fig. 4. Boiler circulation scheme.

The entire algorithm of combustion calculations, followed by the heat transfer calculus has been performed in order to establish the geometry of the superheater and of the reheater. The same procedure has been applied to the base case and to the oxygen enrichment cases, and the results have been compared.

2.1. The combustion calculus

The preliminary steps in the combustion calculus were:

- The determination of the minimum volumes of oxygen, dry air, humid air (an average air humidity of 40% has been assumed), and of the combustion air necessary for burning a cubic meter of fuel;
- The calculus of the volumes of combustion products and of flue gases per cubic meter of burned fuel.

The synopsis of the dissociation reactions is displayed in Table 2.

By using Table 2, we have derived the expressions for the volumes of CO₂, H₂O, H₂, CO, O₂, OH, O, H, N₂, NO, and N and by adding them, the expression of the flue gases volume after dissociation was obtained. After that, we have calculated the dissociated fractions, the equilibrium constants of the dissociation reactions, and the flue gases temperature after dissociation.

Table 2

| Combustion reactions | | | | |
|--|--|--|---|---|
| REACTION | VOLUMES OF REACTANTS | RESULTING VOLUMES AFTER DISSOCIATION | | |
| | | As final products | As secondary products participating in other reactions | Consumed from the reactants |
| 1 | 2 | 3 | 4 | 5 |
| A. PRIMARY REACTIONS | | | | |
| $2CO_2 \rightarrow 2CO + O_2$ $\begin{array}{ccc} 1 & 0 & 0 \\ -2a & 2a & a \\ \hline 1-2a & 2a & a \end{array}$ | y_1 [m ³ CO ₂ /m ³ fuel] | $(1-2a)y_1$ [m ³ CO ₂ /m ³ fuel] $2ay_1$ [m ³ CO/m ³ fuel] | ay_1 [m ³ O ₂ /m ³ fuel] | $-2ay_1$ [m ³ CO ₂ /m ³ fuel] |
| $2H_2O \rightarrow 2H_2 + O_2$ $\begin{array}{ccc} 1 & 0 & 0 \\ -2b & 2b & b \\ \hline 1-2b & 2b & b \end{array}$ | y_2 [m ³ /m ³ fuel] | | $2by_2$ [m ³ H ₂ /m ³ fuel] by_2 [m ³ O ₂ /m ³ fuel] | $2by_2$ [m ³ H ₂ O/m ³ fuel] |
| $2H_2O \rightarrow 2OH + H_2$ $\begin{array}{ccc} 1 & 0 & 0 \\ -2c & 2c & c \\ \hline 1-2c & 2c & c \end{array}$ | | $2cy_2$ [m ³ OH/m ³ fuel] | cy_2 [m ³ H ₂ /m ³ fuel] | $-2cy_2$ [m ³ H ₂ O/m ³ fuel] |
| | Remaining H ₂ O balance | $(1-2b-2c)y_2$ [m ³ H ₂ O/m ³ fuel] | | |
| $N_2 \rightarrow 2N$ $\begin{array}{ccc} 1 & 0 & \\ -d & 2d & \\ \hline 1-d & 2d & \end{array}$ | y_3 [m ³ N ₂ /m ³ fuel] | $2dy_3$ [m ³ N/m ³ fuel] | | $-dy_3$ [m ³ N ₂ /m ³ fuel] |
| B. REACTIONS WITH PRIMARY REACTIONS PRODUCTS AS REACTANTS | | | | |
| $O_2 \rightarrow 2O$ $\begin{array}{ccc} 1 & 0 & \\ -e & 2e & \\ \hline 1-e & 2e & \end{array}$ | $y_4' = y_4 + ay_1 + by_2$ [m ³ O ₂ /m ³ fuel] | $2e(y_4 + ay_1 + by_2)$ [m ³ O/m ³ fuel] | | $-e(y_4 + ay_1 + by_2)$ [m ³ O ₂ /m ³ fuel] |

