

Development of a HTHP with 4 compression stages using R718 as refrigerant

For utilization of geothermal energy

Ole Marius Moen, Research scientist, SINTEF Energy Research

Ole.moen@sintef.no

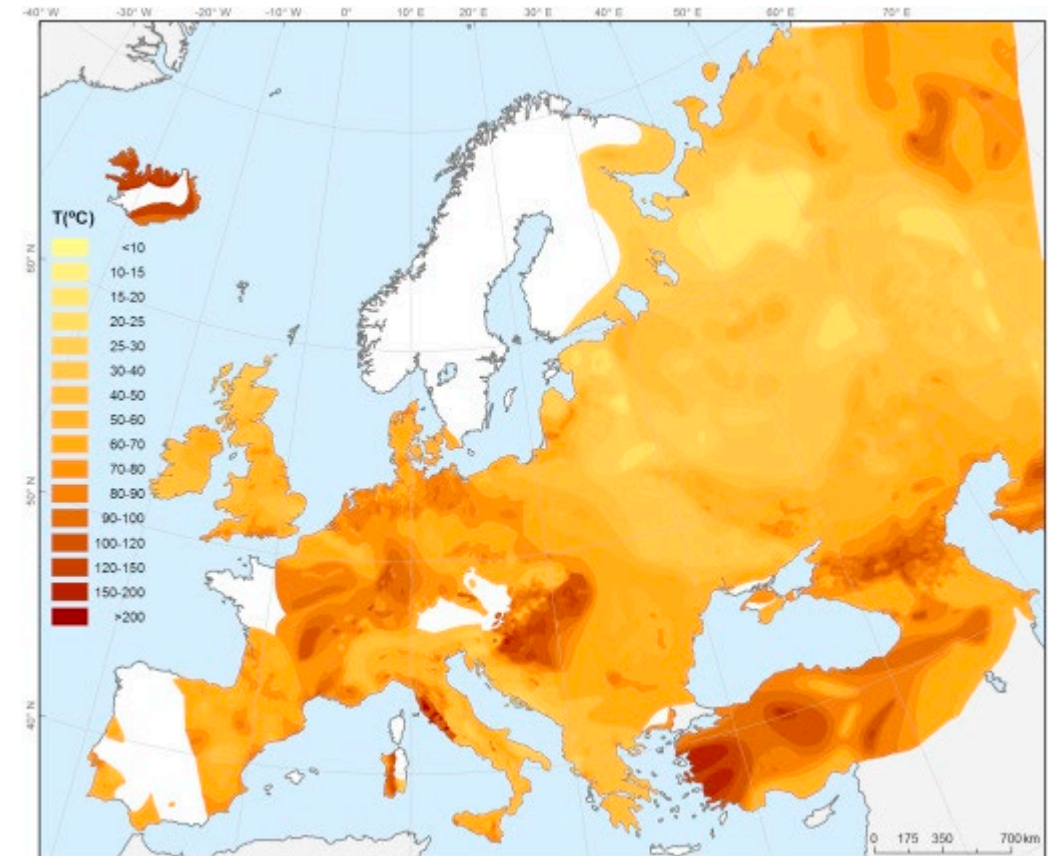


Presentation overview

1. Geosyn Project background and demonstration site
2. System and compressor design requirements
3. Optimization of compressor parameters
4. HTHP layout and performance evaluation
5. Final design and dynamic operation
6. Conclusion and outlook

Geothermal resources in Europe

- While geothermal heat has been exploited for thousands of years, it is still considered an emerging technology¹
 - low visibility
 - limited industrial exploitation in some countries
- Historically, the focus has been high-temperature resources for production of electricity
- Less attention to medium-temperature resources – available in almost 20% European area – on the potential for producing high-temperature heat and cold



Extracted temperatures at 2000 meter depth from the Atlas of Geothermal Resources in Europe (Chamorro et al., 2014)

Geosyn project



GEOSYN aims to develop a geothermal steam heat pump combined with heat-driven cooling systems (steam ejectors) for industrial applications

- HTHP demonstrator achieving temperature lift of 80 Kelvin (120-200°C)
- Steam (R-718) chosen as working fluid
 - Natural, GWP and ODP = 0, non-flammable, non-toxic, cheap and available anywhere, high critical temperature
- Multistage compression with a reciprocating compressor
 - Less explored for oil-free operation with R-718 than turbo^{1,2}, twin-screw³ and MVR fans⁴



