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ENTROPY GENERATION IN THE COMPRESSION REFRIGERATION SYSTEMS IN TRANSIENT REGIMES

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Abstract: *This paper presents the calculation methodology for the entropy generation in the apparatus of a vapor compression refrigeration system. The entropy generation under the perturbation influences (as the deactivation of a piston, the modification of the temperature and of the mass flow rate for the cooled fluid) was calculated and interpreted based on experimental data obtained by the measuring and control equipment.*

Key words: *entropy generation, refrigeration, transient regime.*

1. Introduction

The generation of entropy due to the irreversible thermodynamic processes from an installation represents a measure of the energy losses in each part and also in the entire system. The intensity of the entropy generated flux indicates the zones that can be improved as design and functioning. The analysis based on the second law of thermodynamics represents a common investigation instrument of the performances of thermal and refrigeration systems. Thus, this paper presents an application of this analysis applied on the transient (non-stationary) regimes of functioning for a refrigeration system, based on experimental data obtained from the EDF (Electricité de France) Research Institute.

The researchers of this institute conducted complete series of experiments in the purpose to observe the evolution of a refrigeration system in transient regimes induced by the action of different perturbation influences on an apparatus, keeping constants the other apparatus parameters of functioning.

2. Scheme of the Tested Refrigeration System

The refrigeration system with the principal points and measuring devices (manometers, thermometers, debitmeters) is represented in the Fig.1.

The symbols used in the Fig.1 are:

- **Apparatus**:

Cp-reciprocating compressor; SH – oil separator; Cd- condenser; B – tank of liquid refrigerant; Sr – sub-cooler; VL – expander (thermostatic throttle valve); Vp – Evaporator; Bac – liquid-vapor separator.

- **Measuring devices:**

p – for pressure (manometer), T – for temperature (thermometer); $\dot{m}$ - for the mass flow rate (debitmeter).

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The subscripts "e" and "s" indicate the entrance and the exit of the apparatus.

Some technical characteristics of the system:

- Compressor SABROE HPO 24 with 4 cylinders and volumetric capacity 97 m3/h at 1500 rot/min
- Multi-tubular condenser COCKC 271904 with external refrigerant circulation
- Multi-tubular subcooler OWSG 161511 with external refrigerant circulation
- Multi-tubular evaporator EIKK 214037 with internal refrigerant circulation
- Electrical motor power 37 kW
- Refrigeration power: 22 …160 kW
- COP = 2 … 6 (as a function of working regime)
- Evaporation temperature: - 35 °C …+20 °C
- Condensation temperature: + 35 °C … + 45 °C

One used as refrigerant R410a (mixture 50% R32+ 50 % R125) to cool glycolated water named „mixigel".

The system was taken out of the stationary regime of functioning by several perturbations and the thermodynamic parameters were measured at each 5 seconds and registered in series of experimental data. We analyzed in this paper the influence of the following perturbations:

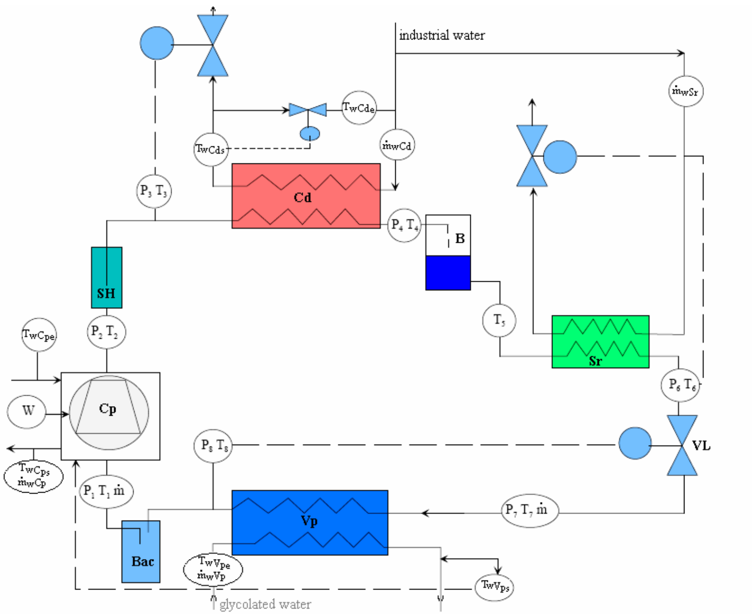

Fig 1. *The scheme of the compression refrigeration systems*

- deactivation of a piston having as consequence the diminution of the volumetric flow rate on the compressor suction line - **experimental data series I**
- augmentation of the refrigeration power demand with increasing temperature of the glycolated water at the entrance in the evaporator – **experimental data series II**
- diminution of the mass flow rate of the glycolated water – **experimental data series III.**

