STATUS OF SCO_2 POWER CYCLE STUDIES AT CEA

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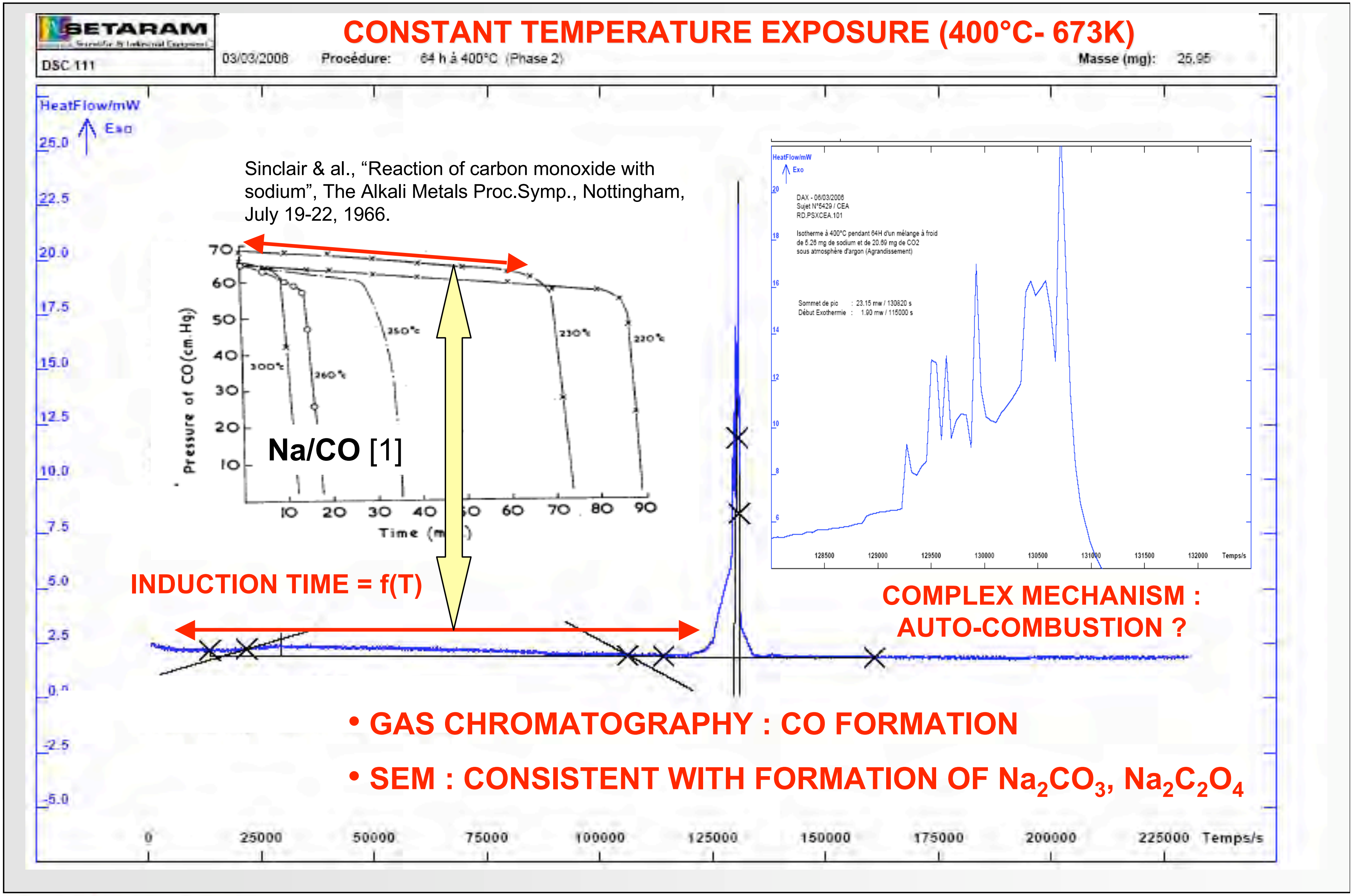
OUTLINES

- Na/SCO_2 chemical interaction
- Cycle thermodynamics: He/SCO_2 & Na/SCO_2
- Cycle components:
  - Intermediate heat exchanger
  - Turbomachinery
- Cycle operation:
  - Part load following
  - Cold sink issue
- Conclusion

Na/CO_2 CHEMICAL INTERACTION : *calorimetric experiments*



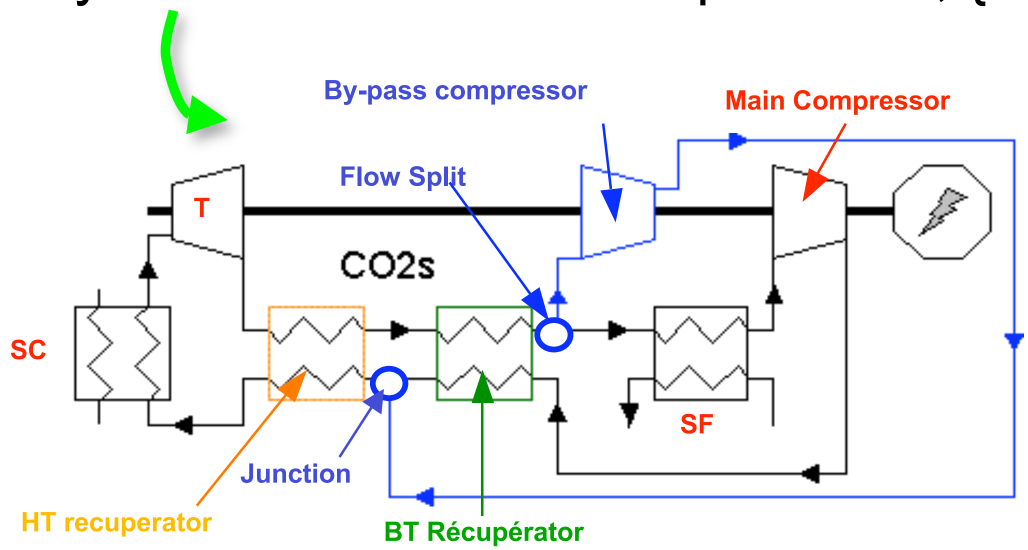
Na/CO_2 CHEMICAL INTERACTION : *results & prospects*

