THE DESIGN MAXIMUM POWER CRITERION

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Rezumat. *Criteriul Novikov–Curzon–Albhorn, de putere maximă, nu poate fi utilizat în proiectarea sistemelor termice deoarece a fost dezvoltat pe un sistem termic dat, pentru care s-au neglijat dependenţele operaţionale între diferenţele medii de temperatură, coeficienţii globali de transfer de căldură şi ariile de schimb termic corespunzătoare relaţiilor termice cu sursele de căldură externe. În plus, practica industrială a demonstrat posibilitatea creşterii puterii prin reducerea ireversibilităţii dacă, se folosesc schimbătoare de căldură cu eficienţe termice superioare. Lucrarea propune un nou model de maximizare a puterii în procesul de proiectare a unor sisteme termice moderne, prin intermediul ciclului Carnot endoreversibil şi definirea unor eficienţe termice aferente principiului II al termodinamicii. Sunt prezentate concluzii cu privire la ireversibilitatea şi sensul transferului de căldură real.*

INTRODUCTION – BASIC IDEAL CYCLES

The thermodynamic analysis and optimization is made on the basis of ideal cycles. An ideal cycle is characterized by no entropy generation, $\dot{S}_{gen} = 0$. By this very concise condition, one can originate a lot of ideal cycles. In classical Thermodynamics the ideal cycles come in contact with two external heat sources, see Figure 1.

An ideal complete reversible (internally and externally) cycle exchanges heat with at least two external heat reservoirs (having infinite heat capacity respectively constant temperatures) at infinitesimal temperature differences. There are infinite variants possible to analyze, all of them complying with the same single rule previously considered in defining the operational thermal frame. In Figure 1, are sketched, in temperature – entropy diagram, three particular ideal engine cycles and the wide-ranging one. In accord to Figure 1, all these ideal possible engine cycles consume the heat Q from the hot reservoir having the constant temperature T and remove out the cyclic waste heat Q_0 to the cold reservoir having the constant temperature T_0 (usually the cold reservoir is the environment and so $T_0 = T_{env}$). The cyclic output work is equal to the difference $(Q - Q_0)$. The heat transfers are made at infinitesimal temperature differences dT and dT_0. Excepting the Carnot cycle (1a), for the other ones, the two isothermal processes are linked by two non-adiabatic processes, and they must exchange each other (internal regeneration of the heat) the heat Q_R. The non-adiabatic processes 1 – 2 and 3 – 4 might be either two constant volume processes, or two constant pressure processes, or two any politropic ones. The internal regeneration of the heat must fulfill the following requirements:

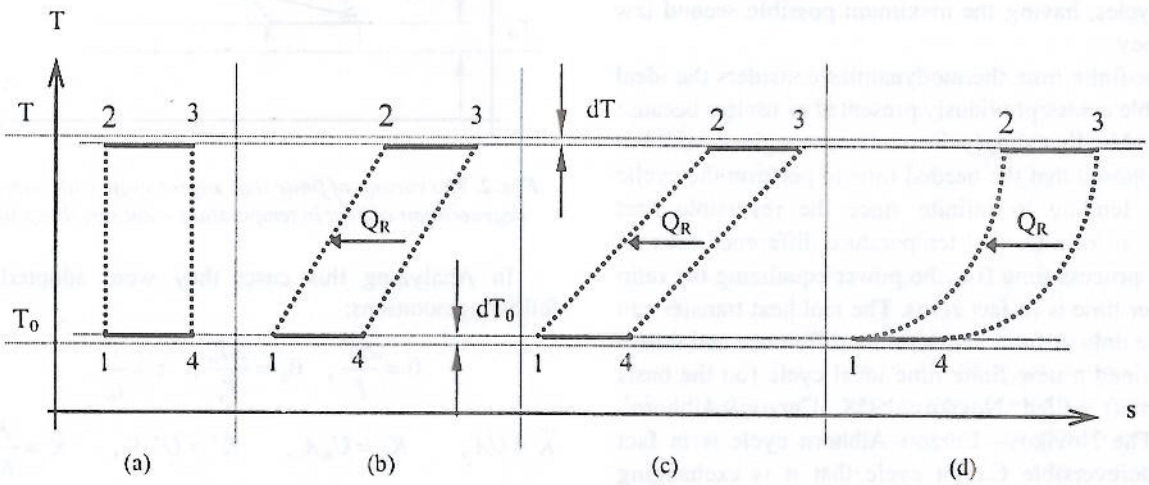

Fig. 1. The possible ideal engine cycle in temperature – entropy diagram:

(a) $s_1 = s_2$; $s_3 = s_4$; (b) $v_1 = v_2$, $v_3 = v_4$; (c) $P_1 = P_2$, $P_3 = P_4$; (d) $c_{1\text{-}2} = \text{const.} = c_{3\text{-}4}$

(see Figure 1b)

$$|Q_R| = mc_v(T_3 - T_4) = mc_v(T_2 - T_1) =$$
$$= mc_v[(T - dT) - (T_0 + dT_0)] = mc_v(T - T_0) \quad (1)$$
$$s_2 - s_1 = s_3 - s_4 = c_v \ln(T/T_0)$$

(see Figure 1c)

$$|Q_R| = mc_p(T_3 - T_4) = mc_p(T_2 - T_1) =$$
$$= mc_p[(T - dT) - (T_0 + dT_0)] = mc_p(T - T_0) \quad (2)$$
$$s_2 - s_1 = s_3 - s_4 = c_p \ln(T/T_0)$$

(see Figure 1d – GENERAL CASE)

$$|Q_R| = mc_n(T_3 - T_4) = mc_n(T_2 - T_1) =$$
$$= mc_n[(T - dT) - (T_0 + dT_0)] = mc_n(T - T_0) \quad (3)$$
$$s_2 - s_1 = s_3 - s_4 = c_n \ln(T/T_0)$$

where c_v and c_p and are c_n the heat capacities at constant volume, constant pressure and for politropic.

It is very easy now to demonstrate for all above considered ideal cycle, that the first law and second law efficiencies are identical.

• First law efficiency

$$\eta_I = \frac{output\ work}{\text{input heat}} = \frac{W}{Q} = 1 - \frac{Q_0}{Q} = 1 - \frac{T_0 + dT_0}{T - dT} = 1 - \frac{T_0}{T} < 1 \quad (4)$$

• Second law efficiency

$$\eta_{II} = \frac{W}{W_{rev}} = 1 - \frac{W_{lost}}{W_{rev}} = 1 - \frac{T_0 S_{gen}}{W_{rev}} = 1 \quad (5)$$

It is noted that the both efficiencies for all entirely reversible cases are maximum possible. Any irreversible (thermally – ΔT and ΔT_0 and ΔT_R finite, and frictionally – internal irreversible flow) engine cycle has both efficiencies smaller.

Development of the second law analysis and optimization methods leans basically upon the complete ideal cycles, having the maximum possible second law efficiency.

The finite time thermodynamics considers the ideal reversible cycles previously presented as useless because they might either supply the maximum engines work but it is supposed that the needed time to perform the cyclic path is tending to infinite since the reversible heat transfer at infinitesimal temperature difference asks an infinite process time (i.e. the power equalizing the ratio work per time is in fact zero). The real heat transfer can be made only at finite temperature difference and thus it was defined a new finite time ideal cycle (on the basis of Carnot) called Novikov–1958, Curzon&Albhorn–1975. The Novikov– Curzon–Albhorn cycle is in fact the endoreversible Carnot cycle that it is exchanging heat with external heat reservoirs at the finite temperature differences $\Delta T = (T - T_M)$ and $\Delta T_0 = (T_m - T_0)$, where T_M is the maximum temperature on the cycle and T_m is the minimum one, and it supplies the maximum real power. Therefore, the finite thermodynamic introduced a new second law-optimizing criterion, respectively the **Maximum Power Generation** that might be used in optimizing thermal systems with a **Minimum Time of Life**. For imposed T, T_0, and supposing any overall heat transfer coefficients and the correspondent heat transfer areas for the heat exchange with hot and cold thermal reservoirs, it results that the output power can be mathematically maximized. The first law efficiency, in this case, becomes:

$$\eta_I = \frac{P}{\dot{Q}} = 1 - \sqrt{\frac{1}{\tau}} \text{ where } \tau = \frac{T}{T_0} \quad (6)$$