3. Entropy Generation the Transient Regime

The usual studies on transient regimes are using energy balance equation, heat transfer laws and movement equations to predict the variation of the thermodynamic parameters [2-4]. Because our study pursues the calculation based of experimental data of the variation of the generated generation as a measure of the thermodynamic irreversibility, one used the general exergy balance equation [1]:

$$\frac{d(U-T_oS)}{d\tau}=\sum\left(1-\frac{T_o}{T_i}\right)\dot{Q}_i-\dot{W}+ \\ +\sum_e \dot{m}(h-T_oS)-\sum_s \dot{m}(h-T_oS)-T_o\dot{S}_{gen} \quad (1)$$

with τ [s] – the time.

This equation was adapted to each apparatus as follows:

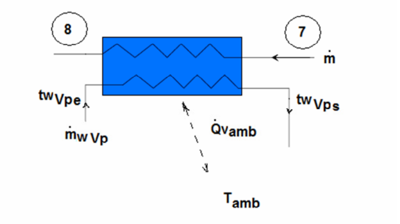

Fig.2. *Evaporator – Vp*

$$\frac{\Delta U_{M+R}-T_o\Delta S_{M+R}}{\Delta\tau}=\left(1-\frac{T_o}{T_{amb}}\right)\dot{Q}_{vamb} \\ +\dot{m}[h_7-h_8-T_o(s_7-s_8)]+ \\ +\dot{m}_{wVp}\cdot c_{mix}\cdot[T_{wVpe}-T_{wVps} \\ -T_o\cdot\ln\frac{T_{wVpe}}{T_{wVps}}]-T_o\cdot\dot{S}_{genVp} \quad (2)$$

with $\quad T_o\cong T_{amb}\Rightarrow\left(1-\frac{T_o}{T_{amb}}\right)\dot{Q}_{vamb}\cong 0$

$M+R$ [kg] - evaporator metal and refrigerant masses

h [kJ/kg] – specific enthalpy of the refrigerant; s [kJ/kg.K] – specific entropy of the refrigerant

c_{mix} [kJ/kg.K] – specific heat of the mixigel (glycolated water)

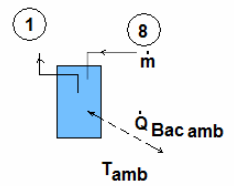

Fig. 3. *Liquid separator – Bac*

$$\frac{\Delta U_{M+R}-T_o\Delta S_{M+R}}{\Delta\tau}=\left(1-\frac{T_o}{T_{amb}}\right)\dot{Q}_{Bac,amb}+ \\ +\dot{m}[h_8-h_1-T_o(s_8-s_1)]-T_o\cdot\dot{S}_{genBac} \quad (3)$$

with $\quad T_o\cong T_{amb}\Rightarrow\left(1-\frac{T_o}{T_{amb}}\right)\dot{Q}_{Bac,amb}\cong 0$

$M+R$ [kg] – separator metal and refrigerant masses.

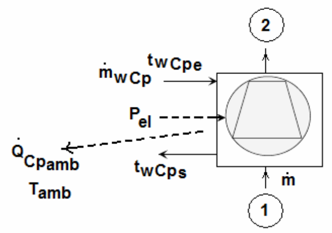

Fig.4. *Compressor - Cp*

$$\frac{\Delta U_M-T_o\Delta S_M}{\Delta\tau}=\left(1-\frac{T_o}{T_{amb}}\right)\dot{Q}_{Cp,amb}+P_{el} \\ +\dot{m}[h_1-h_2-T_o(s_1-s_2)] \\ +\dot{m}_{wCp}\cdot c_w\cdot[T_{wCpe}-T_{wCps} \\ -T_o\cdot\ln\frac{T_{wCpe}}{T_{wCps}}]-T_o\cdot\dot{S}_{genCp} \quad (4)$$

with $T_o\cong T_{amb}\Rightarrow\left(1-\frac{T_o}{T_{amb}}\right)\dot{Q}_{Cp,amb}\cong 0$

P_{el} [kW] – electrical power consumed by the motor that move the compressor.