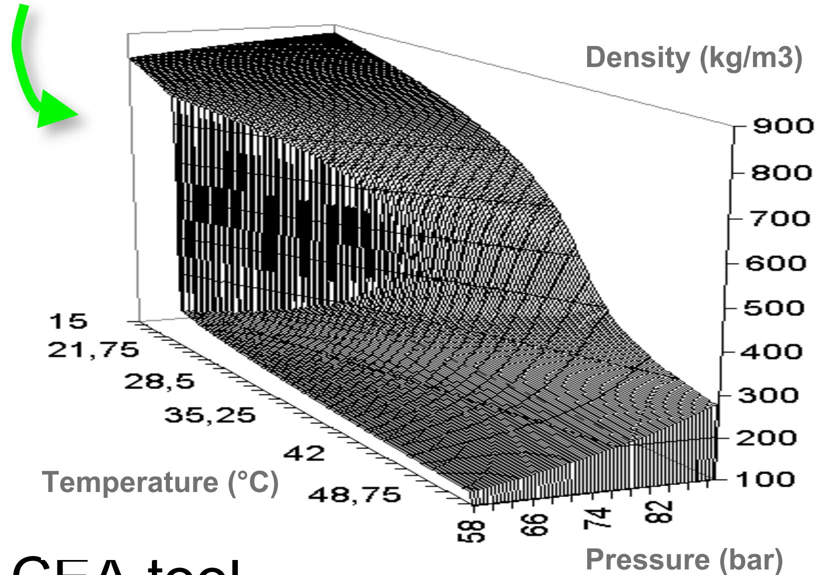
- **PRESENT VIEW OF REACTION SCHEME** *(to be published*)*:
  - – T < 400-500°C (773K) ; complex scenario ; kinetically controlled ;
    - Carbonate & oxalate formation
    - Na/oxalate & oxalate decomposition (CO release)
    - Na/CO reaction (induction time) ;
    - By-products : CO ; Na_2CO_3 ; $Na_2C_2O_4$; C ; Na_2O ; NaCO ; $Na_2C_6O_6$.
  - – T > 500°C (773K) ; no more induction, fast global reaction :
    $2Na + 1.5CO_2 \rightarrow Na_2CO_3 + 0.5C$
- **UNDERTAKEN ACTIONS :**
  - – Study the interaction in more representative conditions :
    direct injection ↑ P in dynamic Na → knowledge of kinetics &
    $\Delta H_{reaction}$ (assessment of of a "westage" scenario occurrence)
  - – Particles issue (carbonate): significant dissolution or trapping ?
  - – Reaction detection systems : efficiency & reliability

**N. Simon & al., "Investigation of sodium-carbon dioxide interactions with calorimetric studies", ICAPP'07, Nice, May 13-18, 2007.*

CYCLE THERMODYNAMICS : *data & tools*

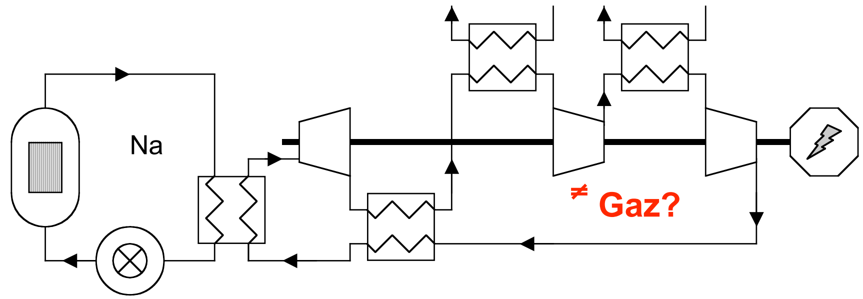
- Cycle architecture: recompression ; {H, S, ρ, ..} data: Span eq. of state





- Mass and energy balances: CYCLOP, CEA tool.
- Optimisation: genetic algorithms, adapted to multi-parameters problem
- Hypothesis for cycle efficiency calculations:
  - Electrical and mechanical losses: 2% alternator, 1.3% shaft
  - TM : η_T = 93% & η_C 88%
  - HE : $\eta_{recuperator}$ = 90% ; η_{IHX} ~ 88% (ΔT = 30°C)
  - Cold cycle operating point = 21°C (294K)

CYCLE THERMODYNAMICS : *Na/Brayton cycle η vs. gas type*

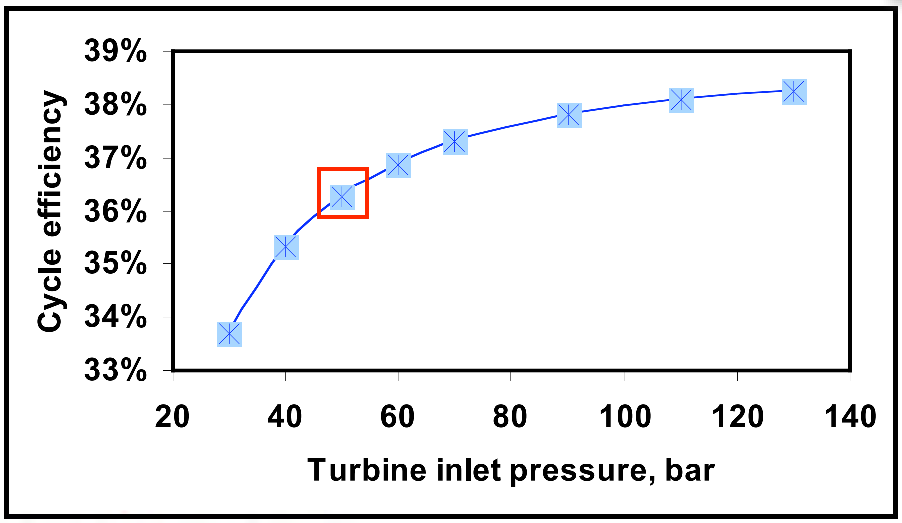


T_{out} core = 550°C (823K) ; T_{in} turb = 520°C ; P_{in} turb = 50 bar (5MPa)