Table 2 (continued)

| 1 | | | 2 | | 3 | | 4 | | 5 | |
|-----------------------------|---|---|-------------------------|--|-----------------------------|--|---|--|------------------------------|--|
| $O_2 + N_2 \rightarrow 2NO$ | | | | | | | | | $-fy_3$ [m^3O_2/m^3 fuel] | |
| 1 | 1 | 0 | y_3 | | $2fy_3$ | | | | | |
| $-f$ | | | [m^3N_2/m^3 fuel] | | [m^3NO/m^3 fuel] | | | | | |
| $1-f$ | | | | | | | | | $-fy_3$ [m^3N_2/m^3 fuel] | |
| | | | Remaining O_2 balance | | $(1-e)(y_4+ay_1+by_2)-fy_3$ | | | | [$m^3 O_2/m^3$ fuel] | |
| | | | Remaining N_2 balance | | $(1-d-f)y_3$ | | | | [m^3N_2/m^3 fuel] | |
| $H_2 \rightarrow 2H$ | | | | | | | | | | |
| 1 0 | | | $(2b+c)y_2$ | | $(1-g)(2b+c)y_2$ | | | | $-g(2b+c)y_2$ | |
| $-g$ $2g$ | | | [m^3H_2/m^3 fuel] | | [m^3H_2/m^3 fuel] | | | | [m^3H_2/m^3 fuel] | |
| $1-g$ $2g$ | | | | | $2g(2b+c)y_2$ | | | | [m^3H/m^3 fuel] | |

Table 3

Heat transfer calculus results

| Heat Exchanger | Heat Rate [kW] | $h_{i, convection}$ [$kW/m^2 K$] | $h_{e, convection}$ [$kW/m^2 K$] | $h_{e, r}$ [$kW/m^2 K$] | $h_{e, cr} = h_{e, convection} + h_{e, r}$ [$kW/m^2 K$] | HTS [m^2] | \bar{T}_g [K] | \bar{T}_W [K] | \bar{T}_{wall} [K] |
|----------------|----------------|------------------------------------|------------------------------------|---------------------------|---|---------------|-----------------|-----------------|----------------------|
| E | 86887 | 14.660 | 0.0290 | 0.001716 | 0.030716 | 15979 | 736.026 | 558.04 | 559.0 |
| F | 170000 | 19.594 | 0.0150 | 0.065555 | 0.080555 | 1321 | 2221.036 | 614.67 | 623.5 |
| RH | 205590 | 17.729 | 0.1375 | 0.045600 | 0.183103 | 887.5 | 1966.036 | 683.60 | 700.9 |
| S | 557542 | 23.144 | 0.0965 | 0.006900 | 0.103400 | 10133 | 1260.785 | 725.41 | 728.6 |

2.2. The Heat Transfer Surfaces Calculus

The general equations used in the calculation of the heat transfer surface areas are related to the external heat transfer surface area of the pipes. These equations allow the estimation of the external heat transfer surface area and its mean temperature. In order to solve these equations we calculated (or assumed if necessary) the following :

- the mean velocities of flows;
- the mean path length of radiation through the flue gases volume;
- the thermal conductivity of the pipe material;
- the outer and inner diameters of the pipes and their geometrical arrangement within the bank;
- the parameters of fluids at the mean temperature.

The results of the calculus are summarized in Table 3.

3. THE NEW BOILER DESIGN

The beneficial consequences of the OEC can be evaluated by considering several situations, depending on the initial hypotheses of the analysis. One can imagine a large number of such situations, since the choice of the parameters that can be considered as input data, is a very diversified one. From these situations, the design of a new generation of advanced boilers, using OEC air was considered.

The analysis performed takes into consideration two cases of boilers, different from the initial one from the standpoint of the heat transfer surface areas. The common feature is represented by the heat rates for each heat exchanger, which are imposed to be equal

with those from the Base Case. We consider the following cases:

1. The preheat temperature is equal to the one from the Base Case ($T_{PH} = 590$ K)
2. The preheat temperature results from the heat balance equation of the air preheater.

The heat rates are as follows:

- The overall heat rate of the boiler:

$$\dot{Q}_B = \dot{m}' \cdot (h_{m3} - h_{m6}) + \dot{m}'' \cdot (h_{m5} - h_{m4}) = 1020000 \text{ [kW]}$$

- The heat rate of the economizer:

$$\dot{Q}_E = \dot{m}' \cdot (h_{m1} - h_{m6}) = 86887 \text{ [kW]}$$

- The heat rate of the furnace:

$$\dot{Q}_F = \dot{m}' \cdot (h_{m2} - h_{m1}) = 170000 \text{ [kW]}$$

- The heat rate of the reheater:

$$\dot{Q}_{RH} = \dot{m}'' \cdot (h_{m5} - h_{m4}) = 205590 \text{ [kW]}$$

- The heat rate of the superheater:

$$\dot{Q}_S = \dot{m}' \cdot (h_{m3} - h_{m2}) = 557542 \text{ [kW]}$$

- The heat rate of the air heater:

$$\dot{Q}_{AH} = \dot{V}_f \cdot \lambda_{excess} \cdot \dot{V}_{air}^{\min} \int_{299.82}^{T_{PH}} c_{p,air} \cdot dT$$

where $\lambda_{excess} = 1.07$ is the air excess coefficient, \dot{V}_{air}^{\min} is the minimum necessary air volume, and T_{PH} is the temperature of the preheated air (imposed).

