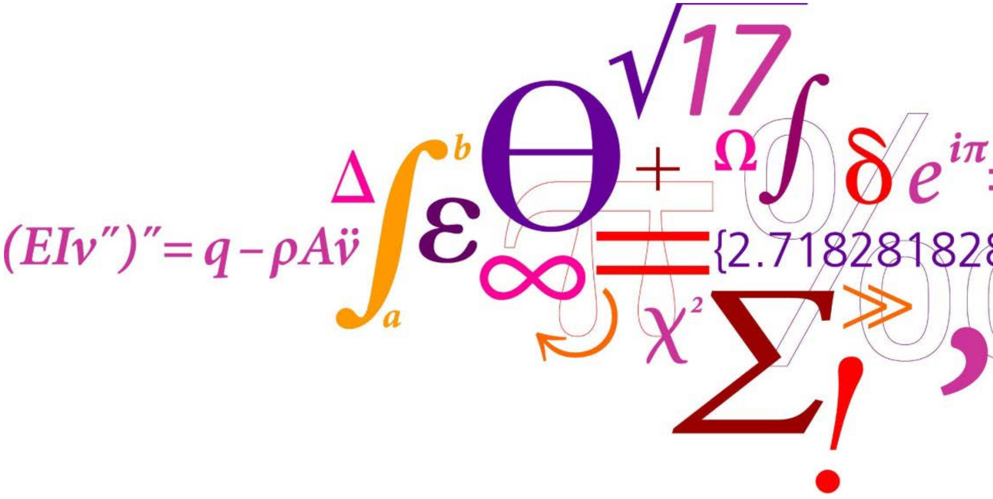
Multi-objective optimization of organic Rankine cycles using pure and mixed working fluids

*Technical University of Denmark,
**Chalmers University of Technology

Authors:
Jesper Graa Andreasen*
Martin Ryhl Kærn*
Leonardo Pierobon*
Ulrik Larsen**
Fredrik Haglind*