www.geosynproject.eu

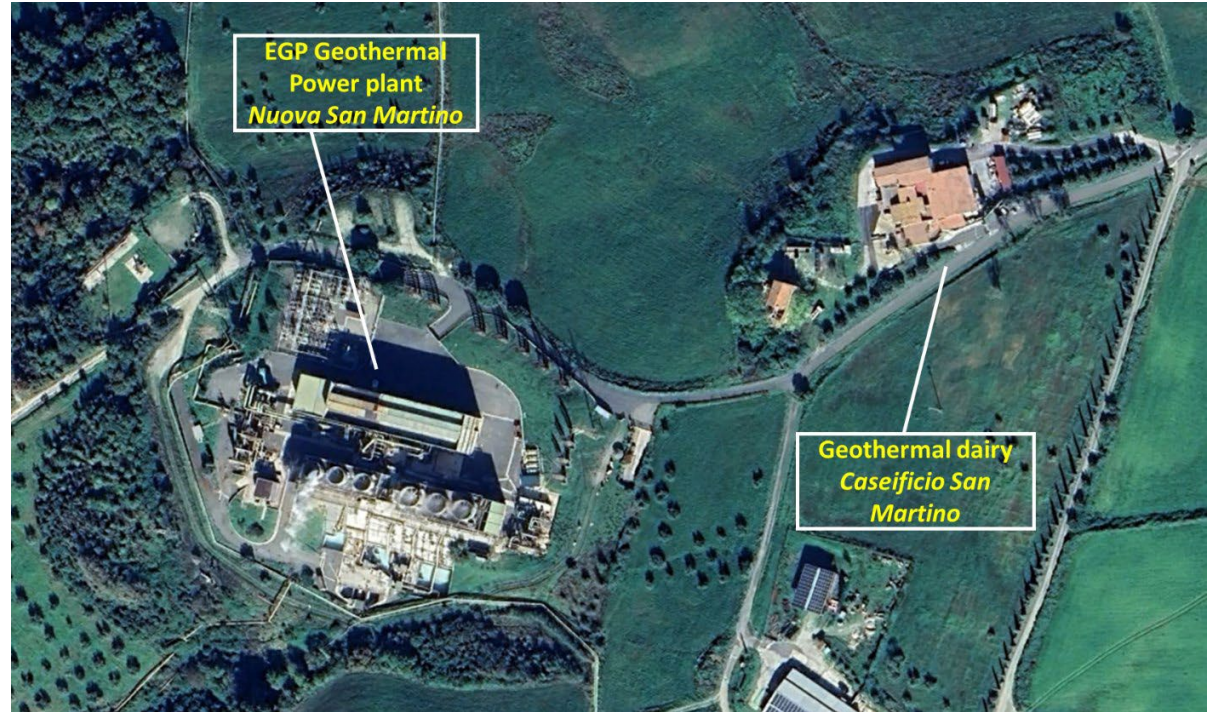
info@geosynproject.eu

#GEOSYNPROJECT



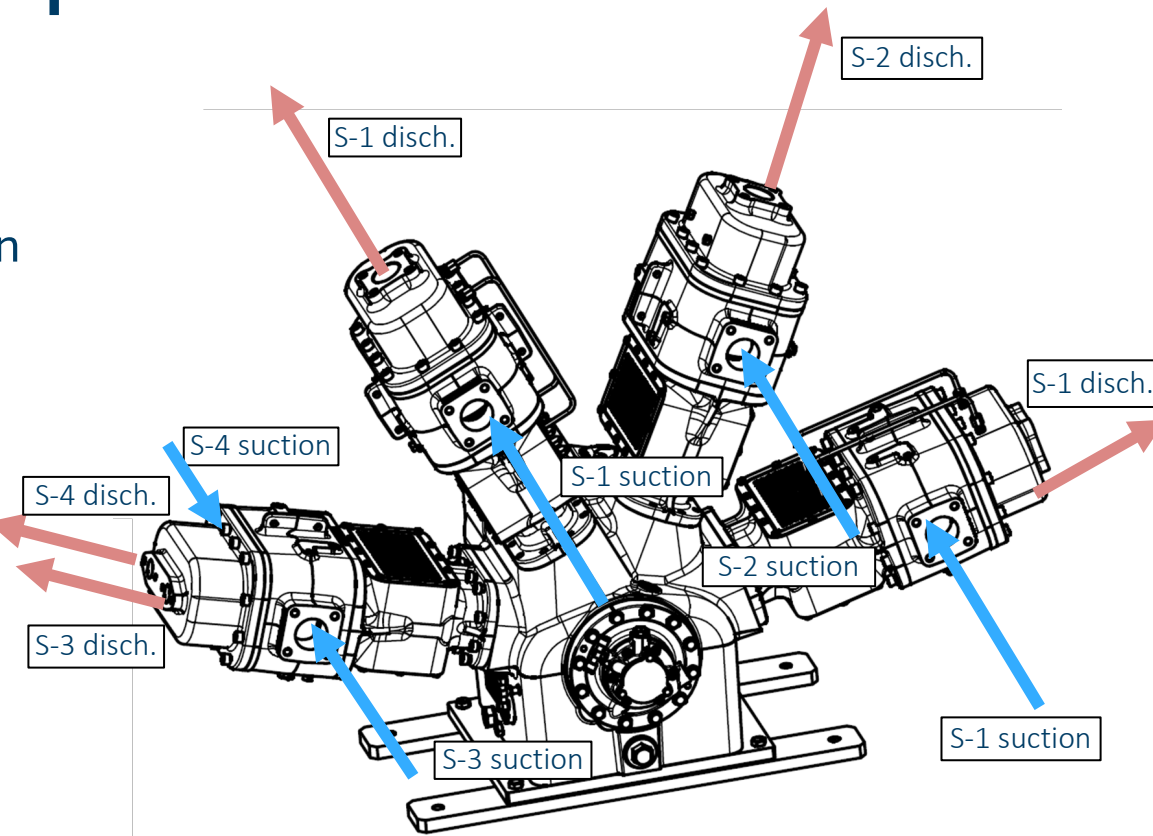
Demonstration site

- Demonstration site: San Martino Dairy factory in Italy
 - Steam available at 3-4 bara, 130-140°C from a nearby geothermal plant
- HTHP operating conditions at design point:
 - Evaporator: 2 bara, 120°C
 - Condenser: 15.5 bara, 200°C
- Maximum pilot thermal Capacity: 50 kW_{th}



Implementation and development of a reciprocating compressor

- Novel prototype manufactured by Officine Mario Dorin
- Open-type with 8 cylinders
- Designed for oil-free operation
 - Seals on the linear connecting rod preventing leakage of steam from cylinders into oil filled crank case
 - Self lubricating material used for piston rings and inner cylinder surface coating to reduce wear and extend lifetime
- Material limitations
 - Coating sensitive to water droplets
 - Sealing sensitive to temperatures



Compressor parameters	Value
Number of cylinders, $N_{c,stage}$ (1 – 2 – 3 – 4)	4 – 2 – 1 – 1
Speed range, RPM_{crank}	500–1450 [rev/min]
Speed at design point	1200 [rev/min]
Estimated isentropic efficiency, η_{is} at design point	0.75
Cylinder stroke length, L_s	65 mm
Maximum cylinder bore diameter, $D_{b,max}$	90 mm

System requirements

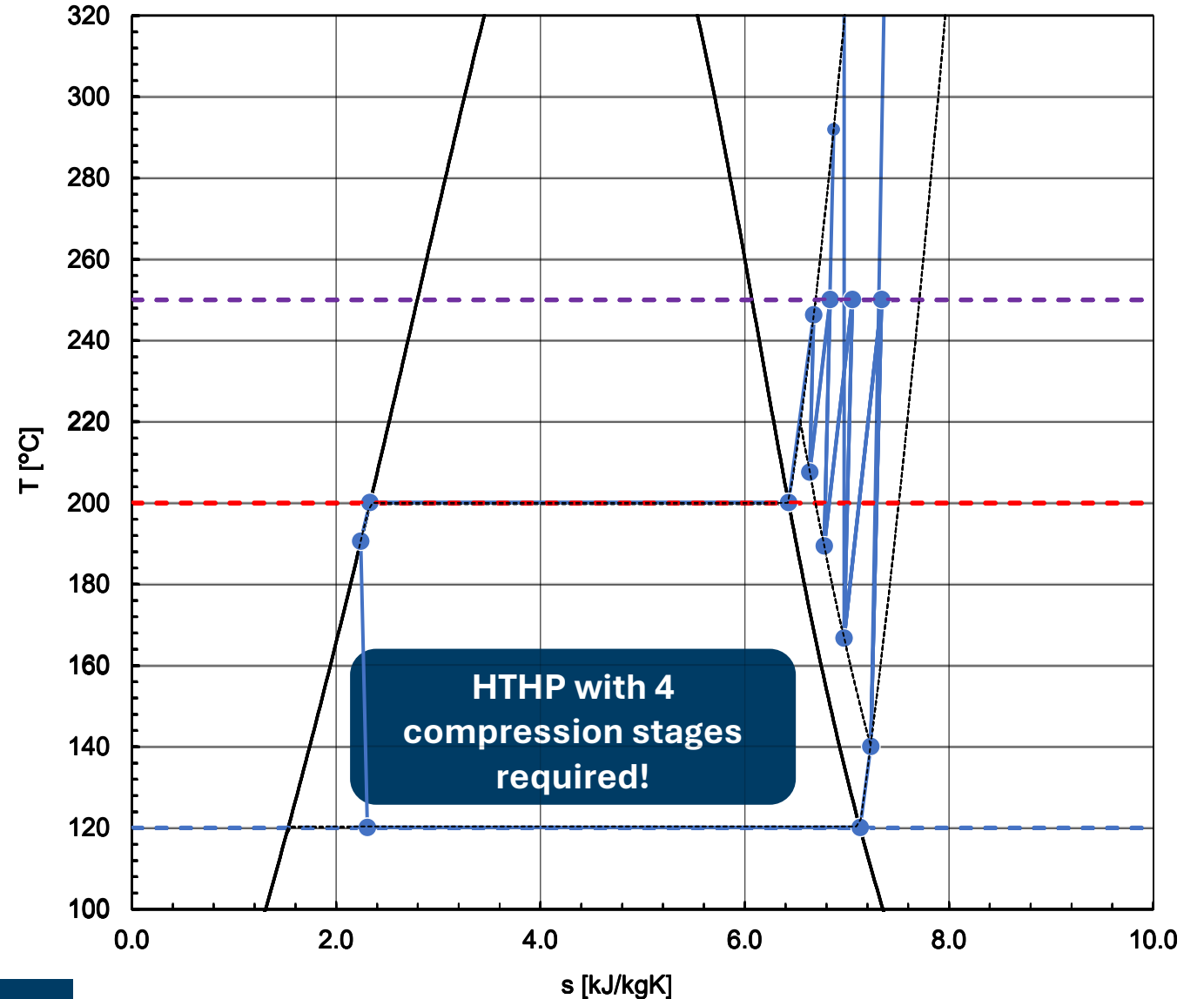
Key requirements

Heat Pump targets

- Evaporator 2 bara, 120°C
- Condenser: 15.5 bara, 200°C
- 80 Kelvin cycle temperature lift

Compressor

- Sealing material → Maximum discharge temperature: 250-270°C
- Cylinder Coating - No liquid presence in compressor → Minimum steam superheat at suction: 20 Kelvin



Optimization of stage cylinder bore diameter and pressure ratio

Specification of cylinder geometry

- Stage bore diameters and cylinder volume must be decided
 - Affects pressure ratio and temperature lift of each compression stage
- 4 compression stages lead to many possible configurations
- Procedure for optimization of bore diameter for maximum COP while maintaining thermodynamic constraints
 - Maximum stage discharge temperature (250 → 270°C)
 - Maximum condenser duty (50 kW)
 - Maximum bore diameter (90 mm)

$$D_b = \sqrt{\frac{240 \cdot \dot{V}_{in}}{\pi \cdot L_s \cdot RPM_{crank} \cdot N_{c,stage} \cdot \eta_{vol}}}$$

Optimization procedure

Constraints

- Max cylinder bore diameter (90 mm)
- Max $T_{\text{disch.}}$ 270°C
- Max condenser duty (50 kW)