Unfortunately, Novikov-Curzon-Albhorn variant of endoreversible Carnot cycle cannot be considered entirely for cycles having an internal heat exchange (regeneration of heat), e.g. Stirling, or all possible alternatives of General ideal one, and as a result we cannot say that for real cycles with internal regeneration of the heat, the ideal maximum power/minimum operation time is Curzon-Albhorn variant of endoreversible Carnot cycle. The same reason might be re-used in order to find the new ideal maximum (power)/(minimum operation time) but taking into account that even the internal regeneration of heat must be completed in finite time and considering also the necessary temperature difference to endorse this heat transfer, see Figure 2.

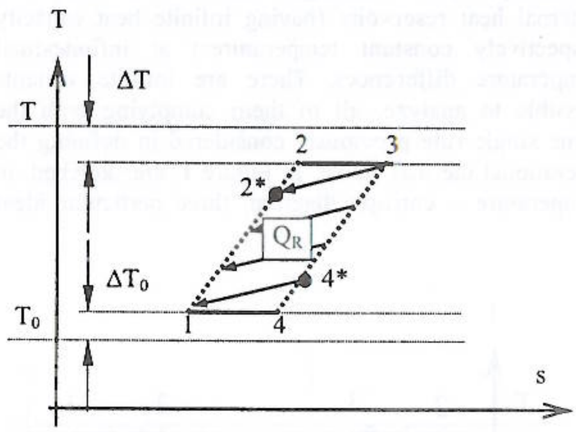

Fig. 2. The variant of finite time engine cycle with internal regeneration of heat in temperature – entropy diagram.

In Analyzing this case, they were adopted the following notations:

$$\theta = \frac{\Delta T}{T}, \quad \theta_0 = \frac{\Delta T_0}{T_0}, \quad \tau = \frac{T}{T_0},$$

$$K = UA, \quad K_0 = U_0 A_0, \quad K^* = U^* A^*, \quad \bar{K} = \frac{K}{K_0}$$

where U, U_0, U^* are the overall heat transfer coefficients for the heat exchange with hot and cold thermal reservoirs and internally made, A, A_0, A^* are the correspondent heat transfer areas.

Analyzing this case, it is yielding:

Input heat

$$\dot{Q} = \dot{Q}_{2-3} + \dot{Q}_{2^*-2} = K\Delta T + K^* \frac{T - \Delta T - T_{2^*}}{\ln\frac{T - T_{2^*}}{\Delta T}} = \tag{7}$$

$$= \dot{m}(T - \Delta T)(s_3 - s_2) + \dot{m}c(T - \Delta T - T_{2^*})$$

Output waste heat

$$\dot{Q}_0 = \dot{Q}_{4-1} + \dot{Q}_{4^*-4} = K_0\Delta T_0 + K_0^* \frac{T_{4^*} - T_0 - \Delta T_0}{\ln\frac{T_{4^*} - T_0}{\Delta T_0}} = \tag{8}$$

$$= \dot{m}(T_0 + \Delta T_0)(s_4 - s_1) + \dot{m}c(T_{4^*} - T_0 - \Delta T_0)$$

Internal heat regeneration – countercurrent balanced heat exchanger

$$\dot{Q}_I = \dot{m}c(T - \Delta T - T_{4^*}) = \dot{m}c(T_{2^*} - T_0 - \Delta T_0) = \\ = \dot{m}c(T - \Delta T - T_0 - \Delta T_0)\varepsilon_I \; ,$$

where $\varepsilon_I = \frac{NTU_I}{NTU_I + 1}$ is the effectiveness of internal heat exchanger

$$\Rightarrow \dot{Q}_{2^*-2} = \dot{Q}_{4^*-4} = \dot{m}c(T - \Delta T - T_0 - \Delta T_0)(1 - \varepsilon_I) = \dot{Q}_S \tag{9}$$