Gas	*T_{in} core*	*Pressure ratio*	*Cycle η*
He	395°C	x 1.7	35.2%
He-N2	395°C	x 2.0	35.7%
N2	**395°C**	**x 2.1**	**36.4%**

CYCLOP

36.4 ↔ 40% Na / steam Rankine cycle (18MPa)



+ 2% if Psec ➔ 120 b ; + 3% ➔ 250 b (25MPa)
+ 0.5% if $\eta_{compressor}$ +1%
+ 0.5% if $\eta_{turbine}$ +1%
+ 0.7% if $\eta_{recuperator}$ +1%

CYCLE THERMODYNAMICS : *Na/SCO_2*

$POWER_{core}$ = 600 MWth

$P_{in\ turbine}$ = 250 bar (25MPa)

$T_{in\ turbine}$= 520°C (793K)

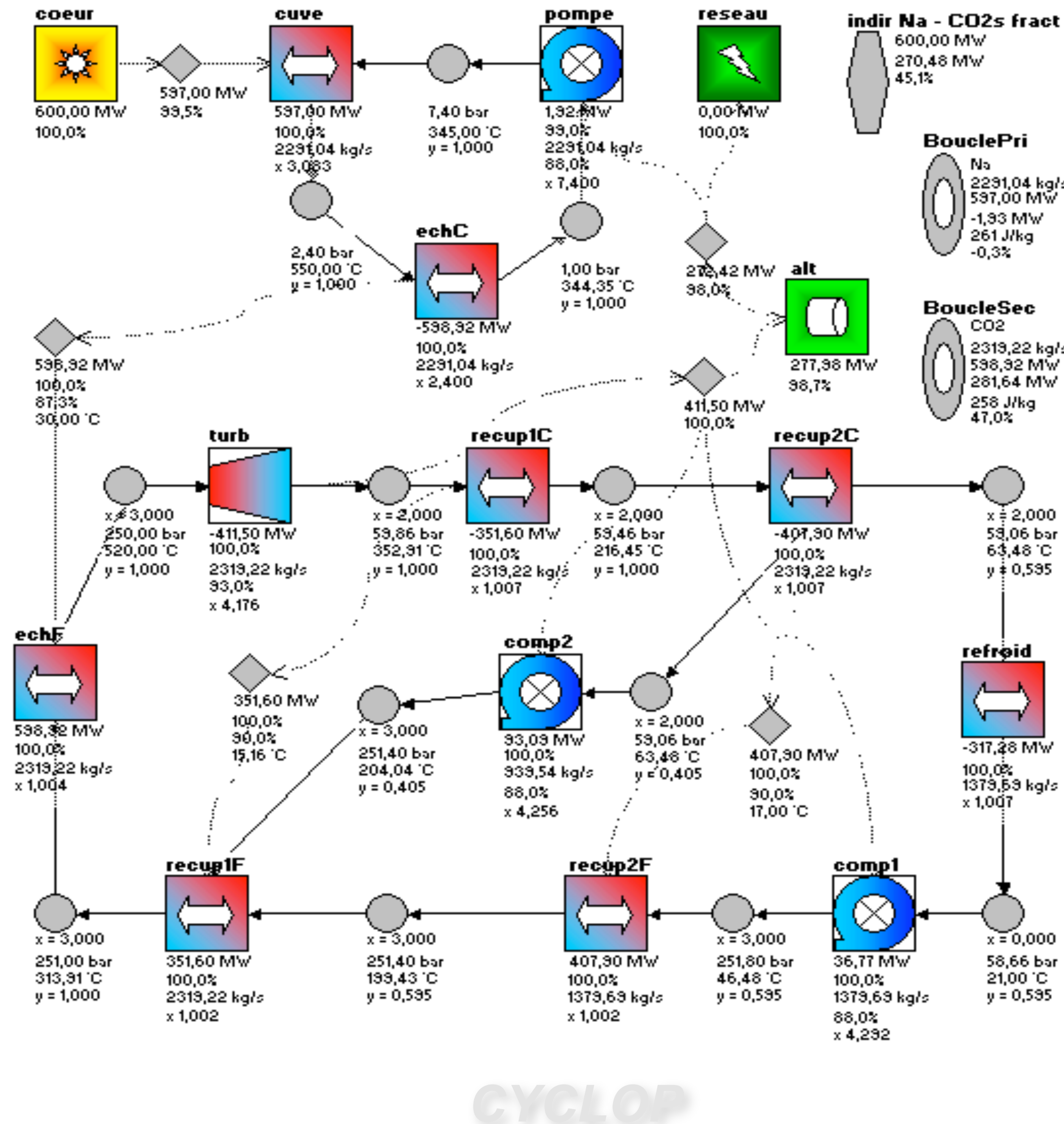
P = 76.9 bar (7.69 MPa)
$T_{out,\ cold\ sink}$ = 32°C (305K)

→ Cycle efficiency = 41.3%

Psat = 59 bar (5.9 MPa)
$T_{out,\ cold\ sink}$ = 21°C (294K)