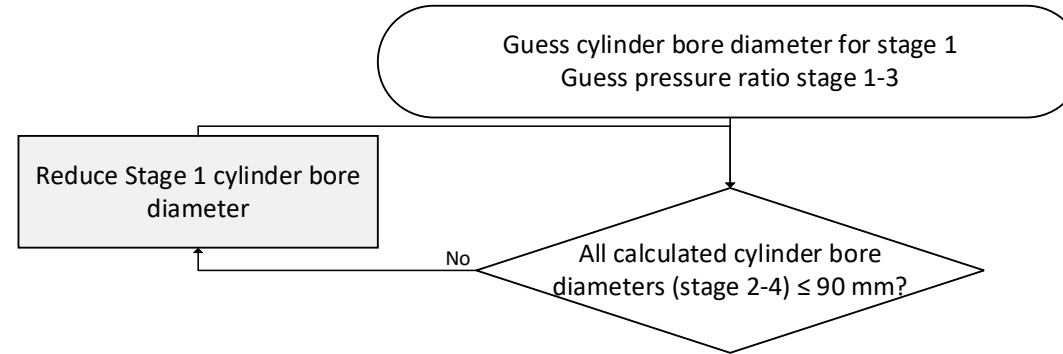
Guess cylinder bore diameter for stage 1
Guess pressure ratio stage 1-3

(Stage 4 pressure ratio calculated from P_{evap} , P_{cond})

Optimization procedure

Constraints

- Max cylinder bore diameter (90 mm)
- Max $T_{\text{disch.}}$ 270°C
- Max condenser duty (50 kW)

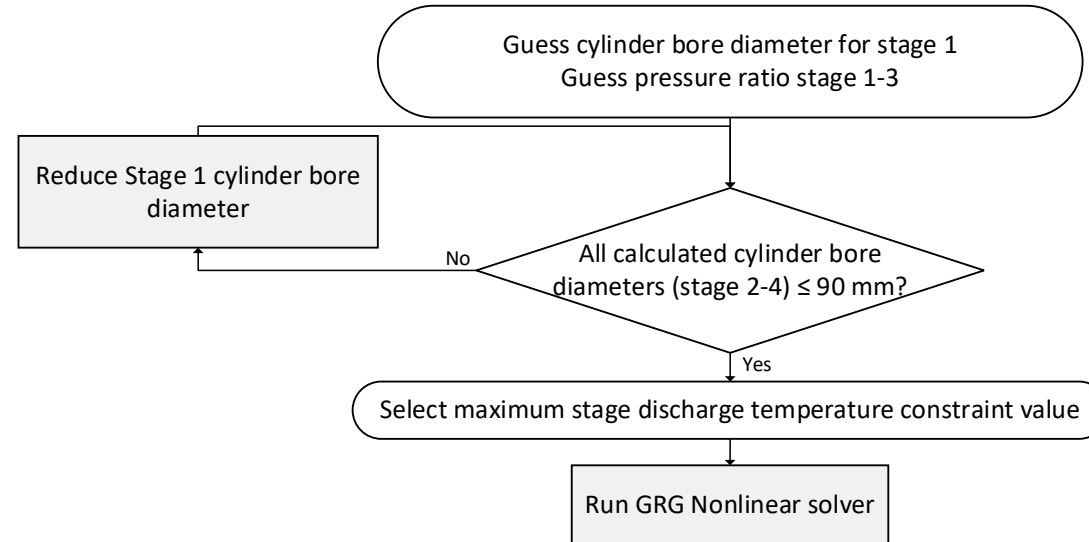


(Stage 4 pressure ratio calculated from P_{evap} , P_{cond})

Optimization procedure

Constraints

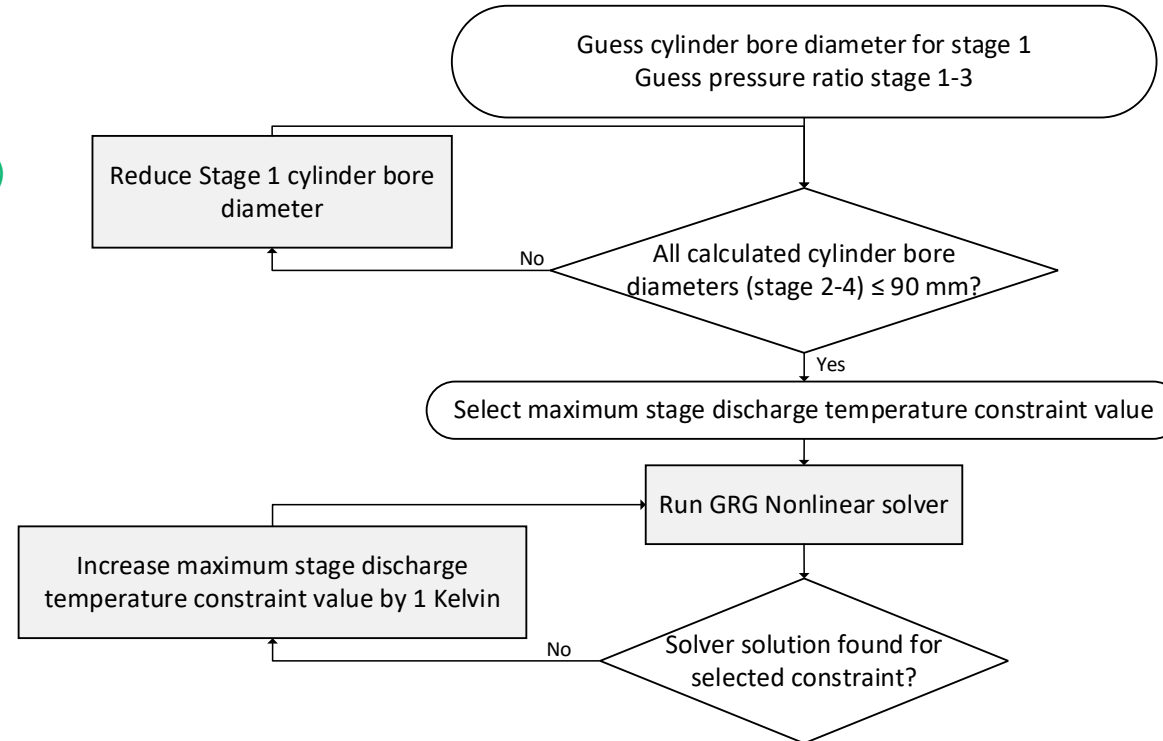
- ✓ Max cylinder bore diameter (90 mm)
- Max $T_{\text{disch.}}$ 270°C
- Max condenser duty (50 kW)



Optimization procedure

Constraints

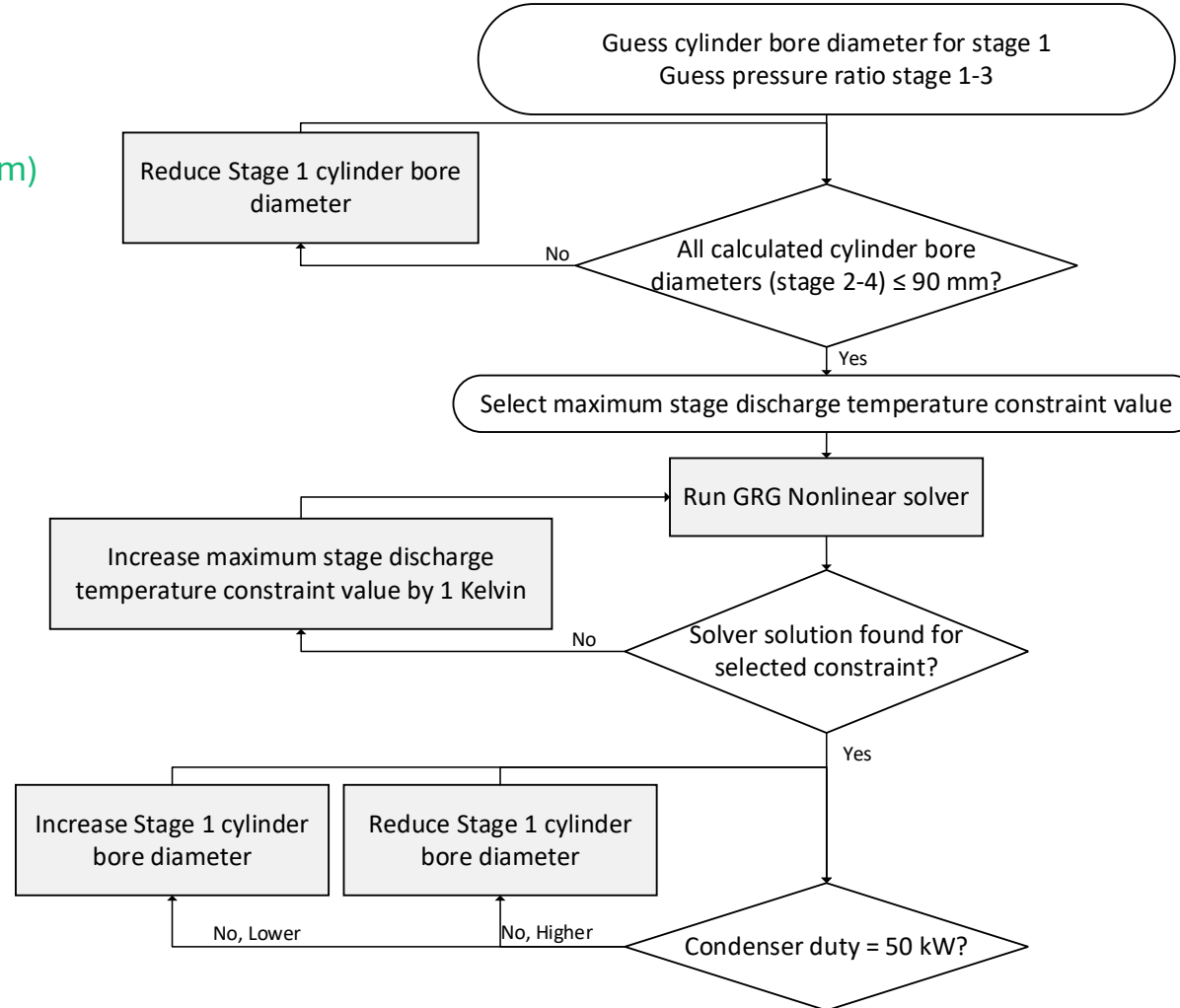
- ✓ Max cylinder bore diameter (90 mm)
- Max $T_{\text{disch.}}$ 250 - 270°C?
- Max condenser duty (50 kW)