Email: jgan@mek.dtu.dk
Phone: +45 4525 4123

DTU Mechanical Engineering
Department of Mechanical Engineering

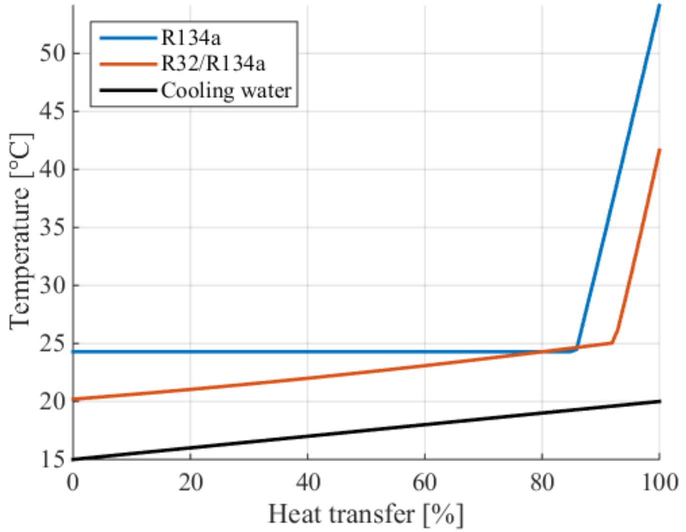


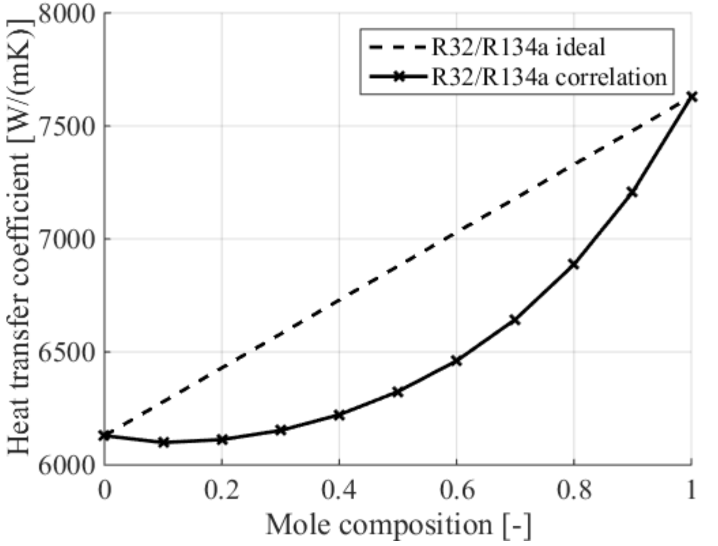
Agenda

- Motivation of the study
- Methodology
  - Case study and models
  - Optimization framework
- Results
- Conclusions

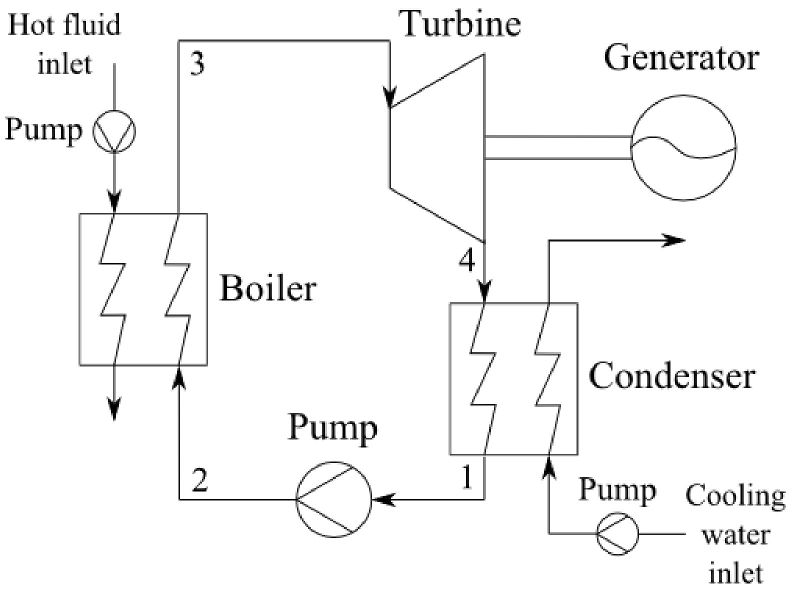
Motivation of the study

- Characteristics of zeotropic mixtures as working fluids:
    - o Reduced heat transfer irreversibilities
    - o Increased heat transfer area due to lower mean temperature difference
    - o Increased heat transfer area due to degradation of heat transfer coefficient





ORC power plant



Parameter description	Value	Unit
Hot fluid (water)		
Hot fluid inlet temperature	90	°C
Hot fluid mass flow	50	kg/s
Hot fluid pressure	4	bar
Condenser		
Cooling water inlet temperature	15	°C
Outlet vapour quality	0	-
Cooling water pressure	4	bar
Working fluid pump		
Isentropic efficiency	0.8	-
Auxiliary pumps		
Isentropic efficiency	0.7	-
Turbine		
Isentropic efficiency	0.8	-
Minimum outlet vapour quality	1	-

Working fluids: R32, R134a and R32/R134a $(0.65/0.35_{\text{mole}})$

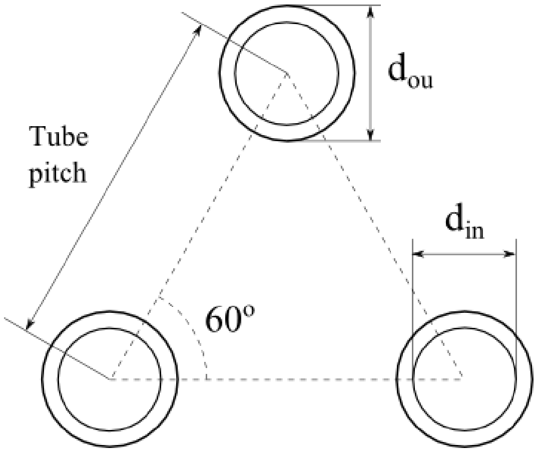
Objective functions:

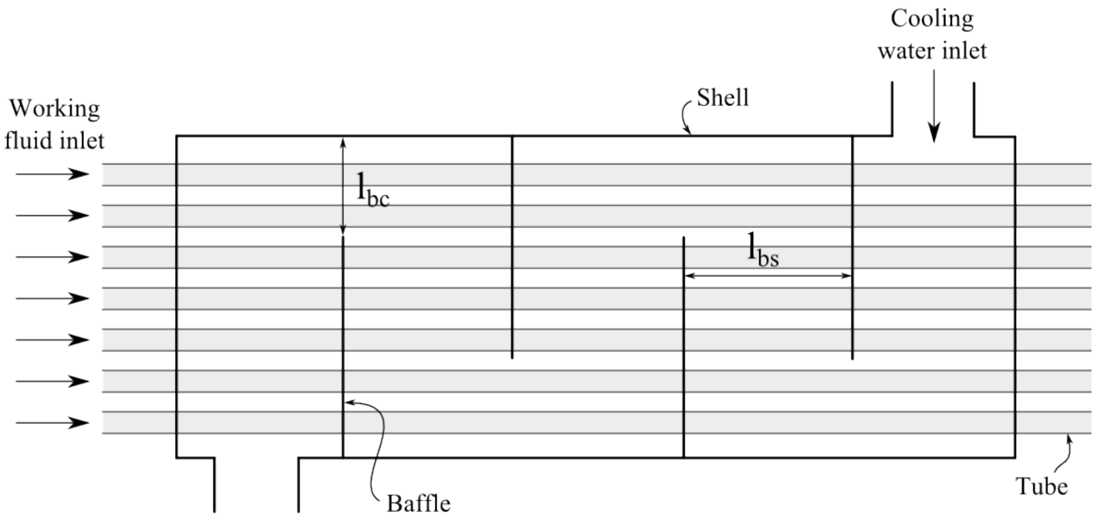
$$\dot{W}_{NET} = \dot{m}_{wf}(h_3 - h_4 - (h_2 - h_1)) - \dot{W}_{aux.,pumps}$$

$$C_{tot} = C_{turb} + C_{wf,pump} + C_{cond} + C_{boil} + C_{gen} + C_{aux.,pumps}$$

Heat exchanger models

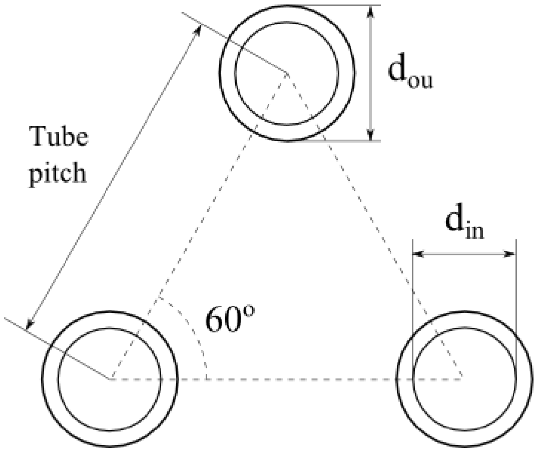
- Shell-and-tube heat exchangers
    - Working fluid in the tubes
    - Triangular tube configuration
    - One tube and one shell pass
    - Bell-Delaware method