→ Cycle efficiency = 45.1%

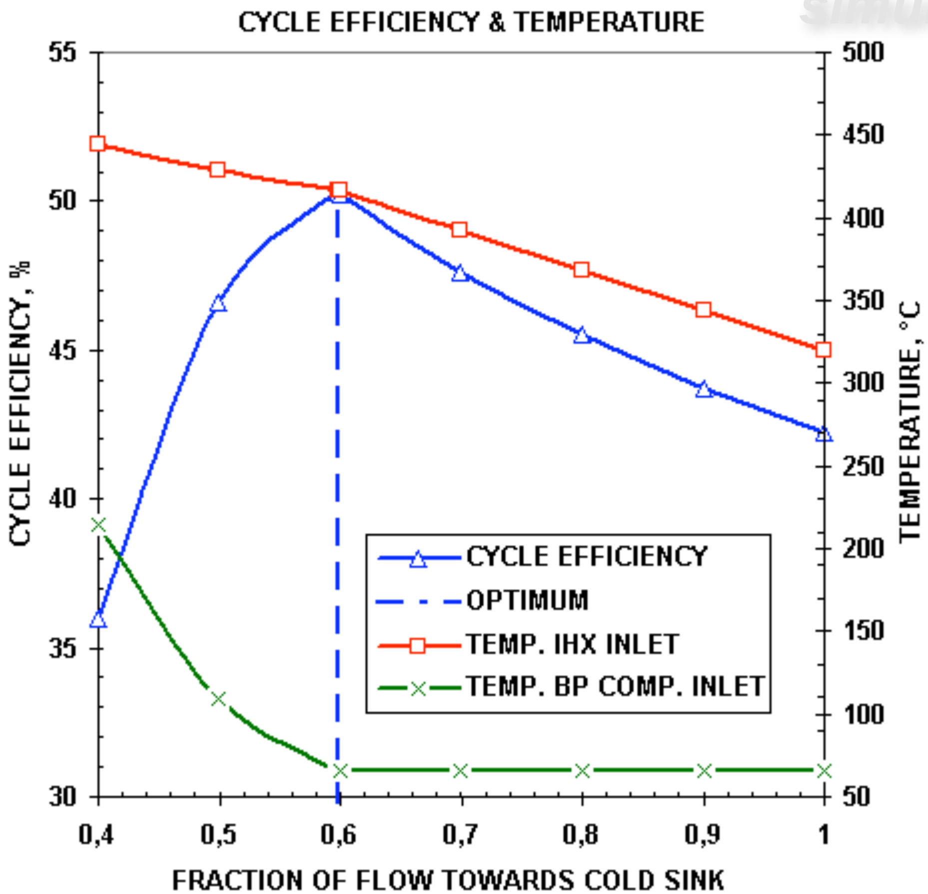
- *$T_{in\ turbine}$= 620°C*
  → η_{Cycle} = 49.1%
- *$P_{in\ turbine}$= 200 & 160 bar*
  → η_{Cycle} = 44.4 & 41.7%

