Design, modeling and experimentation of a reversible HP/ORC prototype

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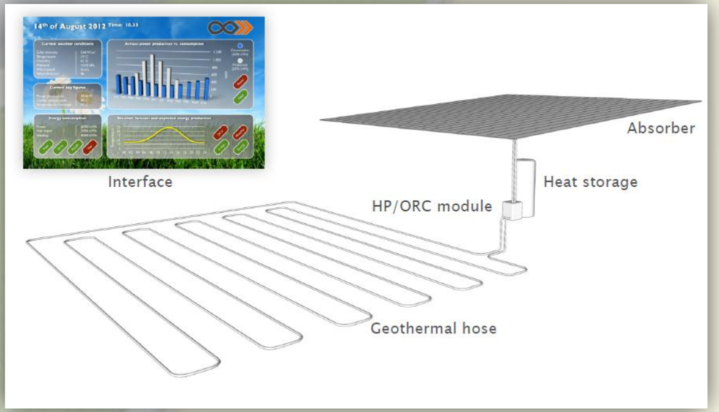
2) Prototype
3) Selling units

Introduction

Description of the HP/ORC unit

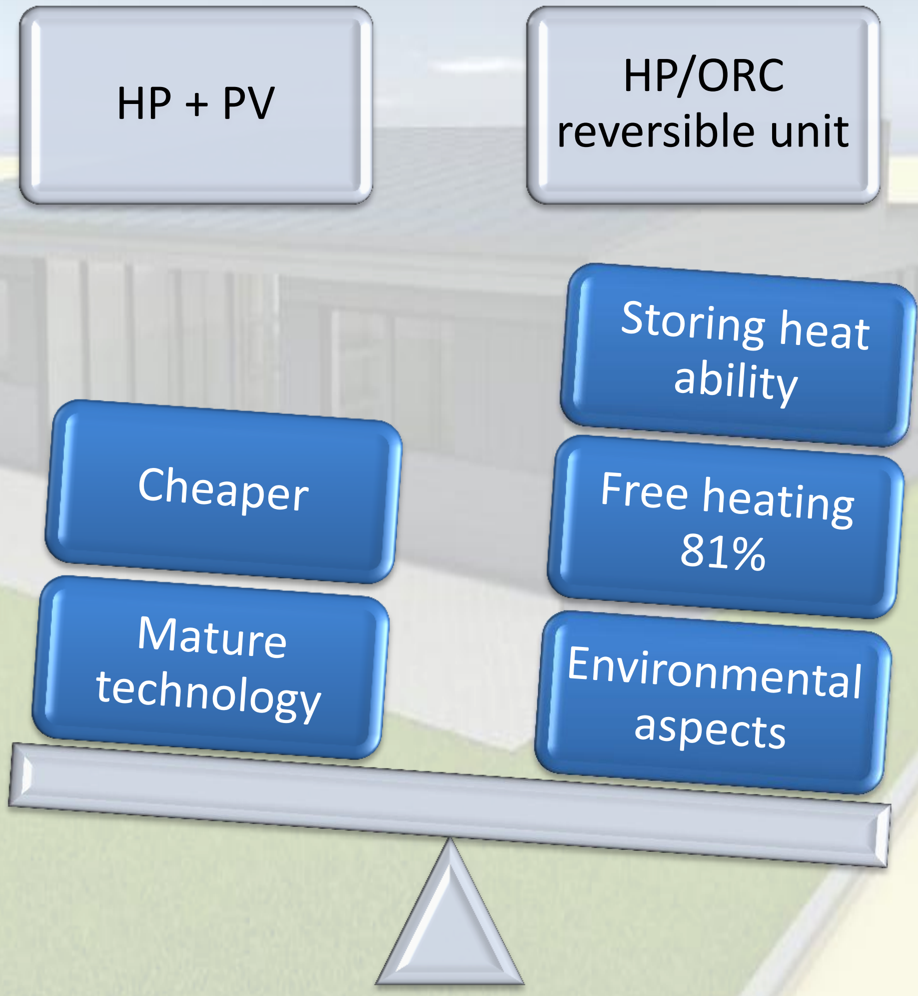
Reversible HP/ORC unit = Heat pump with the ability to work as an ORC

- Almost the same components as residential heat pump (+4 way valve and pump)
- Possibility to produce "green" electricity