The mean temperatures $T_{1g}, T_{2g}, T_{3g} = T_{4g}, T_{5g}, T_{6g}$, and T_{7g} (after the air heater) in the flow sections 1g, 2g, 3g = 4g, 5g, 6g, and 7g were calculated by the trial-and-error method on the basis of the heat balance

equations of the heat exchangers F, RH, S, and E and of the air heater.

The calculated values of $(T_{jg})_{j=(1 \text{ to } 7)}$ are included in Tables 4 (TPH imposed) and 5 (TPH calculated).

Table 4

| Temperatures for TPH imposed | | | | | | |
|---|---------------------|---------------------|---------------------------------------|---------------------|---------------------|---------------------|
| O ₂ mole fraction in air [%] | T _{1g} [K] | T _{2g} [K] | T _{3g} = T _{4g} [K] | T _{5g} [K] | T _{6g} [K] | T _{7g} [K] |
| 100 | 3077.615 | 2951.615 | 2764.615 | 1252.615 | 782.615 | 632.7795 |
| 75 | 2994.708 | 2863.708 | 2666.708 | 1165.708 | 753.708 | 582.3911 |
| 50 | 2850.247 | 2707.247 | 2487.247 | 1024.247 | 717.247 | 517.143 |
| 35 | 2680.266 | 2521.266 | 2265.266 | 949.266 | 691.266 | 468.9633 |
| 25 | 2466.053 | 2280.053 | 1980.053 | 867.053 | 667.053 | 426.9645 |
| 21 | 2326.256 | 2119.038 | 1820.538 | 829.538 | 649.820 | 401.4800 |

Table 5

| Temperatures for TPH calculated | | | | | | | |
|---|---------------------|---------------------|---------------------------------------|---------------------|---------------------|---------------------|---------------------|
| O ₂ mole fraction in air [%] | T _{1g} [K] | T _{2g} [K] | T _{3g} = T _{4g} [K] | T _{5g} [K] | T _{6g} [K] | T _{7g} [K] | T _{PH} [K] |
| 100 | 3089.899 | 2966.899 | 2784.899 | 1322.899 | 857.899 | 638.665 | 722 |
| 75 | 3007.291 | 2878.291 | 2686.291 | 1219.291 | 812.291 | 579.148 | 693 |
| 50 | 2861.863 | 2721.863 | 2506.863 | 1077.863 | 753.863 | 506.699 | 657 |
| 35 | 2690.495 | 2533.495 | 2281.495 | 978.495 | 724.495 | 471.119 | 631 |
| 25 | 2472.477 | 2288.477 | 1989.477 | 879.477 | 679.477 | 425.332 | 607 |
| 21 | 2326.256 | 2119.038 | 1820.538 | 829.538 | 649.820 | 401.480 | 590 |

4. COMPARISON BETWEEN THE BASE CASE AND THE NEW BOILER

For each of the two cases ($T_{PH} = 590$ K and T_{PH} calculated) we have computed the new values of the heat transfer surface areas of the heat exchangers (economizer E, furnace F, reheater RH and superheater S), by using adapted versions of the computer codes that we had developed and used for the Base Case.

These calculations for a supercritical boiler of 1020 MW thermal power that supplies steam for a turbine – generator group of 350 MWe resulted in the shrinkage of the heat transfer surface areas.

The use of OEC in boilers leads to the following beneficial consequences:

- The increase of the fraction of O₂ in the air rises the temperatures of the flue gases and in this way, the heat transfer surfaces can be reduced, mainly due to the higher mean log temperature differences within the heat exchangers of the boiler.
- The possible growth of the temperature of the preheated oxygen-enriched air can cause an additional increase of the temperatures of the flue gases that amplifies the diminution of the heat transfer surfaces of the boiler.
- From the thermodynamic point of view, since the O₂ fraction in the air increases as the final temperature of the flue gases decreases, the OEC can increase the boiler's efficiency by 2% up to 5%.

Within a boiler, the heat transfer processes will be strongly influenced by these specific features of OEC. Thus, the main modifications that might be taken into consideration are:

- higher mean log temperature differences in the heat transfer processes;

- slightly stronger thermal radiation and convection on the side of the flue gases flow;
- higher temperatures of the preheated air – oxygen mixture;
- diminishing of the unavoidable waste heat in the exhaust flue gases.