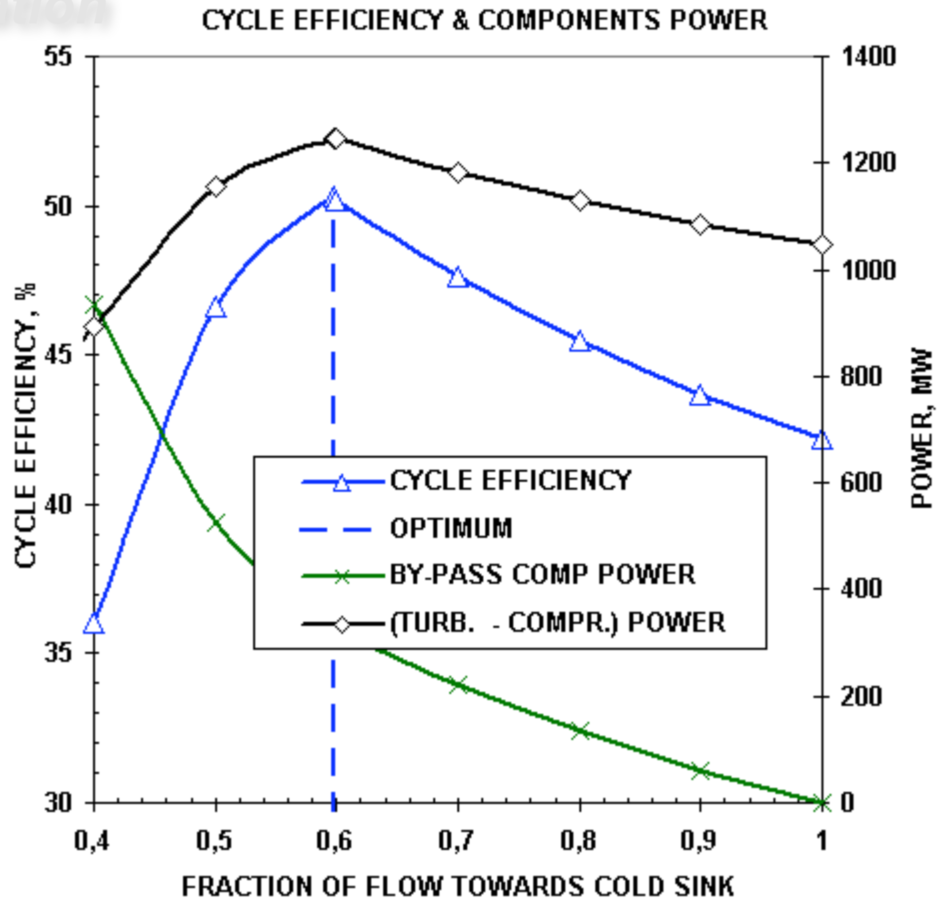


Ref - Na/N_2 : η < 40%
$T_{in\ turb}$ 520°C & P ≤ 250 bar

CYCLE THERMODYNAMICS : *optimal flow split analysis*

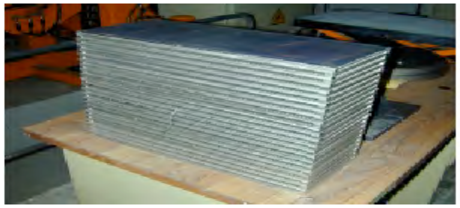
GFR simulation

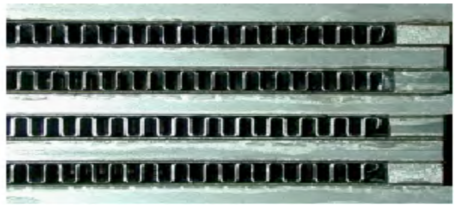


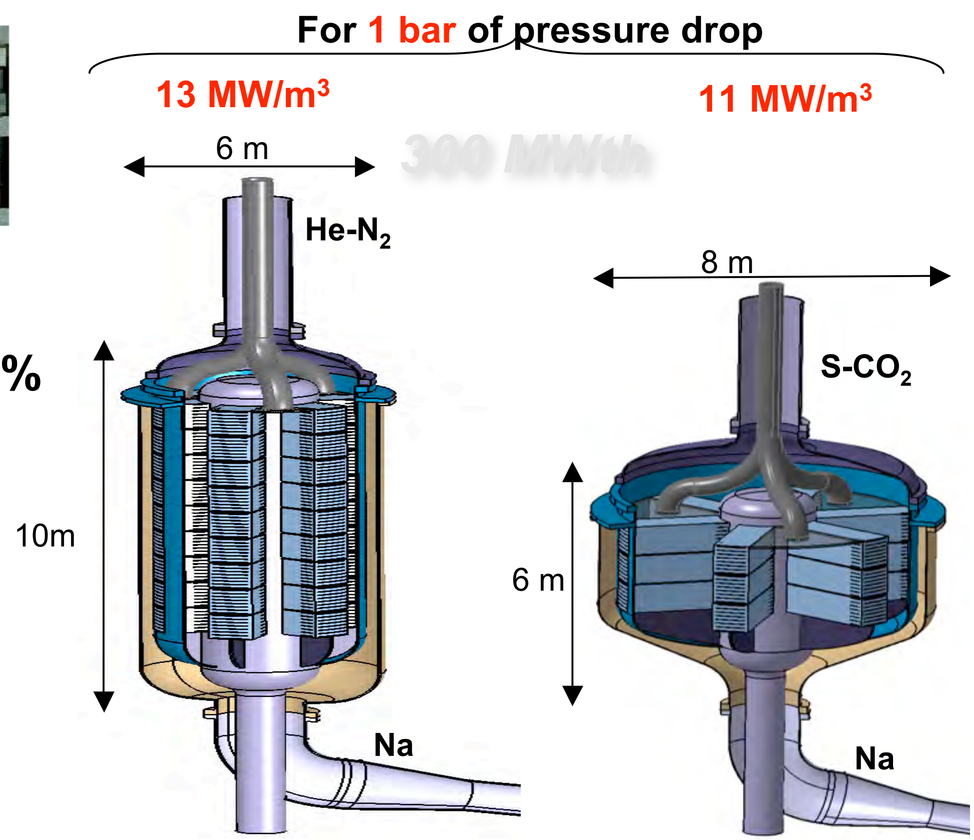


- **Split: ↑ T_{inlet} SC ⇒ $\overline{T_{SC}}$ ↑ ⇒ η cycle ↑**
- **10% from optimal fraction ⇒ ↓ cycle efficiency of ~ 4 points.**