Optimization procedure

Constraints

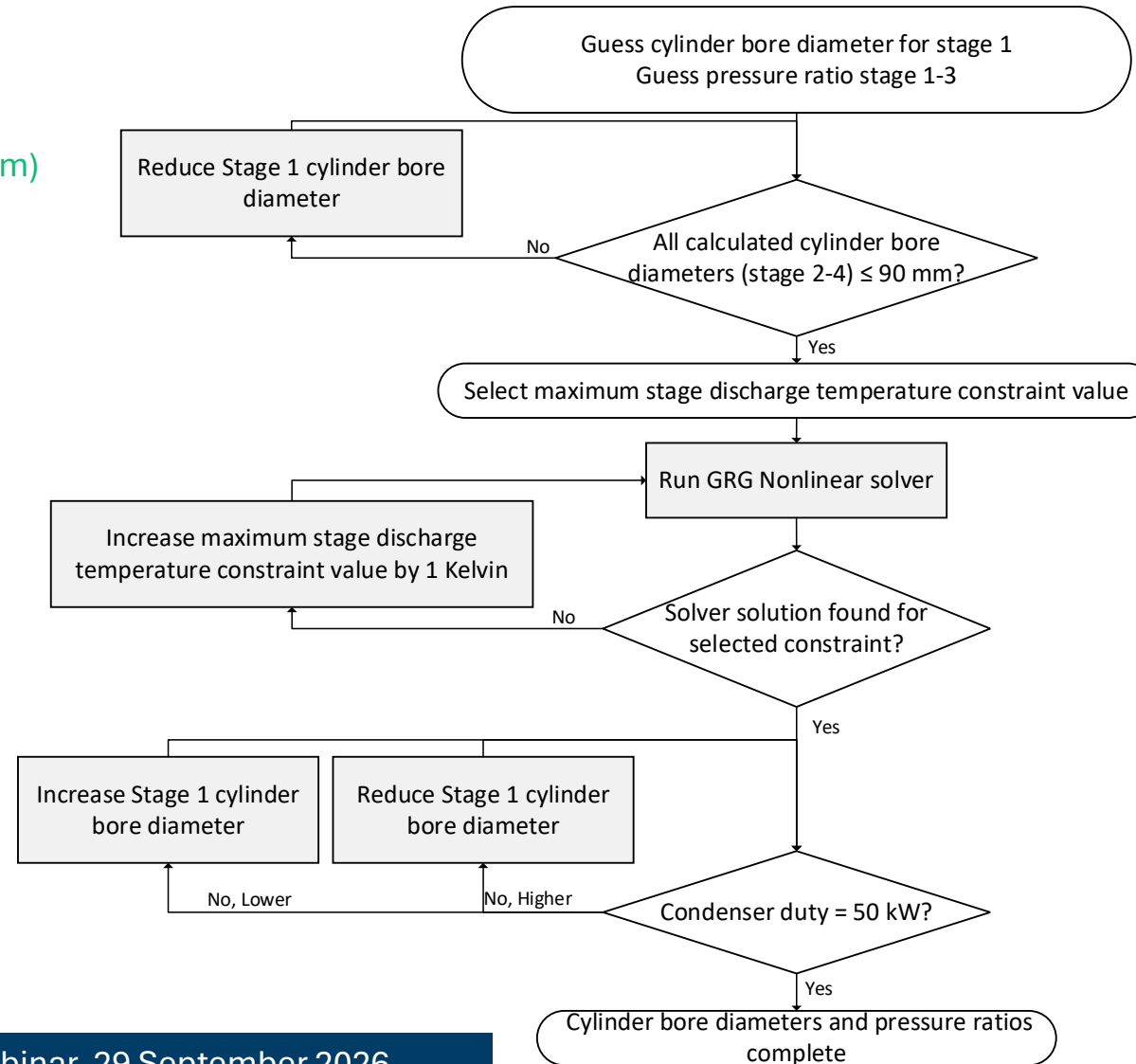
- ✓ Max cylinder bore diameter (90 mm)
- ✓ Max $T_{\text{disch.}}$ 250 - 270°C?
- Max condenser duty (50 kW)



Optimization procedure

Constraints

- ✓ Max cylinder bore diameter (90 mm)
- ✓ Max $T_{\text{disch.}}$ 250 - 270°C?
- ✓ Max condenser duty (50 kW)

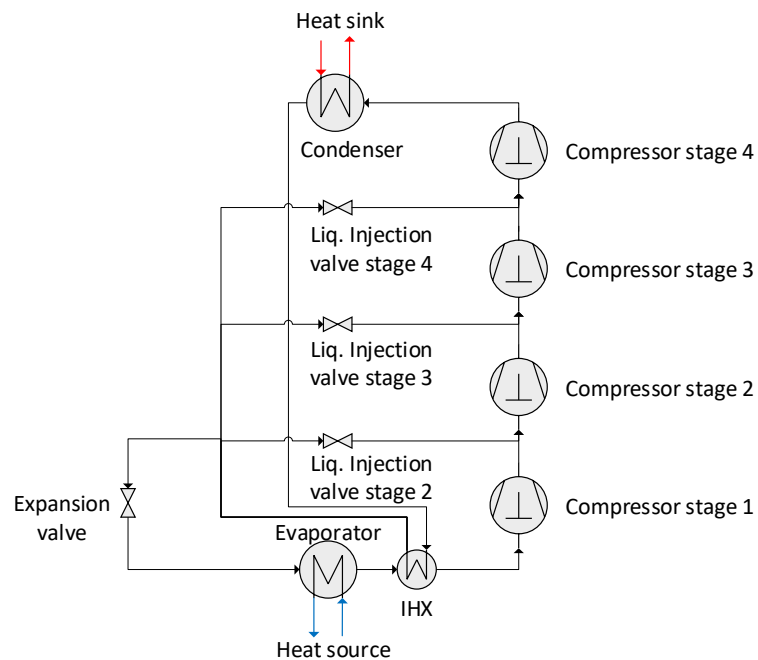


Evaluated HTHP layouts

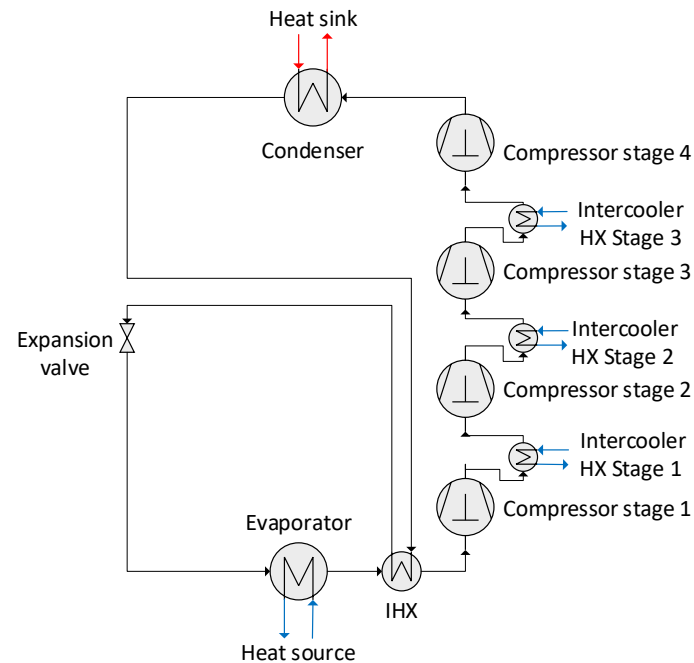
Key design aspects:

- Layouts differentiating based on steam desuperheating method
- Operational safety – Controllable desuperheating process minimizing risk of liquid presence in the compressor
- Performance and complexity

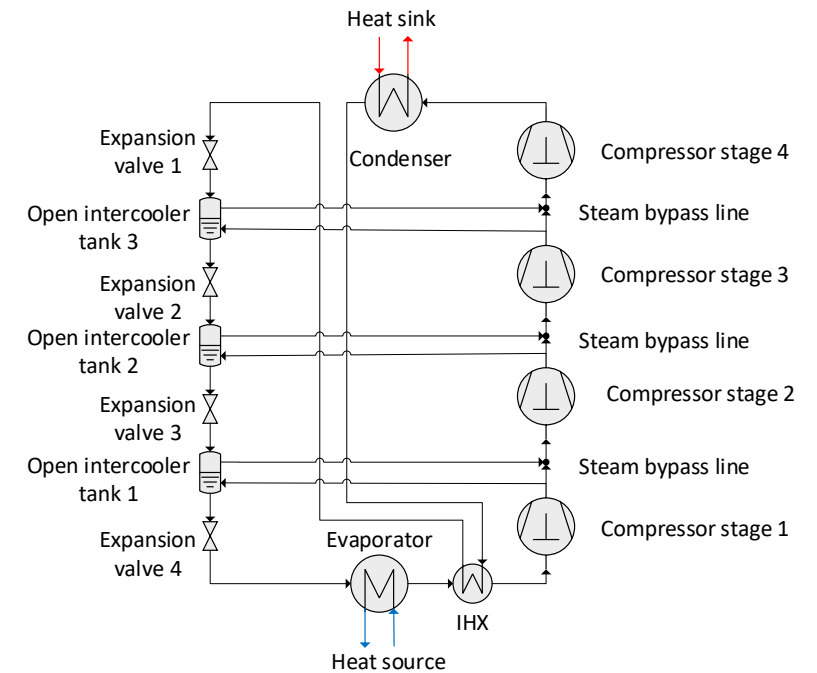
1 – Liquid spray injection



2 – Closed heat exchangers



3 – Open intercooler tanks



Layout 1 – liquid spray injection

Concept

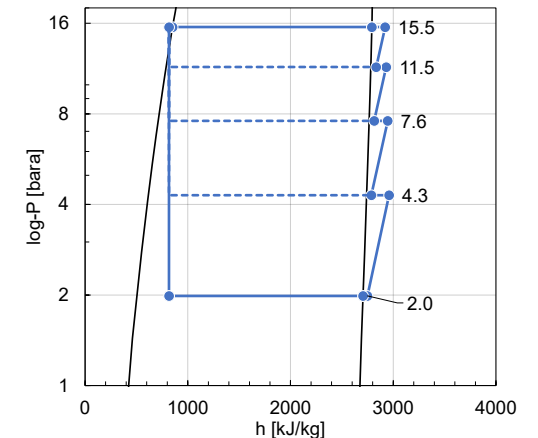
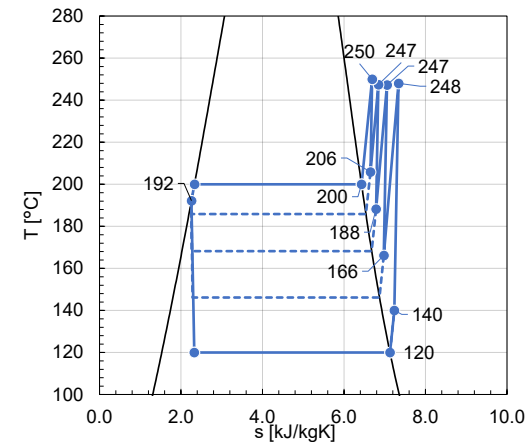
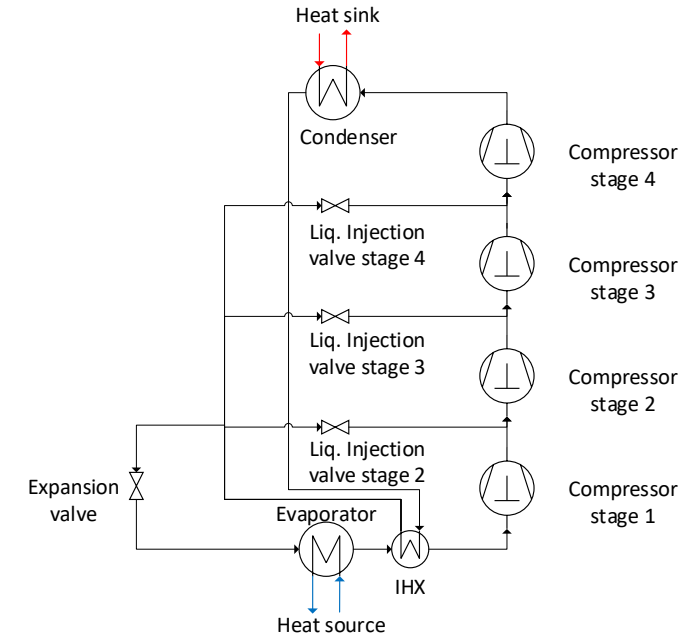
- Interstage desuperheating by injection of pressurized condensate
- Liquid expanded through control valves and atomized through spray nozzles

Advantages

- Relatively simple - No external coolant circuit required
- Utilization of flash vapor improves COP
- Established technology within industrial steam systems

Challenges

- Risk of incomplete evaporation and liquid droplets entering the compressor
- Commercial desuperheater injectors have limited flow turndown capability
- Requires accurate dynamic flow control of the injected liquid



Layout 2 – Closed heat exchangers

Concept

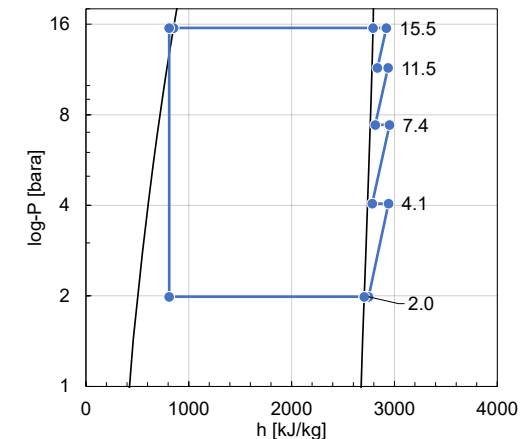
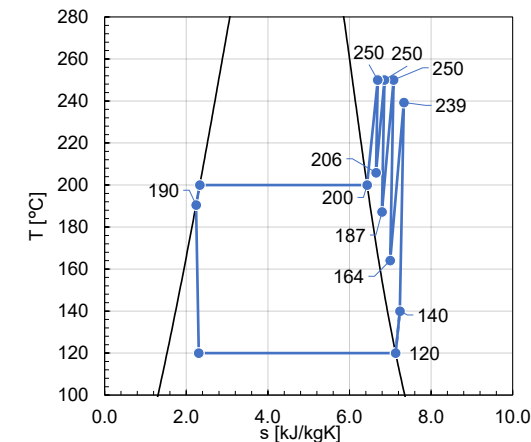
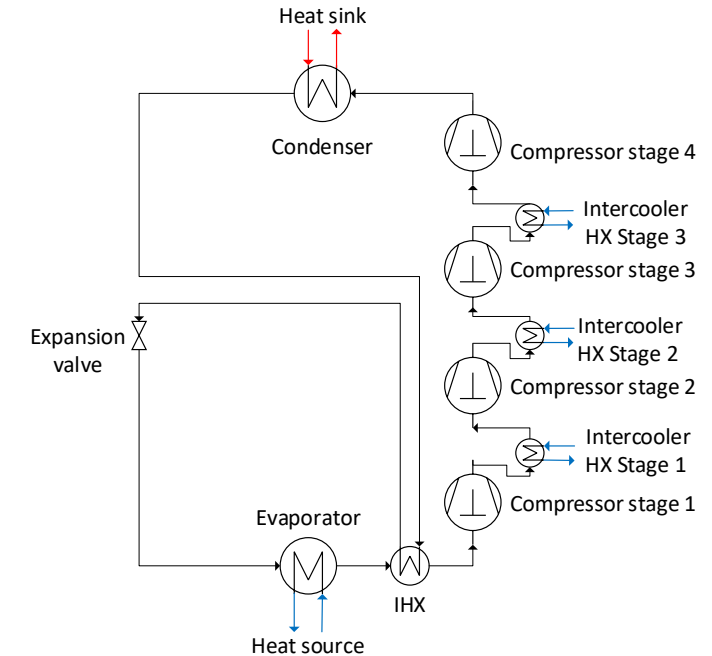
- Interstage desuperheating using closed heat exchangers and an auxiliary coolant
- Working fluid and coolant remain separated