Introduction

Comparison with HP + PV



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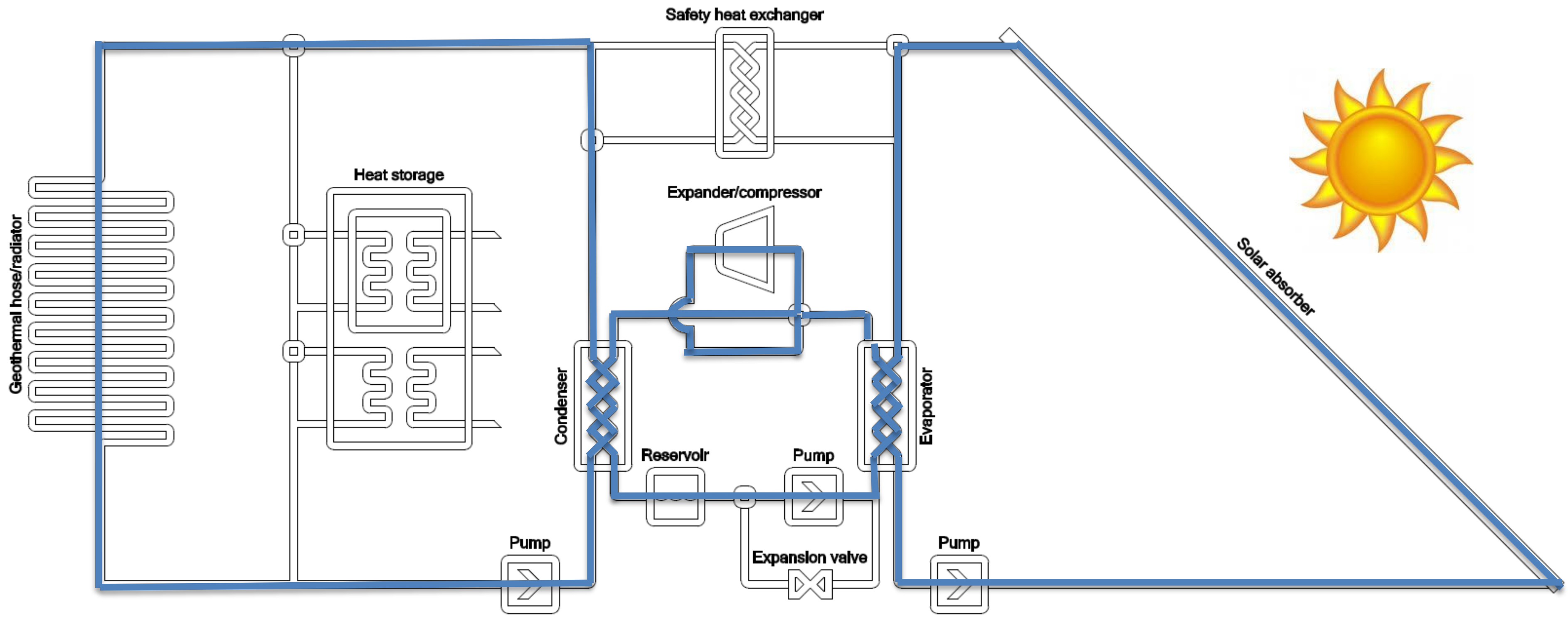
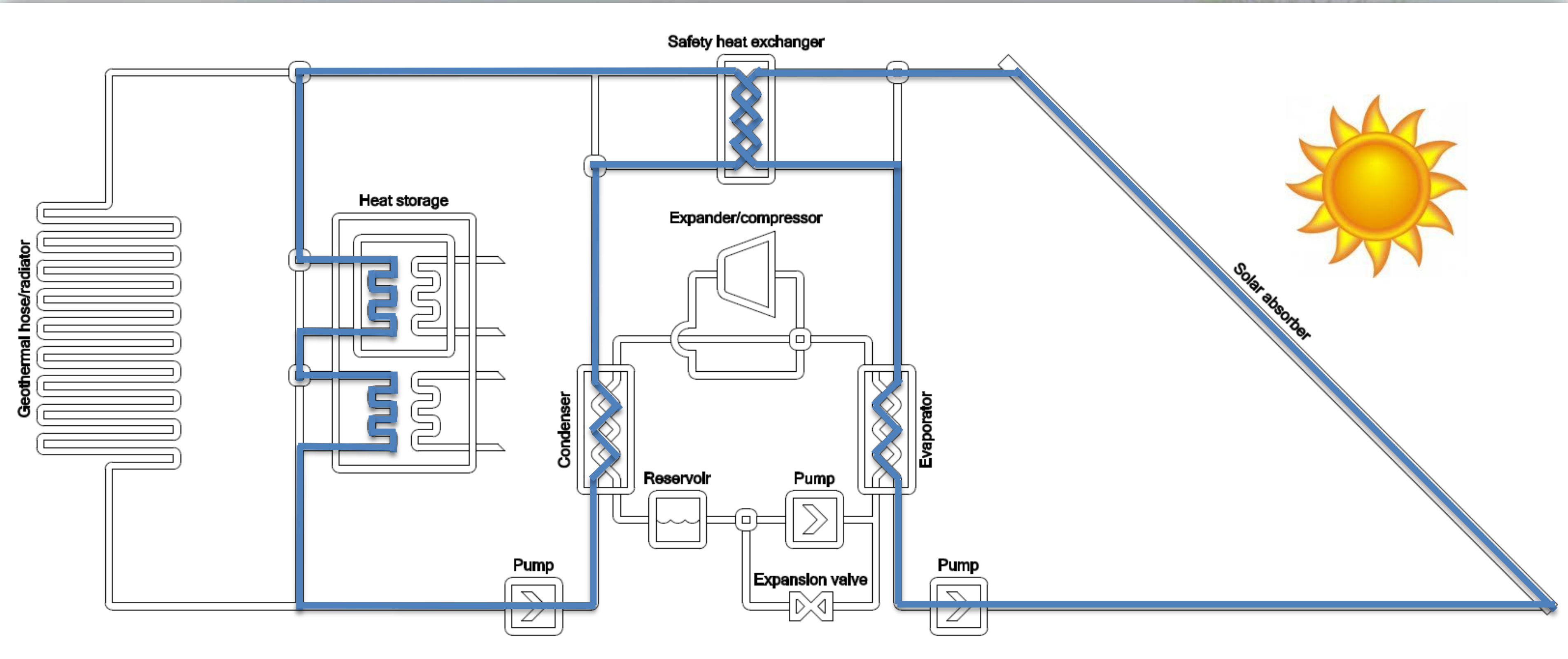
Design