CYCLE COMPONENTS : *Na/SCO_2 & Na/N_2-He heat exchanger*



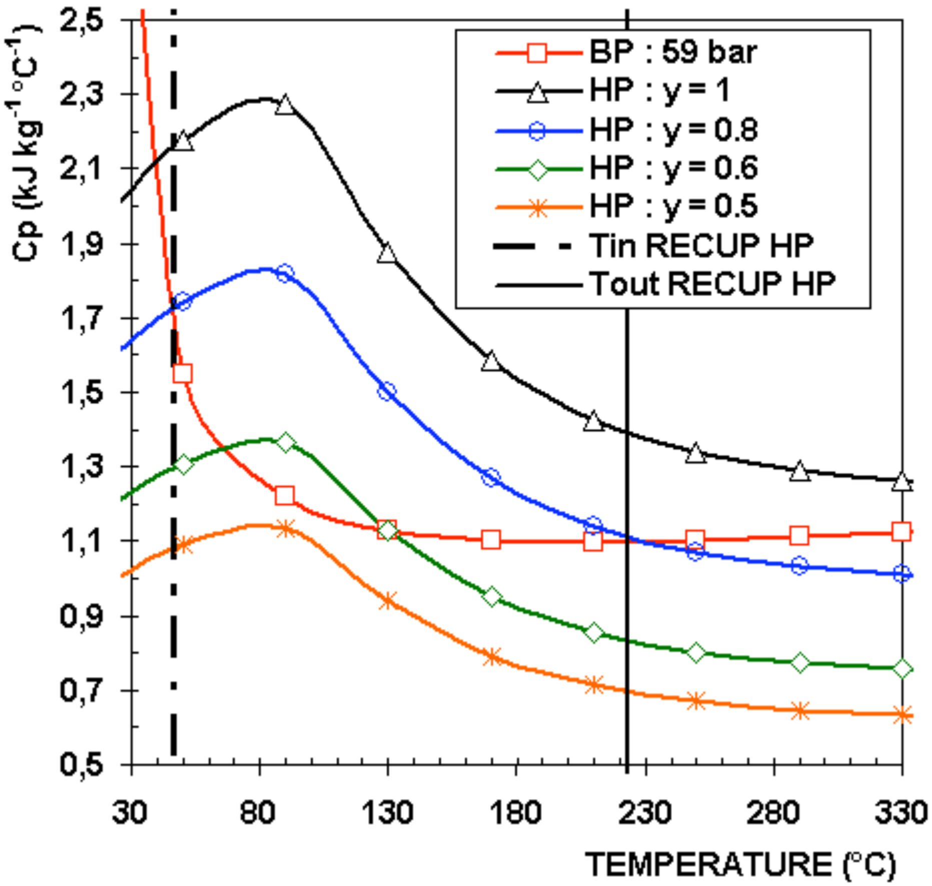


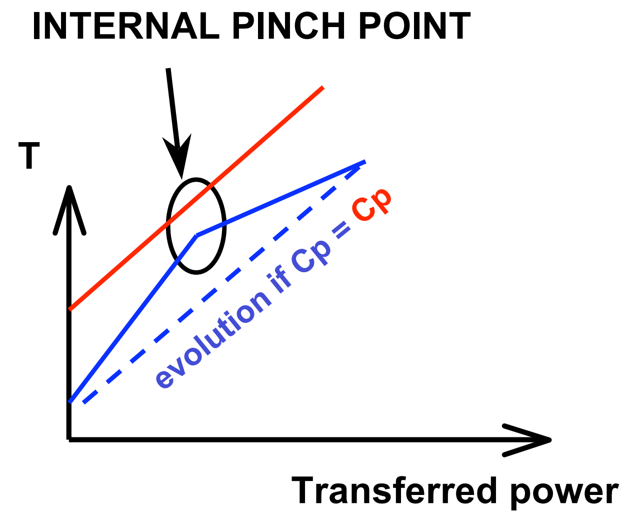


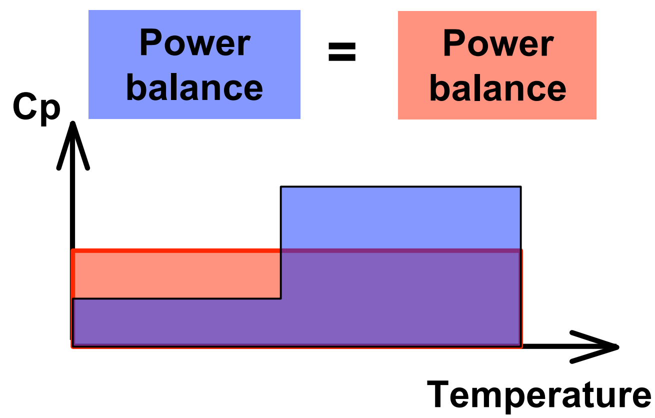
- Plates & fins technology
- Preliminary sizing : 300 MWth, η_{IHX} = 97%
  - gas: d_h ~ 2 mm, offset strip fins
  - Na: d_h = 4 mm, straight fins
- SCO_2 compactness :
  ☺ $\rho = 10 \times \rho_{N2\text{-}He} \Rightarrow \downarrow V \Rightarrow \downarrow \Delta P/L$
  $\Rightarrow \uparrow Re \Rightarrow \uparrow Nu$

  ☹ $\lambda = 1/3 \times \lambda_{N2\text{-}He} \Rightarrow \downarrow h$
- L = 3 m ⇒ d_h SCO_2 can be further reduced
- Sensitivity - basis : η_{IHX} 88% ; ΔP 1 bar ; η_{cycle} 45.1% ⇒ C 28 MW/m^3
  - η_{IHX} 95% ⇒ ☺ η_{cycle} 45.8% ; ☹ C 15 MW/m^3
  - ΔP 2 bar ⇒ η_{cycle} 45% ; ☺ C 40 MW/m^3

CYCLE COMPONENTS : *recuperators, pinch point*







- **Internal pinch-point issue ⇒ Energy balance at inlets / outlets is not sufficient**
- **High power e.g. ~ 2 × turbine power for He/SCO_2 (GT-MHR : about 1 ×).**
  → significant impact of recuperator efficiency : + 5 % $\eta_{recuperator}$ ⇒ + 2 % η_{cycle}
- **{ΔP = 0.4 b, η_{recup}= 90%} ↔ 15-20 MW/m^3 compactness for plates & fins techno.**

CYCLE COMPONENTS : *SCO_2 axial turbomachinery*

- **VERY PRELIMINARY SIZING, ΔH_{MAX} / STAGE ⇒ STAGE COUNT :**
  - **Compressor : Haller criterion for diffusion losses, qualitative, $V_2 / V_1 \leq 0.72$**
  - **Turbine : Craig & Cox abacus, $\Delta H/U^2$ = function (Va/U, η)**

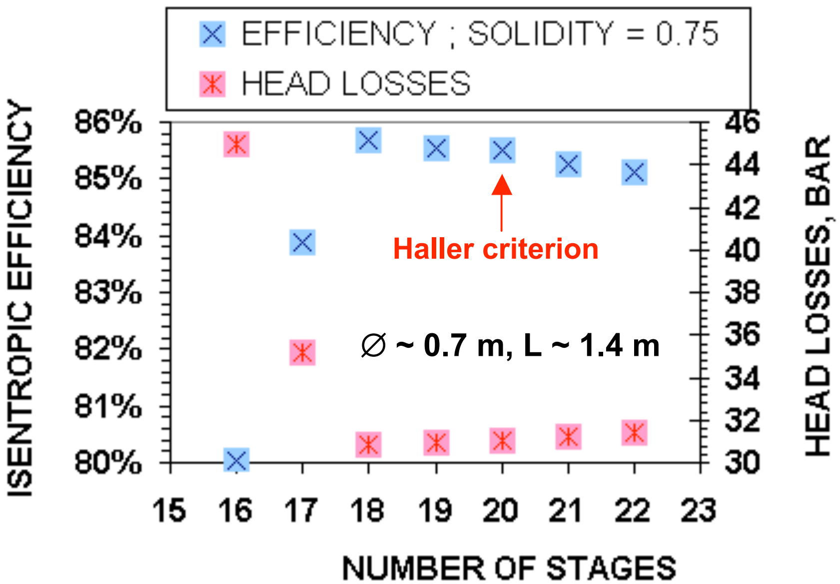
- **600 MW_{th} cycle, $T_{in\text{-}Turb}$ 550°C, 200/76.9 bars, $T_{out\text{-}cold\ sink}$ = 32°C.**