Advantages

- Avoids direct liquid injection
- Significantly reduces risk of liquid droplets entering the compressor
- Enables use of air cooling during start-up
- Few cycle components, potentially compact

Challenges

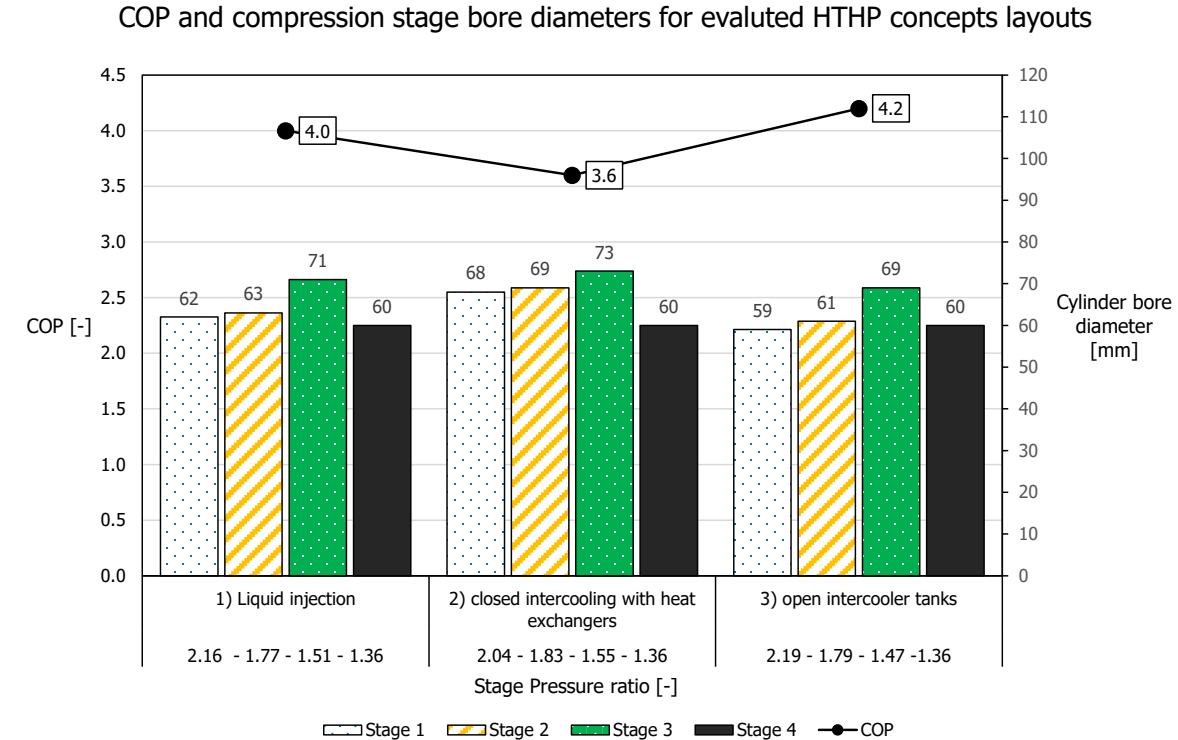
- Heat removed from the HTHP cycle unless recovered elsewhere
- Additional auxiliary cooling circuit is required → More auxiliary equipment, piping, valves and instrumentation
- Possible condensation on heat-exchanger surfaces must be avoided



Performance comparison

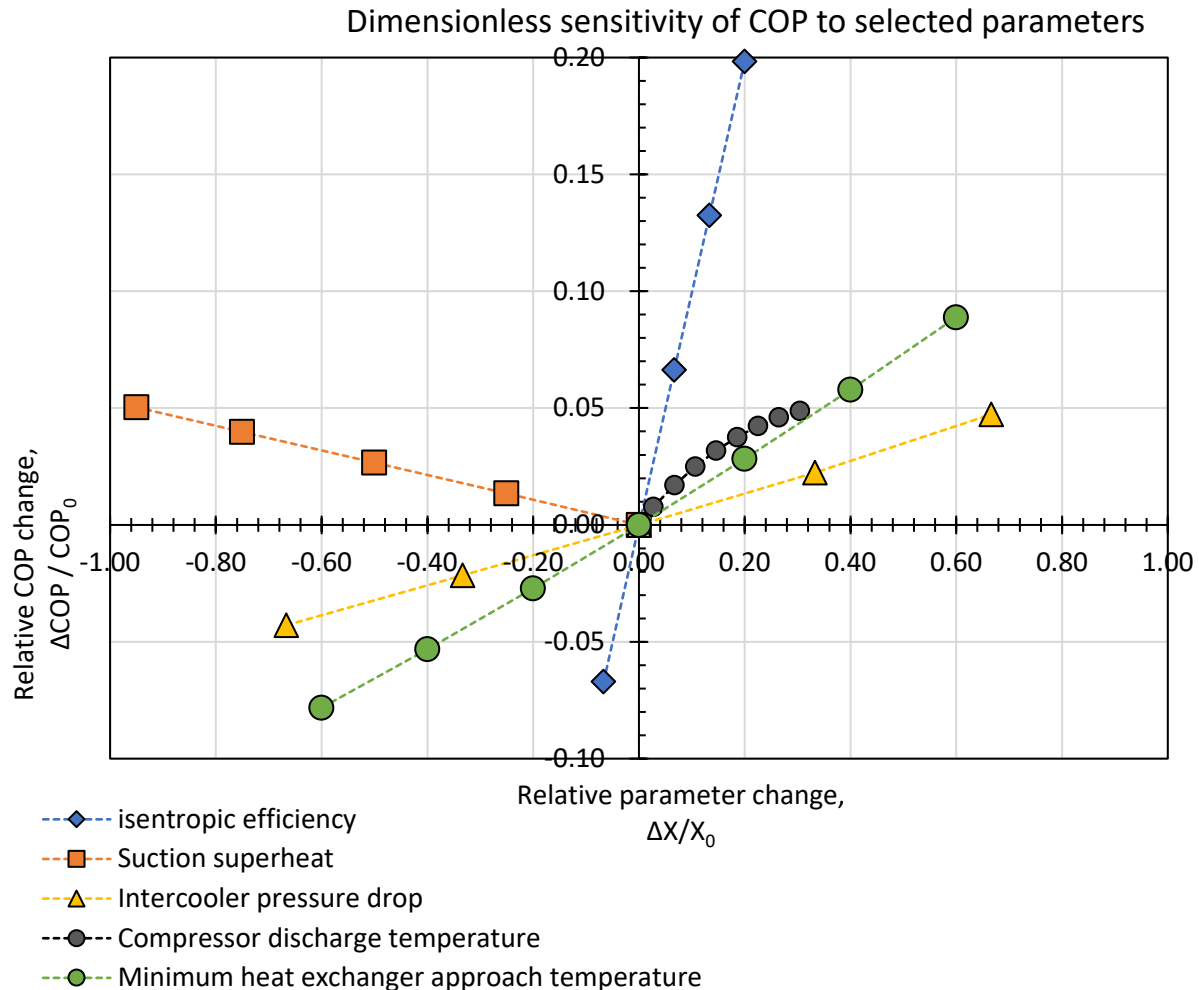
Steady state evaluation at design point conditions

- Key compressor constraints satisfied for all layouts
 - Maximum compressor discharge temperature 250°C
- Utilization of flash vapor leads to significantly improved COP
 - Best effect is with open intercooler tanks, COP = 4.2, $\eta_{\text{Carnot}} = 0.63^1$
- Different layouts lead to wide differences in optimum cylinder bore diameters and stage pressure ratio
 - Reduced bore diameters in stages 1-3 for concepts 1 and 3 leading to reduced compressor duty
 - Highest pressure ratio for stage-1, then progressively decreasing for subsequent stages



Performance sensitivity analysis

On selected layout to evaluate influence from key parameters



Parameter	Baseline value X_0	Range
Isentropic efficiency, η_{is} [-]	0.75	0.7 – 0.9
Maximum compressor T_{disch} [°C]	250, or optimization min.	250 – 330
Suction superheat [K]	20	5 – 20
Desuperheating HX pressure drop [bara]	0.3	0.1-0.5
Condenser and evaporator ΔT_{min} [K]	5	2-8