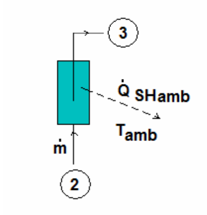

Fig.5. *Oil separator - SH*

$$\frac{\Delta U_{M+R} - T_o \Delta S_{M+R}}{\Delta\tau} = \left(1 - \frac{T_o}{T_{amb}}\right)\dot{Q}_{SH,amb} \qquad (5)$$
$$+\dot{m}[h_2 - h_3 - T_o(s_2 - s_3)] - T_o \cdot \dot{S}_{genSH}$$

with $T_o \cong T_{amb} \Rightarrow \left(1 - \dfrac{T_o}{T_{amb}}\right)\dot{Q}_{Bac,amb} \cong 0$

M+R [kg] - metal and refrigerant masses in the oil separator.

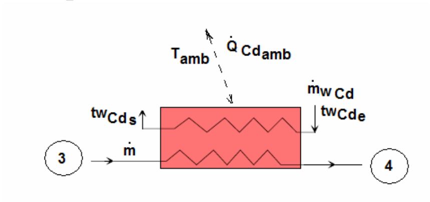

Fig.6. C*ondenser - Cd*

$$\frac{\Delta U_{M+R} - T_o \Delta S_{M+R}}{\Delta\tau} = \left(1 - \frac{T_o}{T_{amb}}\right)\dot{Q}_{SH,amb} \qquad (6)$$
$$+\dot{m}[h_2 - h_3 - T_o(s_2 - s_3)] - T_o \cdot \dot{S}_{genSH}$$
$$+\dot{m}_{wCd} \cdot c_w \cdot [T_{wCde} - T_{wCds}$$
$$-T_o \cdot \ln\frac{T_{wCde}}{T_{wCds}}] - T_o \cdot \dot{S}_{genCd}$$

with $T_o \cong T_{amb} \Rightarrow \left(1 - \dfrac{T_o}{T_{amb}}\right)\dot{Q}_{Cd,amb} \cong 0$

M+R [kg] – condenser metal and refrigerant masses.

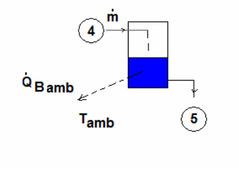

Fig.7. *Tank of liquid refrigerant - b*

$$\frac{\Delta U_{M+R} - T_o \Delta S_{M+R}}{\Delta\tau} = \left(1 - \frac{T_o}{T_{amb}}\right)\dot{Q}_{B,amb} \qquad (7)$$
$$+\dot{m}[h_4 - h_5 - T_o(s_4 - s_5)] - T_o \cdot \dot{S}_{genB}$$

with $T_o \cong T_{amb} \Rightarrow \left(1 - \dfrac{T_o}{T_{amb}}\right)\dot{Q}_{B,amb} \cong 0$

M+R [kg] - metal and refrigerant masses of the tank.

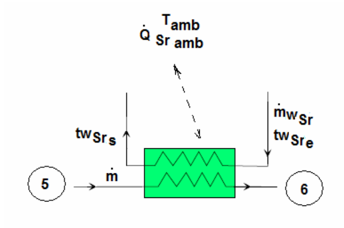

Fig.8. S*ub-cooler - Sr*

$$\frac{\Delta U_{M+R} - T_o \Delta S_{M+R}}{\Delta\tau} = \left(1 - \frac{T_o}{T_{amb}}\right)\dot{Q}_{B,amb} \qquad (8)$$
$$+\dot{m}[h_4 - h_5 - T_o(s_4 - s_5)] - T_o \cdot \dot{S}_{genB}$$
$$+\dot{m}[h_5 - h_6 - T_o(s_5 - s_6)]$$
$$+\dot{m}_{wSr} \cdot c_w \cdot [T_{wSre} - T_{wSrs}$$
$$-T_o \cdot \ln\frac{T_{wSre}}{T_{wSrs}}] - T_o \cdot \dot{S}_{genSr}$$

with $T_o \cong T_{amb} \Rightarrow \left(1 - \dfrac{T_o}{T_{amb}}\right)\dot{Q}_{Sr,amb} \cong 0$

M+R [kg] – subcooler metal and refrigerant masses.