Output power

$$P = \dot{Q} - \dot{Q}_0 = \dot{Q}_{2-3} - \dot{Q}_{4-1} = KT_0\tau\theta\left(1 - \frac{1}{\tau\left(1 - \theta\left(\bar{K} + 1\right)\right)}\right) = \\ = P\big|_{Curzon-Albhorn} \tag{10}$$

We re-get the same relation for output power, and consequently the same θ_{opt}, P_{max}, $\theta_{0,opt}$, but the first law efficiency is lower and obviously depends on overall heat transfer coefficients and, mass flow rate and nature of fluid:

$$\eta_I = \frac{P}{\dot{Q}} = \frac{P}{\dot{Q}_{2-3} + \dot{Q}_S} = \frac{P/\dot{Q}_{2-3}}{1 + \dot{Q}_S/\dot{Q}_{2-3}} =$$

$$\frac{\eta_{Curzon-Albhorn}}{1 + \dot{Q}_S/\dot{Q}_{2-3}} = \left(1 - \sqrt{\frac{1}{\tau}}\right) \Bigg/ \left[1 + \dot{m}c\frac{(1 - \varepsilon_I)\left(\bar{K}\sqrt{\tau} + 1\right)}{\bar{K}\sqrt{\tau}}\right] \tag{11}$$

Remark: The real time internal heat exchange set up a direct gateway between hot and cold heat sinks, $\dot{Q}_{2^*-2} = \dot{Q}_{4^*-4} = \dot{m}c(T - \Delta T - T_0 - \Delta T_0)(1 - \varepsilon_I) = \dot{Q}_S$

THE NEW – MAXIMUM POWER CRITERION

The approach of Novikov–Curzon–Albhorn seems to be not a real second law optimization, it has some unclearness regarding the real link between heat exchange temperature differences, and the overall heat transfer coefficients and the heat transfer areas. Also the industrial practice demonstrated that it is possible to obtain more power by using advanced heat exchangers with higher effectiveness.

In building the New – Maximum Power Criterion, they were identified firstly the second law effectiveness of the endoreversible CARNOT cycle heat exchangers, see Figure 3.

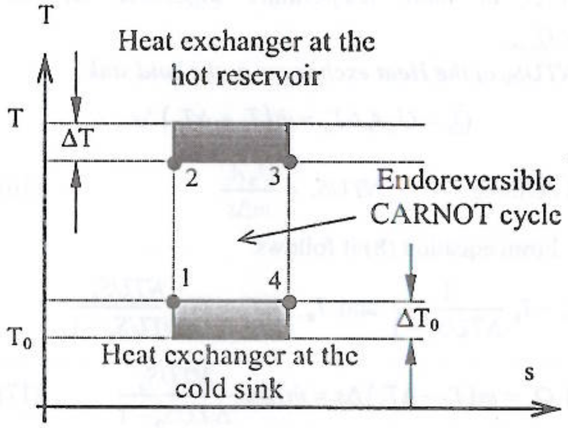

Fig. 3. Scheme of heat exchanges of an endoreversible Carnot cycle.

THE SECOND LAW EFFECTIVENESS OF THE HEAT EXCHANGE AT THE HOT AND COLD RESERVOIR

For the second law effectiveness of this very distinctive heat exchangers (they might be considered acting like a balanced heat exchangers that satisfies simultaneously the constraints of constant temperatures and constant heat flux on both sides) it was introduced the concept of NTUS (number of transfer units per entropy variation rate of the working fluid):

1) NTUS of the Heat exchanger at the hot reservoir

$$\dot{Q} = UA\Delta T = \dot{m}(T - \Delta T)\Delta s$$

and defining $NTUS = \frac{UA}{\dot{m}\Delta s}$ (12)

From equation (12) it results:

$$\Delta T = T\frac{1}{NTUS + 1} \quad \text{and} \quad T - \Delta T = T\frac{NTUS}{NTUS + 1} \quad \text{and}$$

$$\dot{Q} = \dot{m}(T - \Delta T)\Delta s = \dot{m}T\Delta s\frac{NTUS}{NTUS + 1} \tag{13}$$