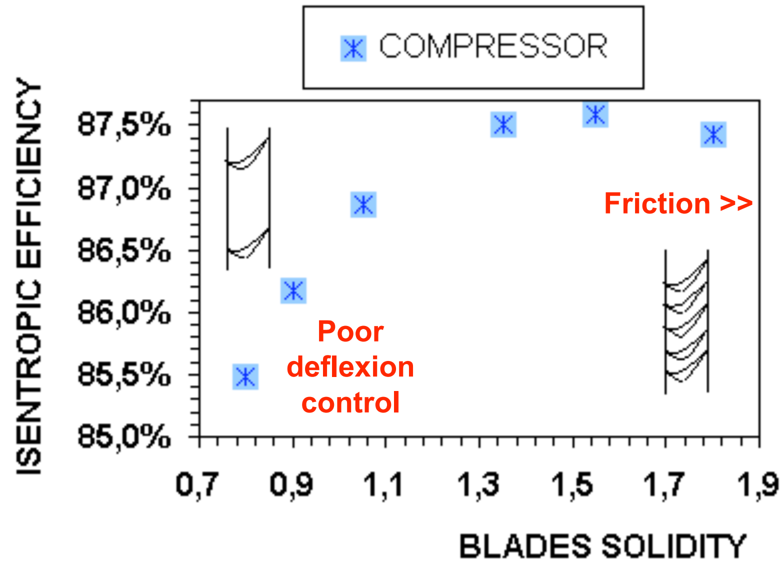
		P_h/P_b	Stages count	$\varnothing_{max}$	Minimum blades height
COMPRESSOR	GT-MHR (He)	2.85	BP+HP : 40	1.7 m	5 cm
	CO_2 SC- BYPASS	2.6	~ 10	< 0.9 m	< 3 cm
TURBINE	GT-MHR (He)	2.7	12	1.9 m	-
	CO_2 SC	2.5	~ 5	1.2 m	-

→ **IMPRESSIVE COMPACTNESS FOR AXIAL TECHNOLOGY**

CYCLE COMPONENTS : *Na/SCO_2 bypass compressor sizing*

- **AXIAL SIZING: mean line analysis ; diffusion, secondary & annular losses**





- **Blades count (solidity) : balance of blades flexion stress ↔ $\eta_{compressor}$ optim.**
- **η = 88% applying correlations from opened literature ; results reliability ?**
  - ☺ **diffusion losses: should be ok, high reynolds number (10^7).**
  - ☹ **annular & secondary losses for such small blades height ?**
  - ☹ **<u>tip clearance losses</u> (not modelised).**
- **RADIAL - ΔH = 99 kJ/kg ⇒ 1 stage ⇒ U ~ 330 m/s ; ∅ ~ 2 / 1 m at 3000 / 6000 rpm**
  ⇒ ∅ = 1 m at 3000 rpm ⇒ ≥ 5 stages
  ↪ η ?

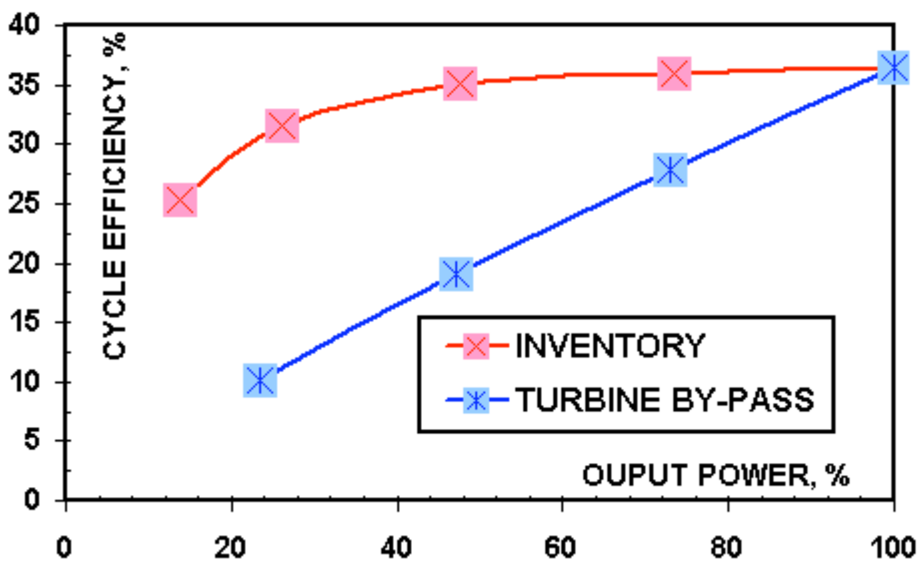
CYCLE OPERATION : *part load control, nitrogen case*

- Na/N_2 PART LOAD CONTROL : Na/N_2 CORE & TURBINE BYPASS OR INVENTORY
- INVENTORY CONTROL PRINCIPLE : MAINTAIN OF VOLUMETRIC FLOWS AT TM INLETS
  - Turbine & Comp. velocity Δ remain constant i.e. at design incidence.
  - ΔH/kg remain constant (Euler) ⇒ so does pressure ratio (perfect gas law)

η_{cycle} AT DESIGN

STATIONARY :
– Mass flow balance
(dρ & dP sym. variation, perfect gas law)

– Pressure equilibrium



IN FACT, INVENTORY CONTROL → η_{cycle} ↓ :