In order to control the temperature, the flue gases have to be recycled. In a retrofit situation, the recycle ratio represents the value required to match the heat transfer for the air-fired furnace. The recycle ratio increases with the size of the furnace and is of the order of 3. The mean velocity of gases must be reduced by the factor of 0.84, which leads to a longer residence time in the radiant section of the furnace.

5. CONCLUSIONS

By comparison with the traditional technologies, the oxygen-enriched combustion (OEC) has the following main characteristics:

- Improved combustion, particularly for low-heating value fuels.
- Higher temperatures of the flame, and of the flue gases.
- Larger thermal conductivity of the flue gases.
- Negligible variations of the dynamic viscosity and Prandtl's number of the flue gases at a certain temperature.
- Lower final temperature of the isobaric cooling process of the flue gases for an imposed heat flux.

Negligible variations of the available useful heat per unit quantity of the fuel. The balance of heat transfer between the water-cooled walls, radiant and reheat panels, and convection section will differ slightly be-

tween air and oxy-combustion but can be compensated for by either changes in operation (burner tilt, etc.) for retrofit or design changes for new units.

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CONFERINȚA NAȚIONALĂ A TERMOTEHNICIENILOR

Ediția a 16- a

Prof.dr.ing. Nicolae LEONĂCHESCU

Am participat la CNT- 16 organizată la Universitatea Petrol-Gaze din Ploiești în zilele de 31 mai și 1 iunie 2007, invitat fiind de amabilul nostru coleg în ale Termotehniciei, prof.dr.ing. Cornel TRIFAN și de distinsa colegă, doamna conf.dr.ing. Tudora N. CRISTESCU.

Organizatorii ploieșteni au fost la înălțime din toate punctele de vedere. Am admirat modul elegant în care au fost primiți oaspeții din celelalte centre universitare, inclusiv cafeaua „cu discuții” de la nivelul rectorului, domnul prof.dr.ing. Vlad ULMEANU.

Mi-a făcut o deosebită plăcere să mă întâlnesc cu universitari deosebiți ca: prof. Ioan I. IRIMIE; prof. Victor ZUBCU; prof. Tudor M. SAJIN; dr.ing. Tudor V. CUCIUC; prof. Gabriel D. IVAN; prof. Nicolae I. ANTONESCU; conf. Aneta HAZI; conf. Ioan CĂLDARE; prof. Marin I. BICĂ; prof. Tănase C. PANAIT; prof. Ion C. IONIȚĂ; prof. Stoian Anghel TIMOC; prof. Stoian PETRESCU; prof. Mircea MARINESCU ș.a.

În programul CNT - 16 au fost înscrise 96 lucrări grupate în două secții (Termodinamică, transferul căldurii și aplicații - 48 lucrări; Mașini și instalații termice - 44 lucrări) și două „mese rotunde” (4 lucrări).

A susține toate aceste lucrări în două zile reprezintă, în toate cazurile, o sarcină imposibil de onorat. Organizatorii au soluționat problema prin publicarea tuturor lucrărilor în două volume ale conferinței la Editura Universității Petrol-Gaze din Ploiești.

Răsfoind cele două volume mă bucur să constat diversitatea și calitatea lucrărilor termotehnicienilor români din 2007! Ele sunt „cartea de vizită” a noastră și reprezintă o certitudine că lupta pentru afirmarea poziției TERMOTEHNICII românești n-a fost pierdută. Publicarea acestor lucrări conferă prestigiu colectivului de catedră din Ploiești care, cu maximă responsabilitate care se impune, a înțeles că orice operă ce ne proiectează în eternitate cere sacrificii și eforturi susținute. Un sincer BRAVO!, colegilor noștri din Ploiești!

Modificările din scenariul optim, gândit inițial, privind desfășurarea CNT- 16, au fost impuse de realități. Priorități de ultim moment au condus la absența unor colegi înscriși în program iar dorința unor invitați - membri SRT - de a-și expune punctul de vedere a generat reacții de toate felurile.

Masa festiva a reprezentat un alt prilej de schimb de informații profesionale iar discuțiile dintre colegi au avut substanță pentru că deschiderea spre dialog și spre receptarea informațiilor și a unor puncte de vedere diferite, chiar critice, reprezintă o pecete stilistică a omului cult și modern.

Expunerile din plen au oferit șansa de a fi audiate comunicările unor termotehnicieni recunoscuți prin valoarea cercetărilor lor și anume: dr.ing. Valentin Silivestru, prof.Tănase C. Panait, prof. Ion C. Ioniță, ing. Mirela Dragomir ș.a.

(continuare la pag. 51)