The effectiveness at the hot source can be set now as:

$$\varepsilon = \frac{\dot{Q}}{\left(\dot{Q}\right)_{NTUS\to\infty}} = \\ = \dot{m}T\Delta s\frac{NTUS}{NTUS + 1}\left(\frac{1}{\dot{m}T\Delta s}\frac{NTUS + 1}{NTUS}\right)_{NTUS\to\infty} ;$$

$$\varepsilon = \frac{NTUS}{NTUS + 1} < 1 \tag{14}$$

In this way, the heat rate at the hot source becomes:

$$\dot{Q} = \varepsilon\dot{Q}_{NTUS\to\infty} = \varepsilon\dot{Q}_{max} = \varepsilon\dot{m}T\Delta s \tag{15}$$

The meaning of $\dot{Q}_{max} = \dot{m}T\Delta s$ *is the maximum reversible heat rate that can be received from the hot source and so the effectiveness,* $\varepsilon < 1$*, in this case reflects the irreversibility caused by the real heat transfer at finite temperature difference* ΔT*, i.e.* $\dot{Q} < \dot{Q}_{max}$.

2) NTUS₀ of the Heat exchanger at the cold sink

$$\dot{Q}_0 = U_0 A_0 \Delta T_0 = \dot{m}\left(T_0 + \Delta T_0\right)\Delta s$$

and defining $$NTUS_0 = \frac{U_0 A_0}{\dot{m}\Delta s} \quad (16)$$

From equation (8) it follows:

$$\Delta T_0 = T_0 \frac{1}{NTUS_0 - 1} \text{ and } T_0 - \Delta T_0 = T_0 \frac{NTUS_0}{NTUS_0 - 1}$$

and $$\dot{Q}_0 = \dot{m}\left(T_0 - \Delta T_0\right)\Delta s = \dot{m}T_0\Delta s \frac{NTUS_0}{NTUS_0 - 1} \quad (17)$$

The effectiveness at the cold sink sets now as:

$$\varepsilon_0 = \frac{\dot{Q}_0}{\left(\dot{Q}_0\right)_{NTUS_0 \to \infty}} = ;$$

$$= \dot{m}T_0\Delta s \frac{NTUS_0}{NTUS_0 - 1}\left(\frac{1}{\dot{m}T_0\Delta s}\frac{NTUS_0 - 1}{NTUS_0}\right)_{NTUS_0 \to \infty}$$

$$\varepsilon_0 = \frac{NTUS_0}{NTUS_0 - 1} > 1 \quad (18)$$

In this way, the heat rate at the hot source becomes:

$$\dot{Q}_0 = \varepsilon_0\left(\dot{Q}_0\right)_{NTUS_0 \to \infty} = \varepsilon_0 \dot{Q}_{min} = \varepsilon \dot{m} T_0 \Delta s \quad (19)$$

The meaning of $\dot{Q}_{min} = \dot{m}T_0\Delta s$ *is the minimum reversible heat rate that can be rejected to the cold sink and so the effectiveness,* $\varepsilon_0 > 1$*, in this case reflects the irreversibility caused by the real heat transfer at finite temperature difference* ΔT*, respectively* $\dot{Q}_0 > \dot{Q}_{min}$.

Figure 4 shows the dependences between the second law effectiveness, NTUS and NTUS₀ and the irreversibility of heat transfers.