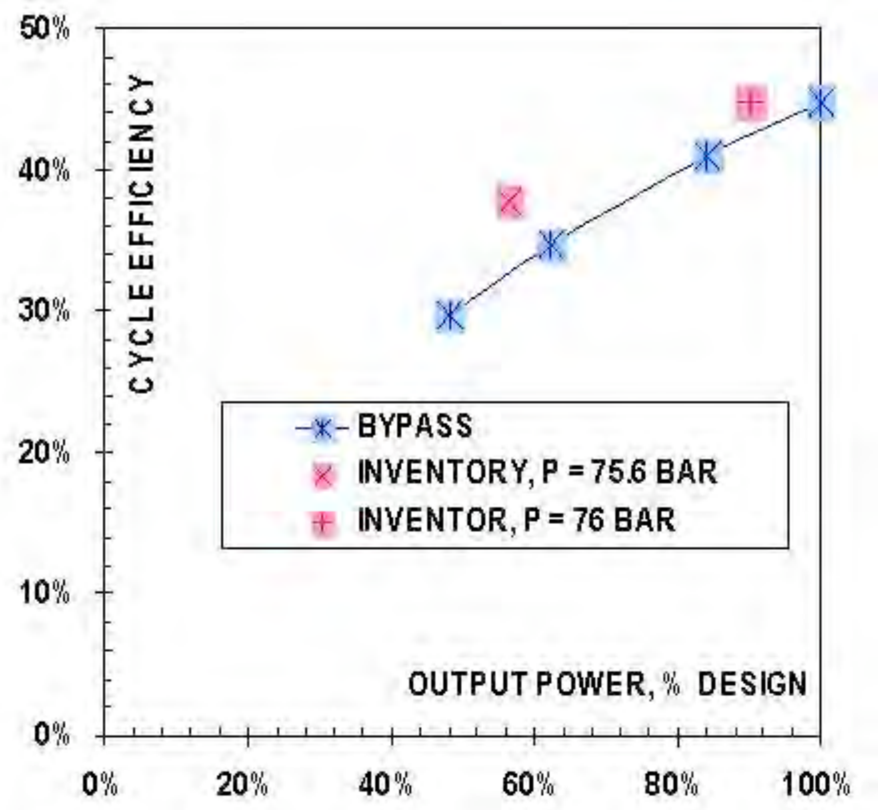
- Turbine power ↓ proportionaly to pressure ↓ whereas head losses pumping power has a slower ↓ with pressure.
- Turbomachinery efficiency ↓ because :
  - Reynolds number ↓
  - operating point is slightly modified (due to head losses) → blades incidence

CYCLE OPERATION : *part load control, SCO_2 case*

- **INVENTORY CONTROL STUDY : GFR ; η_{design} = 44.8% ; 76.9 bar / 32°C**

	Pressure decrease	% nominal pressure ratio	% nominal density
Main compressor	-4%	74%	-47%
Bypass compressor	-4%	~ 100 %	-5.5%
Turbine	-50%	~ 100%	-48.5%

- **ASYMMETRY :**
  - **Turbine & bypass compressor : close to perfect gas behaviour $\Rightarrow W = \Delta H = f(P_{out}/P_{in}, Te)$,**
  - **Main compressor, real gas behaviour $\Rightarrow dH = f(P,T)$**
  - **Simulation = maintain of volumetric flow at both compressor inlets (ΔH/kg)**
    - **Change of turbine volumetric flow**
    - **Additional valves at turbine & bypass compressor outlets for pressure equilibrium ($\Rightarrow$ losses)**
    - **Flow split modification to adjust flow rate (asym. density change)**



- **VERY COMPLEX : valves + weak pressure variation**

CYCLE OPERATION : *cold sink issue*

ISSUE : IMPACT OF SEASONAL VARIATION OF COLD SINK TEMPERATURE ON CYCLE OPERATION AND EFFICIENCY.

NITROGEN			H2O			CO2		
T (°C)	P (bar)	Density (kg/m3)	T (°C)	Psat (bar)	Density (kg/m3)	T (°C)	Psat (bar)	Density (kg/m3)
21	23,3	26,8	21	0,025	997,9	21	58,8	762,9
31	23,3	25,9	31	0,045	995,3	25	64,5	711,5
61	23,3	23,4	61	0,209	982,7	31	74	563,5

→ **SCO_2 cycle designed for condensation (η: + 4 pts) implies significant change of low pressure to meet saturation. Process ? Cycle efficiency ?**

CO2_SC

T (°C)	P (bar)	Density (kg/m3)	ΔH COMP. (kJ/kg)	P_{out} (bar)	
32	76,9	598,1	18,4	200	
27	76,9	739,4	15,9	200	
37	76,9	273,7	32,4	200	→ Vol. flow rate = × 2.2 ; W = × 1.8

→ **Even when designed for 76.9 b & 32°C, SCO_2 cycle implies significant change of density as well as power required to maintain pressure ratio.**

→ **What is the new operating point and associated cycle efficiency in case of single shaft for compressors and turbine ?**

→ **May possibility of compressor speed change simplify and optimise cycle process and efficiency ?**

→ **Need of turbomachinery performance maps (off-design).**

CONCLUSION

- **Na / SCO_2 CHEMICAL INTERACTION : key point for sodium fast reactors !**

- **CYCLE THERMODYNAMICS : attractive efficiency at design, but :**
  - **reduced gain / nitrogen depending on part load conditions occurrence (i.e. other part load following mode to be found).**
  - **adaptation to cold sink temperature change to be studied : relevance of condensation cycle & speed change requirement?**

- **CYCLE COMPONENTS :**
  - **compactness.**
  - **main concern is for compressor efficiency due to its very small blades.**
  - **relevance of a radial compressor instead of an axial: efficiency of a such a component ?**

- **CYCLE OPERATION : stability concern due to significant variations of physical properties close to critical point → need of a dynamic code to study this point (with a good description of components running !)**

CYCLE THERMODYNAMICS : *He/SCO₂*

$POWER_{core}$ = 2400 Wth

$P_{in\ turbine}$ = 250 bar (25MPa)

$T_{in\ turbine}$ = 650°C (923 K)

P = 76.9 bar (7.69MPa)

$T_{out,\ cold\ sink}$ = 32°C (305 K)

→ **Cycle efficiency = 44.8%**

Psat = 59 bar (5.9MPa)

$T_{out,\ cold\ sink}$ = 21°C (294 K)

→ **Cycle efficiency = 48.7%**

ref - GTMHR : η ~ 47%

T_{in} turb 850°C

He direct cycle