ORC mode



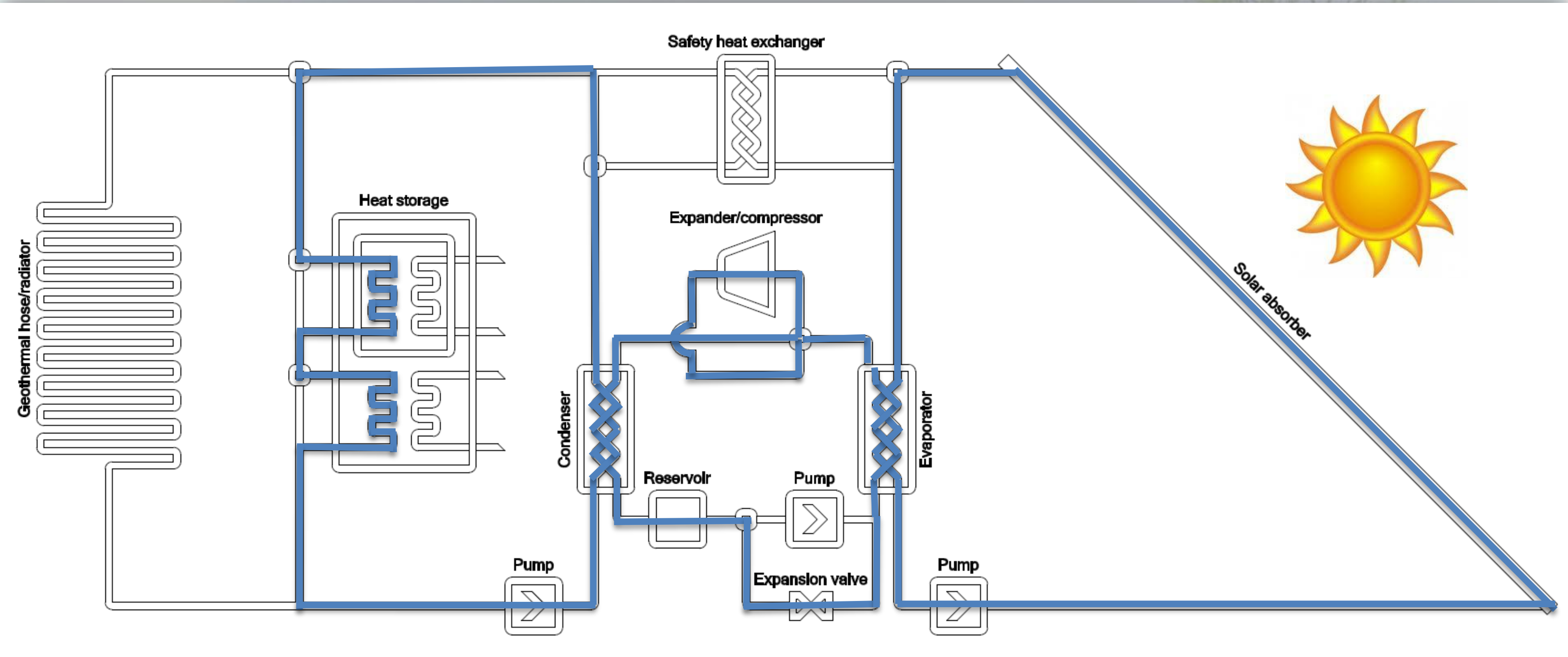
Design

Direct heating mode



Design

HP mode



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System sizing

Sizing – Introduction

Sizing difficulties because of the large difference between ORC and HP :