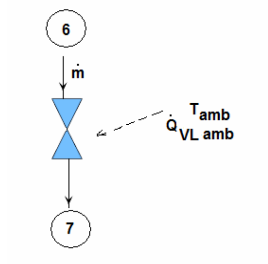

Fig.9. *Expander - VL*

$$\frac{\Delta U_M - T_o \Delta S_M}{\Delta\tau} = \left(1-\frac{T_o}{T_{amb}}\right)\dot{Q}_{VL,amb} - \dot{m}\cdot T_o(s_6 - s_7) - T_o\cdot \dot{S}_{genVL} \qquad (9)$$

One neglected m_R (the refrigerant mass inside the apparatus) in the case of compressor and of the expander because the transit speed of the refrigerant in this apparatus is much higher than in the other.

4. Results and Conclusions

The EES (Engineering Equation Solver) computational had as result graphics that offer a view on the variation of the entropy generation in time.

One will present and discuss this variation in the most important apparatus of the installation (compressor, condenser, sub-cooler, expander) because the calculation shows that the intensity of their fluxes of generated entropy is much higher than in the oil separator, and the liquid-vapor separator.

4.1. Series I – Deactivation of a Piston

The variation of the flux of generated entropy is according to Fig. 10…15.

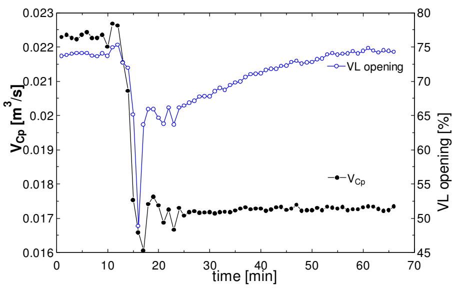

Fig. 10. *Perturbation series I*

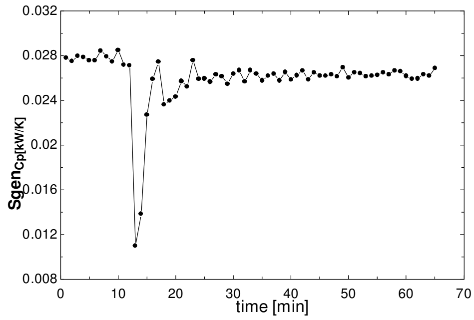

Fig.11. *Compressor entropy generation*

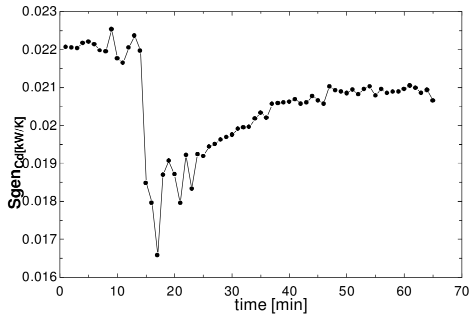

Fig.12. *Condenser entropy generation*

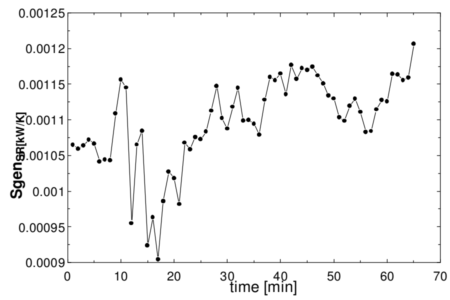

Fig.13. *Subcooler entropy generation*

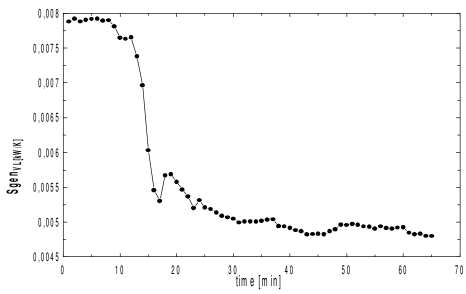

Fig.14. *Expander entropy generation*

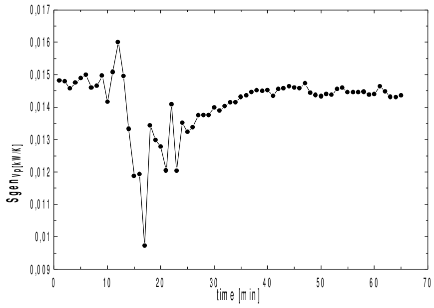

Fig.15. *Evaporator entropy generation*

One notices that the abrupt diminution of the volumetric flow rate (Fig.1) of the refrigerant has as first reaction the closing, then the opening of the opening of the orifice of the expander that adjust the quantity of refrigerant sent to the evaporator.