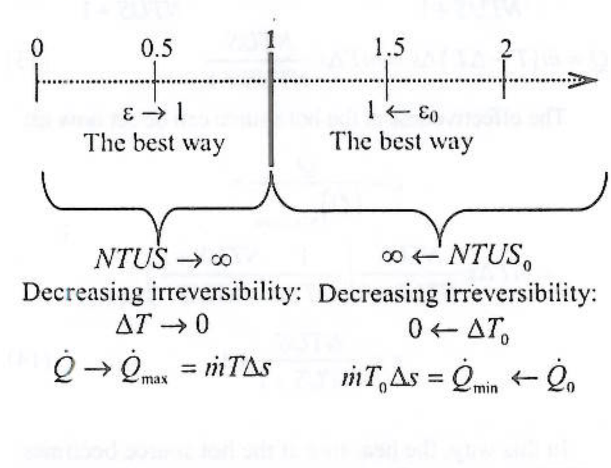

Fig. 4. The dependence second law effectiveness – NTUS, NTUS₀ – irreversibility

This method to describe the second law effectiveness can be applied at any other heat exchanger by the intermediary of the mean thermodynamic temperature of non-adiabatic processes, Baehr – 1973, and the heat transfer temperature differences related to the heat transfer processes.

By combining the previous equations (15) and (19) it finds:

$$P = \dot{Q} - \dot{Q}_0 = \dot{Q}\left(1 - \frac{\dot{Q}_0}{\dot{Q}}\right) = \dot{Q}_{max}\varepsilon_0\left(1 - \frac{\dot{Q}_{min}}{\dot{Q}_{max}}\frac{\varepsilon_0}{\varepsilon}\right) = \dot{m}T\Delta s\varepsilon_0\left(1 - \frac{1}{\tau}\frac{\varepsilon_0}{\varepsilon}\right) \quad (20)$$

Analyzing this expression, it is yielding that the delivered power cannot be optimized mathematically, but can be increased step by step by using new heat exchangers with the second law efficiencies closer and closer to the unity. The first and second law efficiencies for *The New Maximum Power Criterion* analysis are:

$$\eta_I = 1 - \frac{1}{\tau}\frac{\varepsilon_0}{\varepsilon}, \quad \eta_{II} = \frac{1 - \frac{1}{\tau}\frac{\varepsilon_0}{\varepsilon}}{1 - \frac{1}{\tau}} \quad (21)$$

Hence, the first law efficiency of the New – Maximum Power Criterion is non-restrictive, but it shows the ways to come in finite time closer and closer to Carnot machine, see following Figures 5, 6 and 7 that show the dependence of first law efficiency on τ, ε and ε₀.

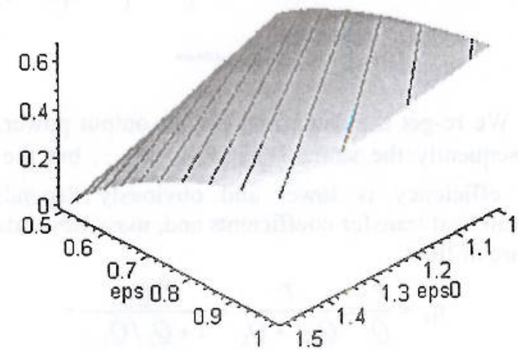

Fig. 5. First law efficiency function of second law effectiveness, $\tau = 3$.

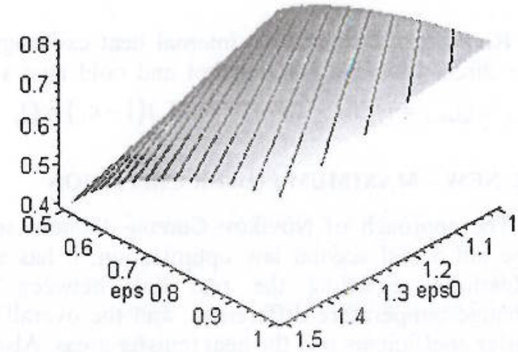

Fig. 6. First law efficiency function of second law effectiveness, $\tau = 5$.

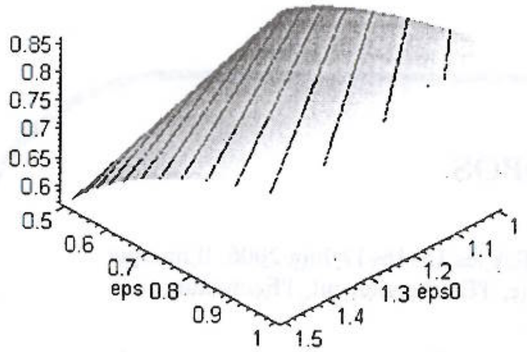

Fig. 7. First law efficiency function of second law effectiveness, $\tau = 7$.