Different temperature levels → different flows

→ different pressure levels

Sizing based on the ORC mode because of :

- The higher thermal power (62 kW versus 7 kW)

- More functionning time

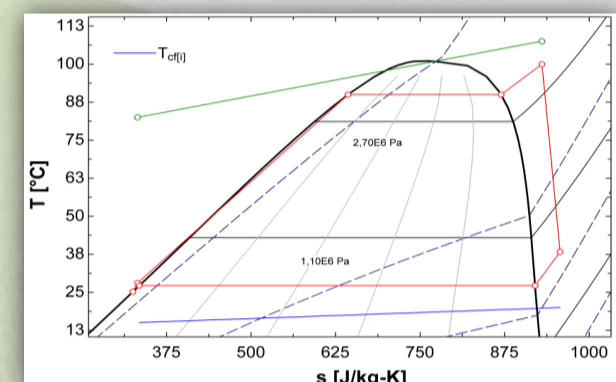
System sizing

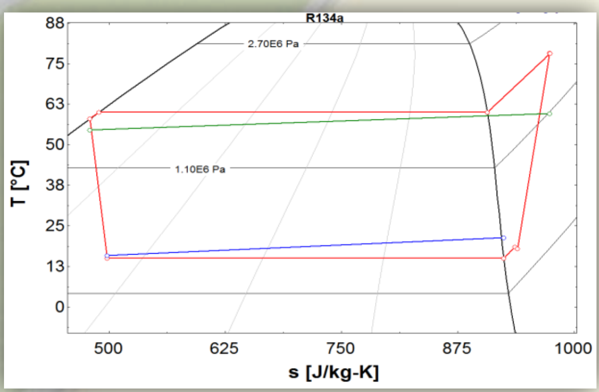
Sizing – Nominal conditions

	ORC nominal conditions		HP nominal conditions	
Evaporator	T_{ev} [°C]	90*	Q_{ev} [kW]	8**
	$\Delta T_{w,h}$ [°C]	25	$\dot{m}_{w,h}$	$= \dot{m}_{w,h}$ (ORC)
	Pinch point [°C]	5	Pinch point [°C]	5
	Overheating [°C]	10	Overheating [°C]	3
Condenser	$T_{wc,su}$ [°C]	15	T_{cd} [°C]	60 (for DHW production)
	$T_{w,c,ex}$ [°C]	20	Pinch point [°C]	7,5
	Pinch point [°C]	7,5	Sub-cooling [°C]	2
	Sub-cooling [°C]	2	ΔT_{wh} [°C]	5

* ≈ $P_{comp,max}$ = 32bars

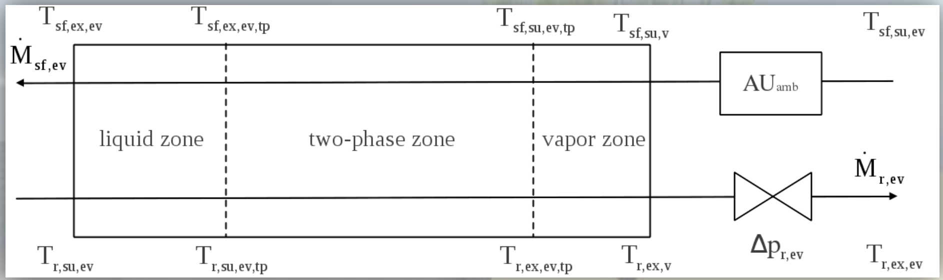
** (I=90 W/m2) Representative of Denmark winter conditions





System sizing

Modeling – heat exchangers

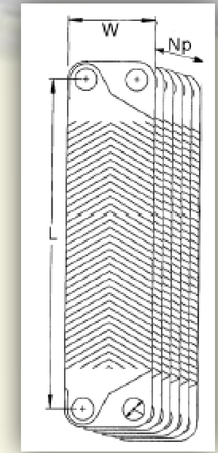


- Evaporator and condenser = plate heat exchangers described by a **3-zone model**.
- **Heat transfer**
  - ✓ 2 convective resistances in series
  - ✓ Appropriate heat transfer and pressure drop correlations have been used [1-3].
  - ✓ Average value over the 2-phase zone
  - ✓ Total heat transfer area = sum zones areas
- **Pressure losses**
  - ✓ Frictional losses
  - ✓ Two-phase zone: integration versus x.
  - ✓ Only on the vapor side