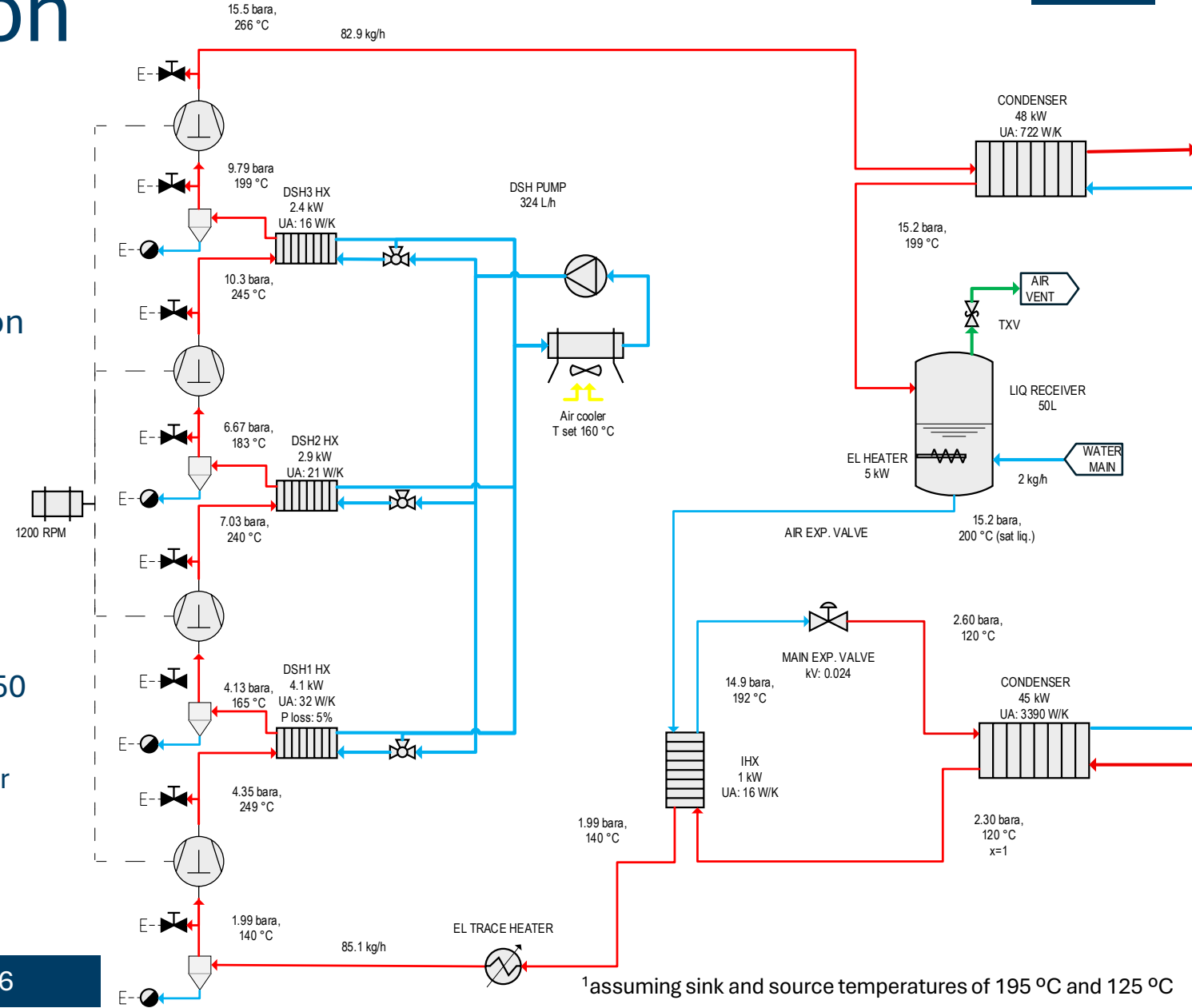
Results (Baseline COP = 3.33):

- Isentropic efficiency has the **largest impact** on COP
 - Example 0.75 $\rightarrow$ 0.85, COP = 3.79
- Reducing suction superheat provides **limited performance gains**
 - Example 20K $\rightarrow$ 5K, COP = 3.49
- Increasing allowable discharge temperature shows **diminishing returns**
 - Example: 250°C $\rightarrow$ 300°C, COP = 3.46
- Heat exchanger design (minimum approach temperature and pressure drop) has **significant combined impact**):
 - Example: 5K $\rightarrow$ 3K and 0.3 bar $\rightarrow$ 0.15 bar, COP = 3.65

Volumetric efficiencies and interstage pressure losses included

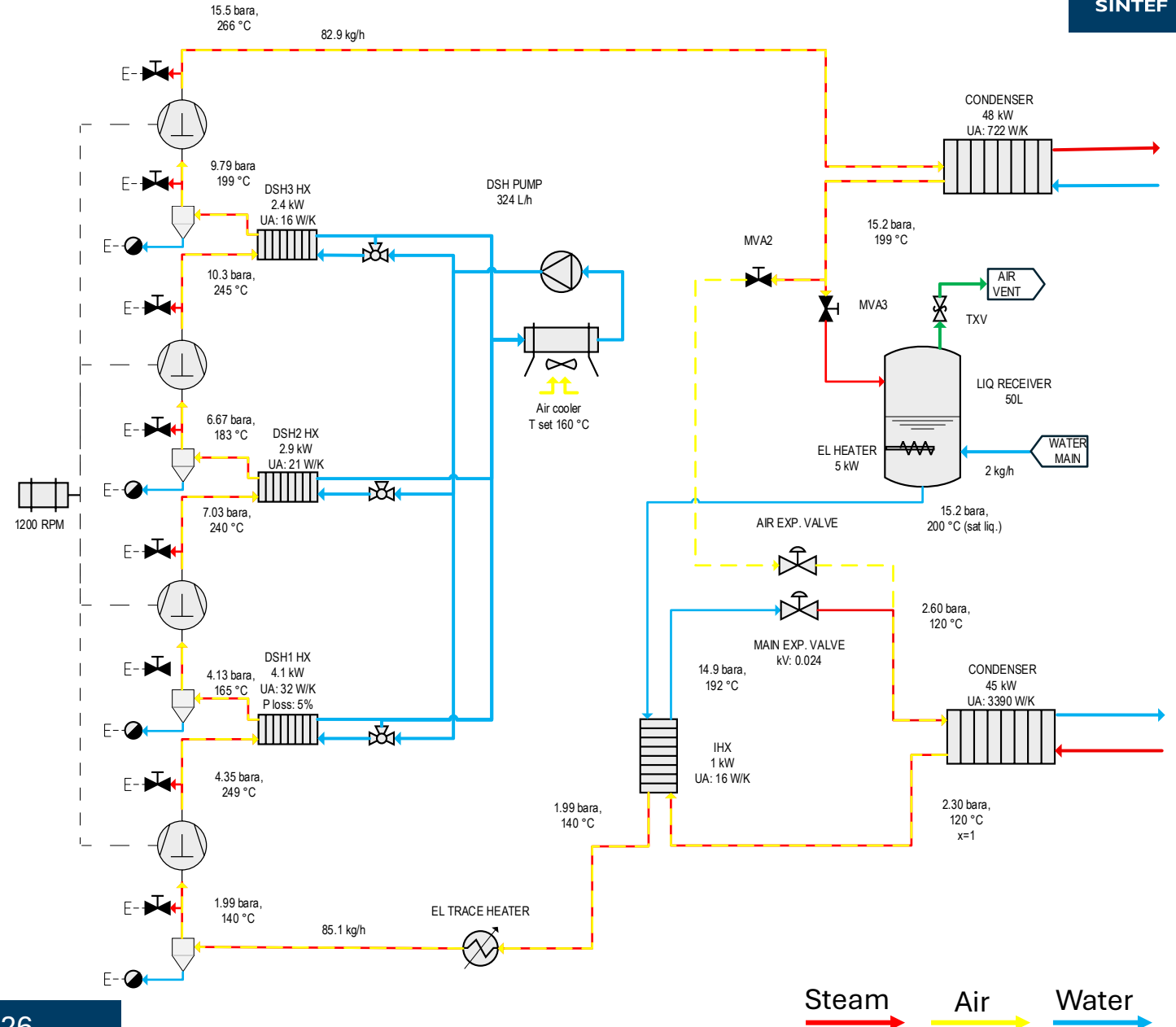
Final layout selection and performance

- Based on concept with closed heat exchanger intercooling
 - Lower operational risk than liquid injection
 - Lower system complexity than open intercooler tanks
 - Allows intercooling of air during pre-heating
- Slight revision of bore diameters due to manufacturing constraints:
 - 80 – 80 – 90 - 75 mm
- Performance @ design¹: $COP = 3.36$, $\eta_{Carnot} = 0.50$
 - Max. $T_{disch.}$ increased to 266 °C
 - < 1% COP reduction compared to optimal cylinder bore diameter