A first abrupt diminution of the entropy generation in all the apparatus is followed by its increasing (except the expander where the entropy continues to diminish) in time. Globally in the entire system, the flux of entropy increases to a values less high than the initial value.

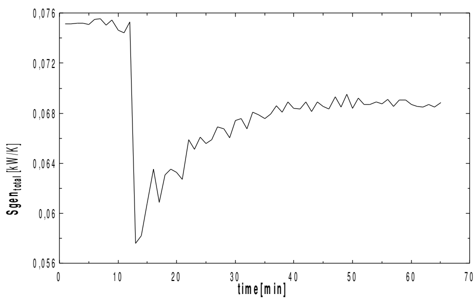

Fig.16. *Total entropy generation*

4.2. Series II – Augmentation of the Refrigeration Power Demand

The variation of the flux of generated entropy is according to Fig. 17…23.

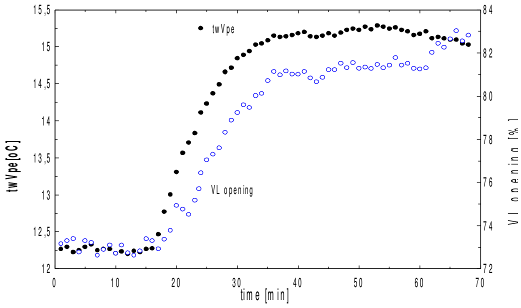

Fig.17. *Perturbation series II*

An increasing refrigeration demand is transmitted to the installation by the augmentation of the temperature of the "mixigel" at the entrance in the evaporator. The expander responds immediately by a larger opening of the orifice (Fig.17).

This perturbation leads to a higher intensity of the irreversibility in each apparatus (except the expander where it is decreasing) and in the entire system (Fig.18…21).

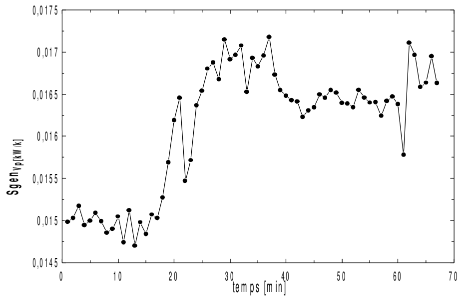

Fig.18. *Evaporator entropy generation*

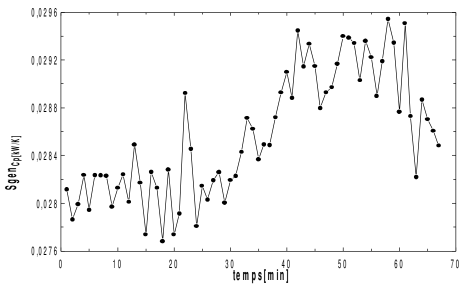

Fig.19. *Compressor entropy generation*

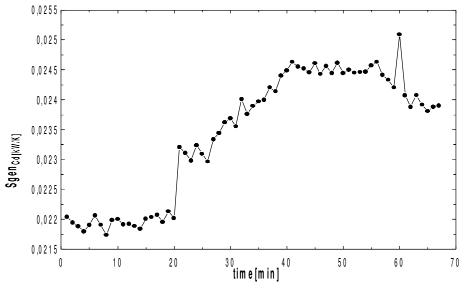

Fig.20. *Condenser entropy generation*

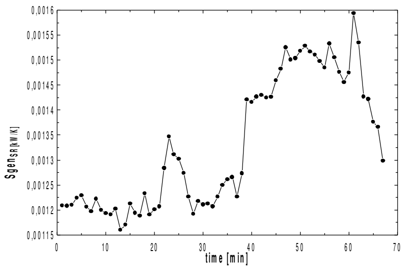

Fig. 21. *Subcooler entropy generation*

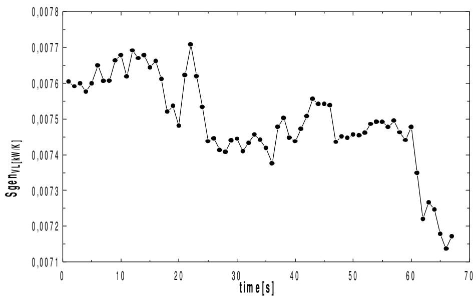

Fig.22. *Expander entropy generation*

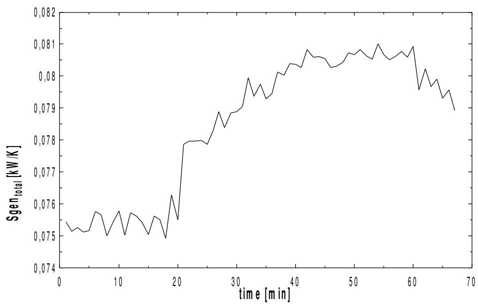

Fig.23. *Total entropy generation*

4.3. Series III – Diminution of the Volumetric Flow Rate of the Glycolated Water

The variation of the flux of generated entropy is according to Fig. 24...30.