$$\frac{1}{U} = \frac{1}{h_f} + \frac{1}{h_{sf}}$$

$$\bar{h}_{tp} = \int_0^1 h_{tp} dx$$

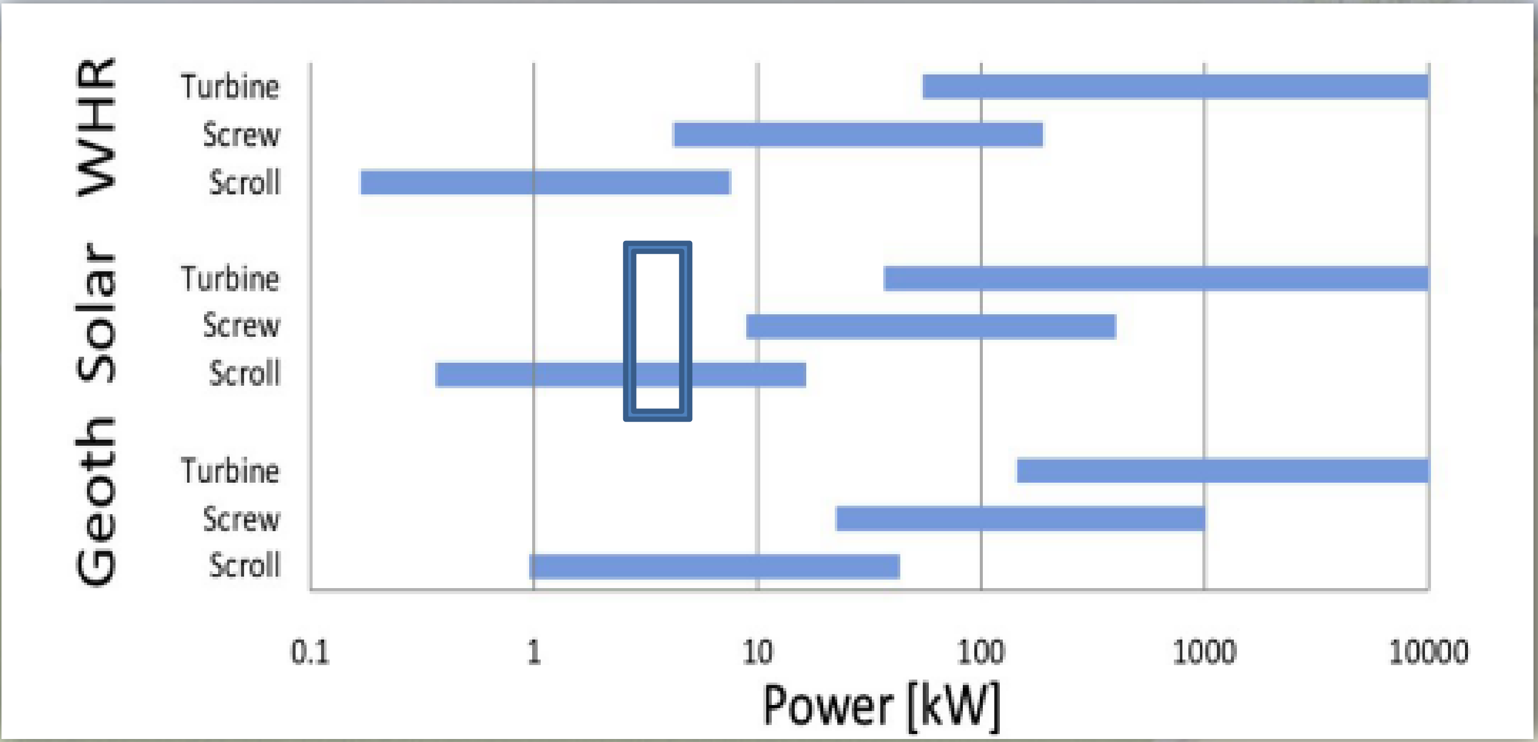
$$A_l + A_{tp} + A_v = (N_p - 2) \cdot L \cdot W$$



$$\Delta p_{tp} = \int_0^1 \frac{2 \cdot f_{tp} \bar{v} \cdot G^2}{D_h} dx \cdot L$$

System sizing

Selection – Compressor / expander



Choosing the adapted expander technology [4]

System sizing

Modelling – Compressor / expander

- Compressor = manufacturer correlations [5]