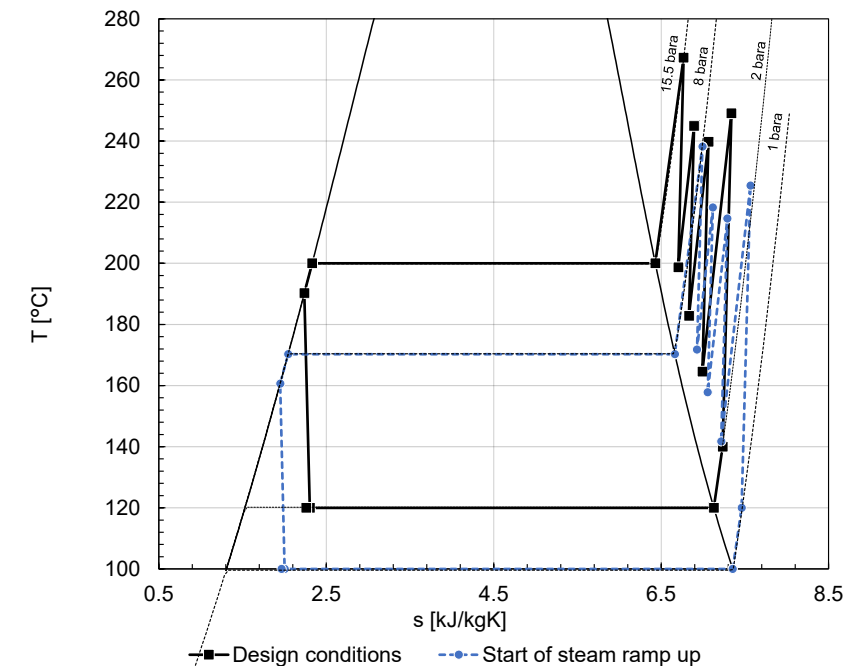
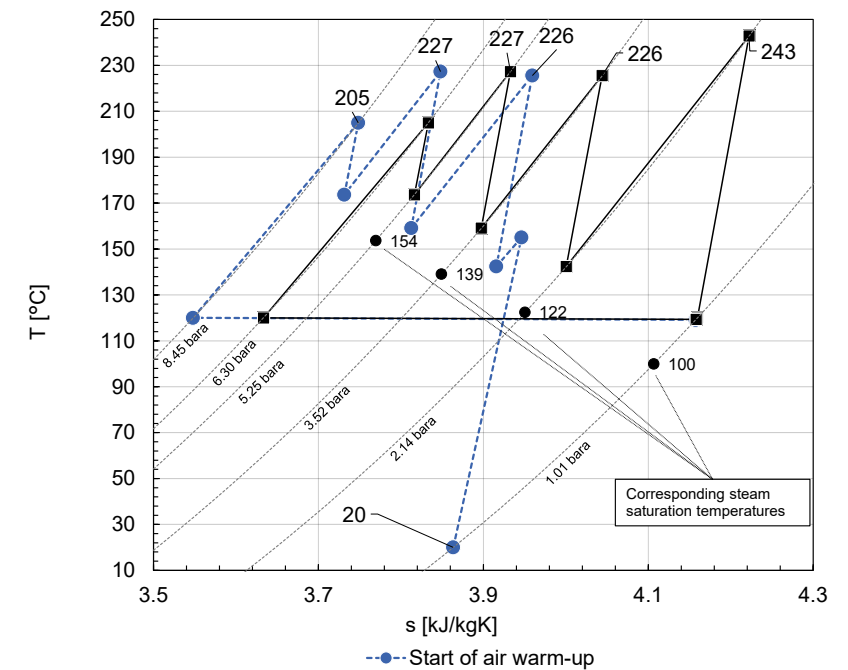
Start-up strategy

- Controlled pre-heating with air until design point temperatures
- Introduction of steam via HP liquid receiver
- Additional liquid risk mitigating actions
 - High-temperature desuperheating water coolant loop to avoid HX wall temperatures below sat. steam
 - Steam traps and separators before each compression stage



Start-up strategy – cycle temperature evaluation

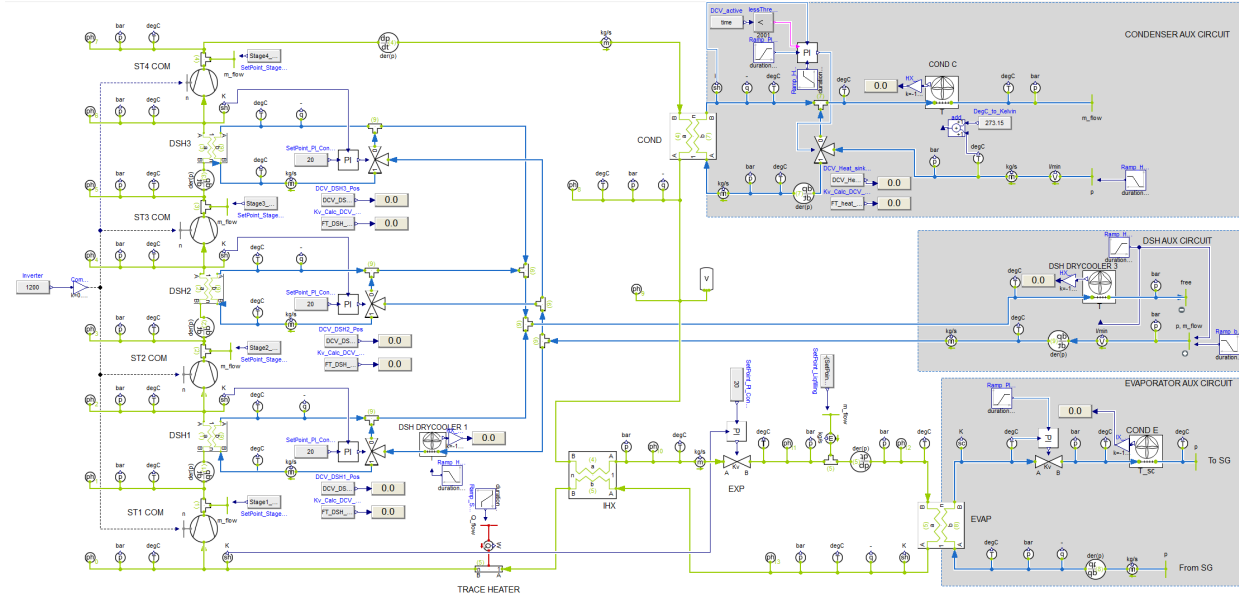
- Air compression gives a **larger temperature rise than steam**, resulting in high discharge temperatures
- Solution: reduce the **Stage 1 inlet pressure and temperature target to 1 atm, 120 °C**
- Introduce steam at prevailing pressures once all compressor-stage inlet temperatures reach the **local steam saturation temperature + 20 K**
- Cycle pressures are then gradually ramped to design conditions



Dynamic simulations

Dymola model of HTHP and auxiliary systems

- Validate HTHP design, operational strategy and dimensioning of components



Conclusion and outlook

- Development of 4-stage R-718 heat pump can lead to high layout complexity and many possible configurations
- Reciprocating compressor sensitive to liquid droplets and high temperatures - Key focus on design and operational risk reducing measures
 - Desuperheating concepts
 - Optimization of cylinder geometry and pressure ratios - significant variations between layouts
- Closed intercooling offers the best balance of performance, safety and complexity
 - Selected layout achieves $\text{COP} = 3.4$ and $\eta_{\text{Carnot}} = 0.50$
 - Can be integrated with air-preheating
- Pushing compressor superheat and temperature limit constraints yields poor risk–reward

Next steps in the Geosyn project

- Compressor testing and system laboratory experiments → Pilot demonstration



Funded by
the European Union

Ole Marius Moen, Research scientist, SINTEF Energy Research
Ole.moen@sintef.no

Contributors: Krzysztof Banasiak, Joshua R. Dowdell, Sigurd Sannan, Vaishak Somasundaram, Paride Gullo, Lorenzo Talluri, Andrea Zini, Adriano Milazzo, Giuseppe Vitri, Filippo Luchetti, Mauro Dallai