SOME CONCLUSIONS

1. The new design maximum power criterion highlights the main way in building new advanced power systems. The delivered power cannot be optimized mathematically, but can be increased systematically by using new heat exchangers with the second law effectiveness closer and closer to the unity.
2. The second law effectiveness can make the difference between the heat transfers, at the hot and the cold reservoirs. Therefore, the effectiveness at the hot source, $\varepsilon < 1$, reflects the irreversibility caused by the real heat transfer at finite temperature difference, i.e. $\dot{Q} < \dot{Q}_{max}$, and the effectiveness at cold sink, $\varepsilon_0 > 1$, returns the irreversibility caused by the real heat transfer at finite temperature difference, respectively $\dot{Q}_0 > \dot{Q}_{min}$.
3. Since the second law effectiveness also adds up the internal irreversibility by the intermediary of the NTUS and $NTUS_0$, the new maximum power criterion might in fact judge the overall irreversibility, internal and external. The entropy generation caused by the internal irreversibility is put in storage in the entropy variation of the working fluid at the hot and the cold sinks, by $\dot{S} = \dot{m} \cdot \Delta s$ and $\dot{S}_0 = \dot{m} \cdot \Delta s_0$.
4. The real time internal heat exchange set up a direct gateway between hot and cold heat sinks

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AVANT-PROPOS

Le colloque COFRET'06 s'est tenu à TIMIŞOARA les 15–16–17 Juin 2006. Il marque la troisième édition d'un colloque dédié à l'Energie, l'Environnement, l'Economie et la Thermodynamique, un outil privilégié dans ce cadre.

Les travaux rapportés étaient répartis en 10 sections. Il est marquant de voir que la section 5 Energie – Ecologie et Développement Durable a connu le plus de propositions, marquant ainsi l'émergence et la sensibilité du moment ; elle est suivie de la section 8 Matériaux Composites et Recyclages, qui émarge de la même philosophie.

Viennent ensuite la rubrique 10 Energie Renouvelable et Biomasse, à égalité avec les rubriques 6 Transfert de Chaleur et de Masse et 3 Moteurs et Turbines ; ces dernières rubriques plus classiques marquent une rémanence de préoccupations toujours actuelles, de même que les Machines à froid et Pompes à chaleur (rubrique 4).

Au total, 136 communications ont été acceptées pour présentation soit orale, soit sous forme de poster.

Vu le succès de cette manifestation, couplée au Workshop NETBIOCOF (Renewable Energy Resources dans le cadre du 6° PCRD), un prolongement a été décidé pour valoriser les meilleurs travaux présentés dans des revues scientifiques. Avec l'aide du comité scientifique et d'experts français et roumains, une sélection a été faite qui conduit à 15 articles retenus pour publication dans deux numéros de TERMOTEHNICA.

La valorisation doit se poursuivre dans d'autres journaux ; les démarches sont en cours. On remarquera toutefois qu'il devient difficile de publier des numéros spéciaux suite à des manifestations. La politique éditoriale ne semble pas favorable à cette démarche, qui est celle de Termotehnica : que ce journal en soit remercié ici.

Nous profitons de cette tribune pour remercier les collègues roumains qui se sont investis aussi dans cette aventure commune, et tout particulièrement I. IONEL. Nous n'oublions pas le support important de l'ADEME qui nous soutient sans faille depuis de nombreuses années.

Nous espérons que l'action du réseau EURECO va encore s'élargir tant thématiquement que géographiquement et que le rendez-vous évoqué pour 2008 à Nantes tiendra ses promesses et marquera une ouverture industrielle forte, nécessaire à demain.

Dans cette attente, que la fin de l'année 2006 soit l'occasion d'un répit et d'une réflexion, permettant un nouveau départ en 2007 (surtout avec l'intégration européenne de la Roumanie).

Nancy, le 28 Novembre 2006 — Michel FEIDT