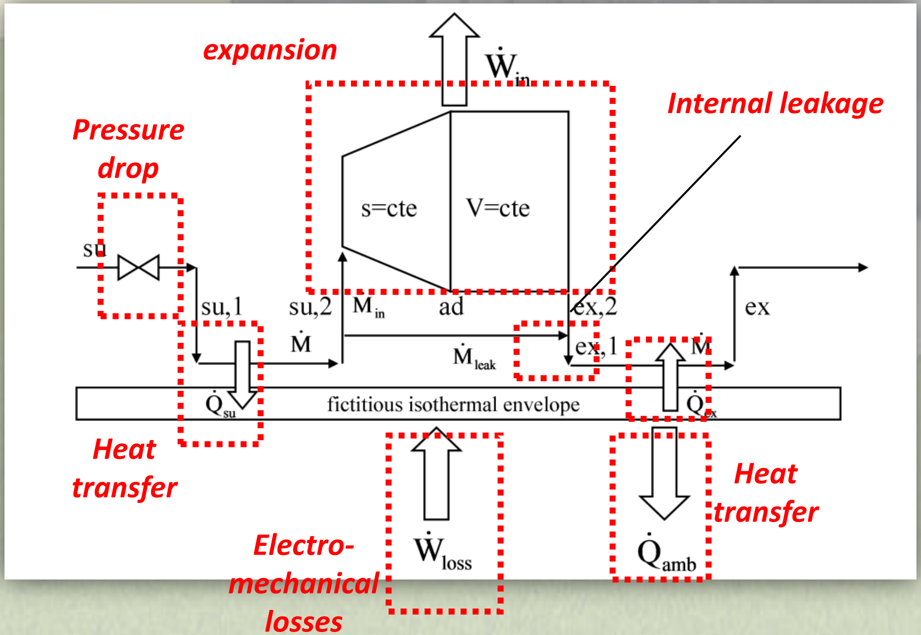
$$\dot{W}_{cp} = C_0 + C_1.T_{ev} + C_2.T_{cd} + C_3.T_{ev}^2 + C_4.T_{ev}.T_{cd} + C_5.T_{cd}^2 + C_6.T_{ev}^3 + C_7.T_{cd}.T_{ev}^2 + C_8.T_{ev}.T_{cd}^2 + C_9.T_{cd}^3$$

$$\dot{M}_{cp} = C_{m0} + C_{m1}.T_{ev} + C_{m2}.T_{cd} + C_{m3}.T_{ev}^2 + C_{m4}.T_{ev}.T_{cd} + C_{m5}.T_{cd}^2 + C_{m6}.T_{ev}^3 + C_{m7}.T_{cd}.T_{ev}^2 + C_{m8}.T_{ev}.T_{cd}^2 + C_{m9}.T_{cd}^3$$

Ambient losses neglected:

$$h_{ex,cp} = h_{su,cp} + \frac{\dot{W}_{cp}}{\dot{M}_{cp}}$$

- Expander = semi-empirical model [6]



Parametres	Values
$\dot{V}_s$ [m³]	98,04. 10^{-6}
R_v [-]	2,9
A_{leak} [m²]	4,5.10^{-7}
$Au_{su,n}$ [W/K]	30
$Au_{ex,n}$ [W/K]	20
Au_{amb} [W/K]	10
α [-]	0,23
$\dot{W}_{loss\_0}$ [W]	120
D_{ex} [m]	0,0056

Calibrated on experimental data

Connected to the grid (fz = cst = 50 Hz)

System sizing

Sizing – Compressor / expander

Compressor		A	B	C
Swept volume [cm^3]		80 	100 	120 
Heat pump	Power consummed [W]	**2,687**	3,211	4,276
	ε_s [%]	59,8	60	50
	η_{abs} [%]	52,9	56	57
	COP [-]	2,4	2,4	2,1
ORC	Power generated [W]	4,013	4,733	**5,718**
	ε_s [%]	67,8	68	68,2
	η_{abs} [%]	52,58	55,31	58,3
	η_{ORC} [%]	7,5	7,6	7,6

The size of the scroll machine (which defines the net power of the system, both in HP and ORC mode) results from a tradeoff between winter and summer conditions. This can only be optimized using yearly simulations,

System sizing

Modeling – Other components

Pump (variable speed)

High pressure drop and low volume flow ➔ volumetric pump (Plunger)

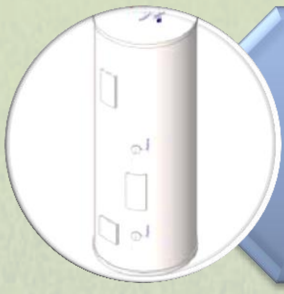
Criteria : Tighness and relatively high efficiency

$\varepsilon_s = 0{,}5$

Absorber (simple linear model) [7] :

$$\dot{Q}_{abs} = S_{abs}\,(\text{-}26{,}2\text{-}1{,}22\ T_{amb} - 1{,}783\ \Delta T_{abs} + 0{,}9034\ I)$$

$$\eta_{abs} = \frac{\dot{Q}_{abs}\eta_{glazing}}{I}$$



Storage [8] :

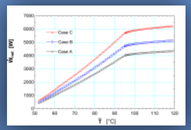
$$Q_{stock} = 4{,}2.\,V_{Stock}^{0{,}47}.\,(T_{stock} - T_{amb})$$

Yearly simulation of the system

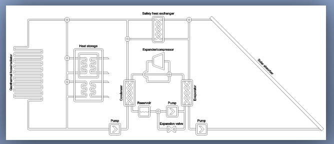
Off-design performance

For a given configuration (fluid, expander size, recuperator or not), evaluation of the system performance over a wide range of evaporation/condensation temperatures (ORC and HP).