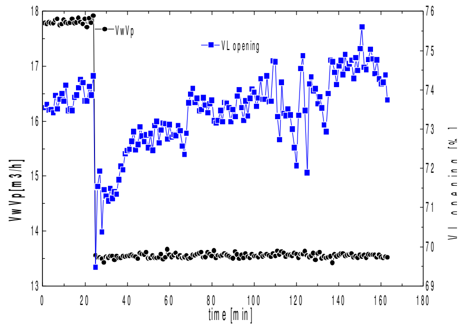

Fig.24. *Perturbation*

In this case, the abrupt diminution of the volumetric flow rate of the cooled glycolated water (Fig.24) has as first reaction the abrupt closing, then the opening of the orifice of the expander to a bigger value than the initial one.

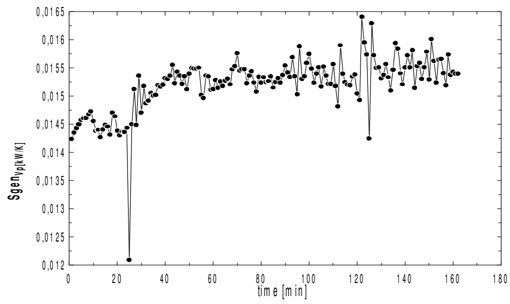

Fig.25. *Evaporator entropy generation*

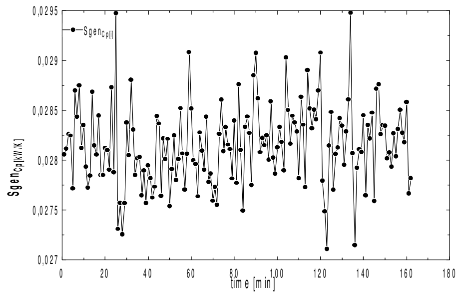

Fig.26. *Compressor entropy generation*

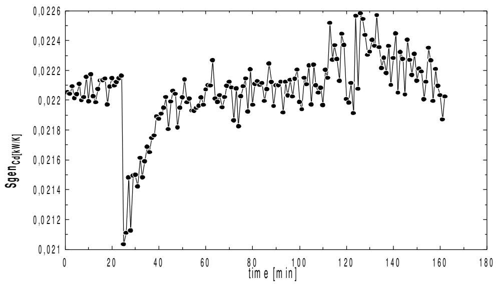

Fig.27. *Condenser entropy generation*

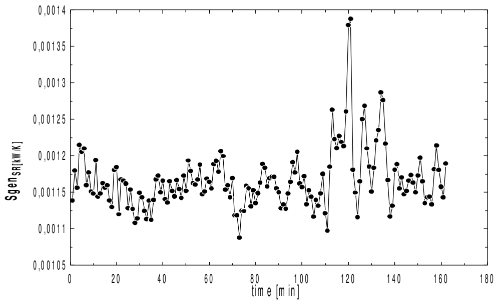

Fig.28. *Subcooler entropy generation*

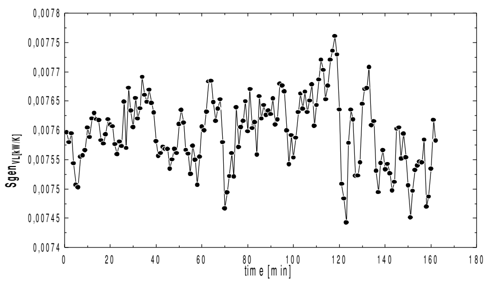

Fig.29. *Expander entropy generation*

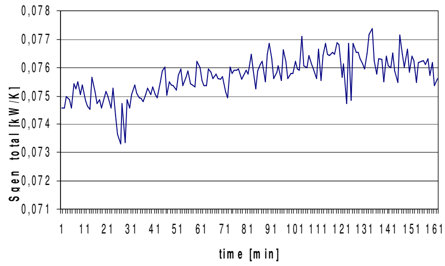

Fig.30. *Global entropy generation*

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