Establishment of performance curves from these simulations as a function of the system configuration and of the temperature levels.



Implementation of these curves in the yearly simulation model, which optimally switches between the three operating modes depending on the weather and of the heat demand for the given month.



System sizing

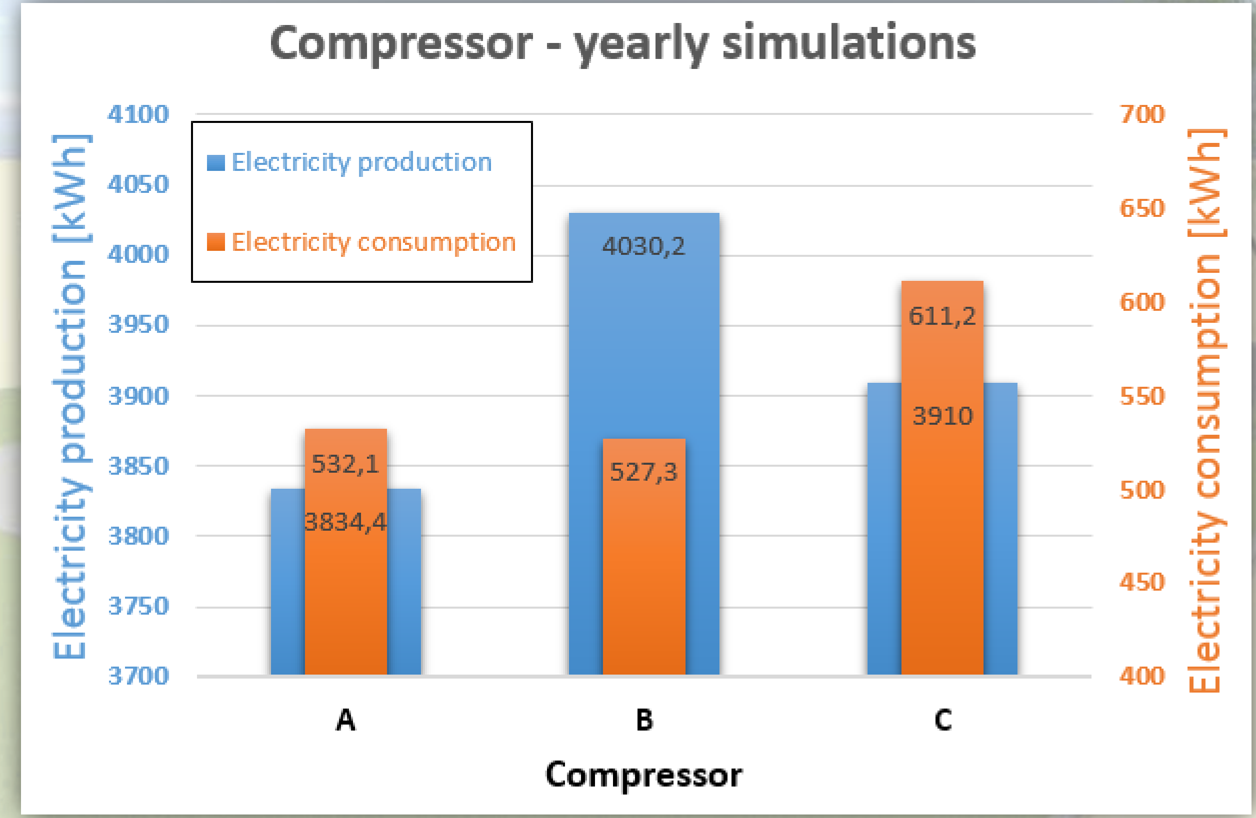
Sizing – Fluid selection

- Low cost
- Wide availability
- Low specific volume
- High thermal conductivity
- Acceptable pressure levels
- High adiabatic enthalpy drop
- High specific heat
- High thermal stability
- Low viscosity
- Non-corrosive
- Non-toxic
- Easily recyclable
- Material and lubricating oil compatibles
- With a melting point lower than the lowest ambient temperature throughout the year

Fluid	W_net [kWh]	Improvement	ODP	GWP	Toxicity	Flammability	Conclusion
R124	5079.42	45.28%	-	-	+	+	Environmental reasons
R600	4239.98	21.27%	+	+	+	-	Flammability
R152a	3969.01	13.52%	+	+	+	+/-	Flammability
R600a	3814.85	9.11%	+	+	+	-	Flammability
R134a	3496.27	0.00%	+	-	+	+	Best compromise
R245fa	3349.76	-4.19%	+	-	+/-	+	Toxicity + low W_net
R123	3105.28	-11.18%	-	-	-	+	Environmental reasons + low W_net

Yearly simulation of the system

Compressor selection



Electrical energy produced over one year reaches 4030 kWh and the monthly efficiency of the cycle varies between 4.3 and 6.4%

The monthly COP of the heat pump varies from 2.6 to 3.3, for a yearly electrical energy consumption of 527.3 kWh.

The direct heating mode provides 62.3 kWh of heat throughout the year.

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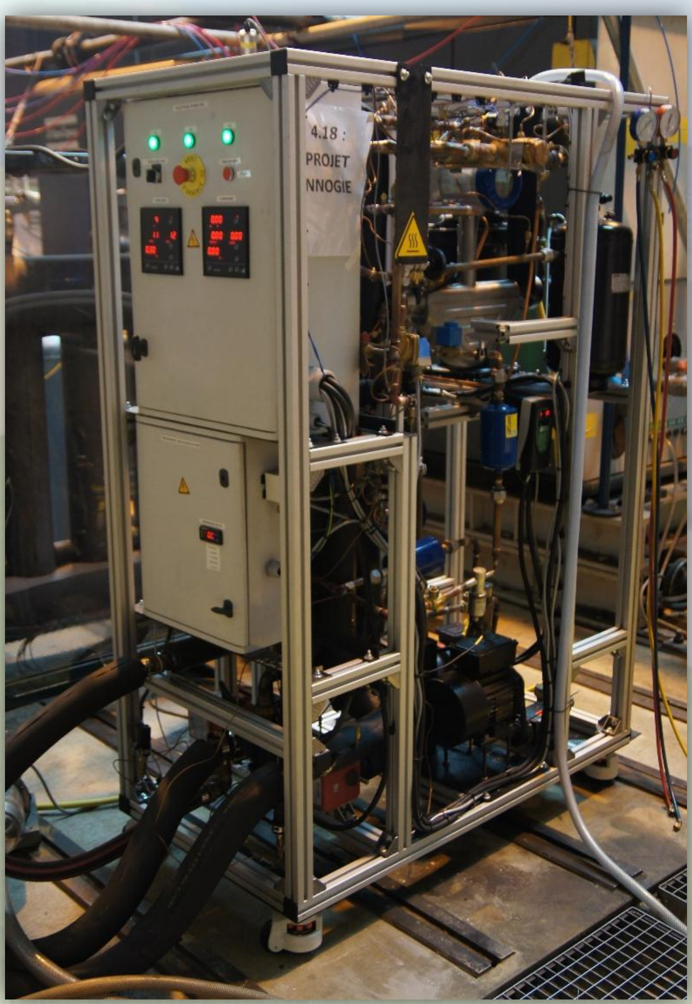
Experimental results

Best efficiency points



$$\varepsilon_{is,comp} = \frac{\dot{m}_r(h_{\text{comp,ex,s}} - h_{comp,su})}{W_{comp,el}}$$

$$\varepsilon_{is,exp} = \frac{W_{exp,el}}{\dot{m}_r(h_{\text{comp,ex,s}} - h_{comp,su})}$$

$$\varepsilon_{vol} = \frac{\dot{m}_r}{\rho\, N\, Vs}$$

$$\eta_{ORC} = \frac{W_{exp,el} - W_{pump,el}}{\dot{Q}_{ev}}$$

	HP	ORC
$P_{comp,el}$ [kW]	3.4	3.1
Q_{ev} [kW]	12	44
Q_{cd} [kW]	14.4	38
$P_{pump,el}$ [W]	-	600
$\varepsilon_{comp/exp,is}$ [-]	0,56	0.64
T_{ev} [°C]	16	78
T_{cd} [°C]	52	22
P_{cd} [bar]	13.7	6
P_{ev} [bar]	5.1	25.7
COP / η_{ORC} [-]	4.2	5.7 %

Performances lower than theory because : 1) low expander efficiency, 2) non thermally insulated pipes, 3) limited power of the boiler, 4) Huge pressure drop on the four way valve 5) necessary subcooling to avoid pump cavitation

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- **Conclusions and perspectives**

Perspectives

Energy production increase over a year

Geometry → ≈ 10%

Increase of the maximum pressure → ≈ 3%

Variable speed → ≈ 4%

Improved pump (ε_s = 0,5 → ε_s = 0,75) → ≈ 9%

Improved fluid

Recuperator → ≈ 4%

New 4 way valve presenting a lower pressure drop

Conclusions and next steps

- Sizing and simulation of a reversible HP/ORC system
  - Yearly produced electrical energy = 4030 kWh
  - Monthly efficiency of the ORC = [4.3% - 6.4%]
  - Monthly COP of the heat pump = [2.6 - 3.3]

- First experimental results
  - ORC ➔ η_{ORC} = 5.7 %
  - HP ➔ COP = 4.2

- Potential means of improvements

- Next steps:
  - ➢ Monitor the prototype installed in a real building.
  - ➢ More detailed simulation / validation of the model of components with experimental data.

Thank you!

Further information:

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More experimental results and model validation at turboexpo conference next year

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