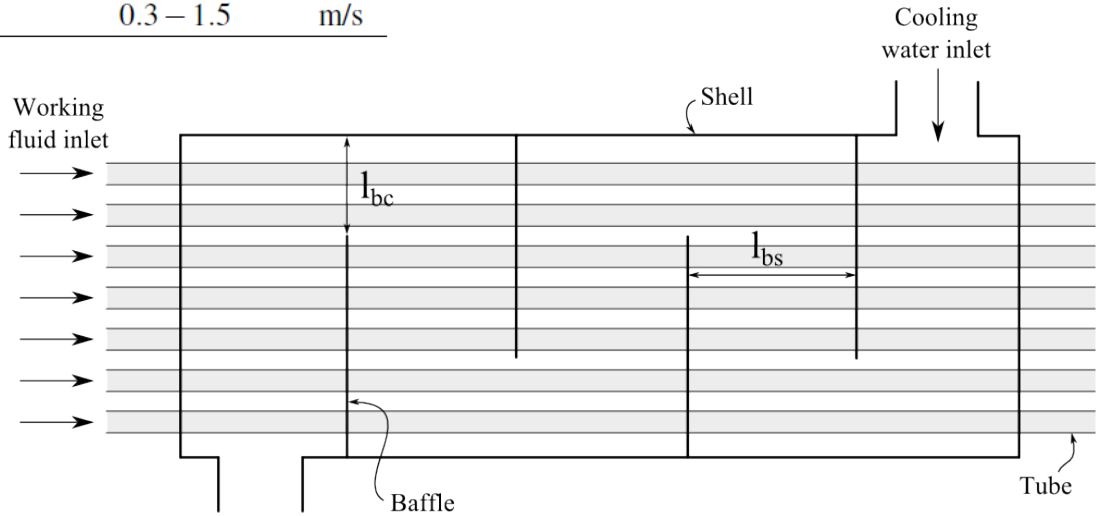




Heat exchanger models

Parameter description	Value/range	Unit
Tube configuration	Triangular 60°	-
Tube thickness	3	mm
Tube pitch	$1.5 \cdot d_{ou}$	mm
Baffle cut	$0.25 \cdot d_s$	mm
Tube wall conductivity	16	W/mK
Number of control volumes	30	-
Condenser velocities		
Tube side inlet	5 – 22	m/s
Tube side outlet	0.5 – 4	m/s
Shell side inlet	0.3 – 1.5	m/s
Boiler velocities		
Tube side inlet	0.9 – 4	m/s
Tube side outlet	5 – 22	m/s
Shell side inlet	0.3 – 1.5	m/s





Heat transfer correlations

- Single-phase heat transfer, Gnielinski
- Condensation heat transfer, Shah

$$\alpha_{2p,cond} = \begin{cases} \alpha_I & J_g \geq 0.98(Z+0.263)^{-0.62} \\ \alpha_I + \alpha_{Nu} & J_g < 0.98(Z+0.263)^{-0.62} \end{cases}$$

$$\alpha_I = \alpha_{LT}\left(\frac{\mu_L}{14\mu_V}\right)^{0.0058+0.557P_r}\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{P_r^{\,0.38}}\right]$$

$$\alpha_{Nu} = 1.32Re_L{}^{-1/3}\left[\frac{\rho_L(\rho_L-\rho_V)g\lambda_L{}^3}{\mu_L{}^2}\right]^{1/3}$$

- Bell-Ghaly correction implemented for condensation of mixtures

Heat transfer correlations

- Boiling heat transfer correlation, Gungor & Winterton + Thome

$$\alpha_{2p,boil} = \alpha_L \left[1 + 3000(BoF_c)^{0.86} + 1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_L}{\rho_V}\right)^{0.41}\right]$$

$$F_c = \left[1 + \left(\frac{\alpha_{nb,id}}{q_{nb}}\right)(T_{dew} - T_{bub})\left[1 - \exp\left(\frac{-Bq_{nb}}{\rho_L h_{LV} \beta_L}\right)\right]\right]^{-1}$$

- Heat transfer correlation for flow across tube bundles, Martin
- Single-phase pressure drop, Blasius
- Two-phase pressure drop, Müller-Steinhagen & Heck

Cost correlations

- Single stage axial turbine, Astolfi et al.

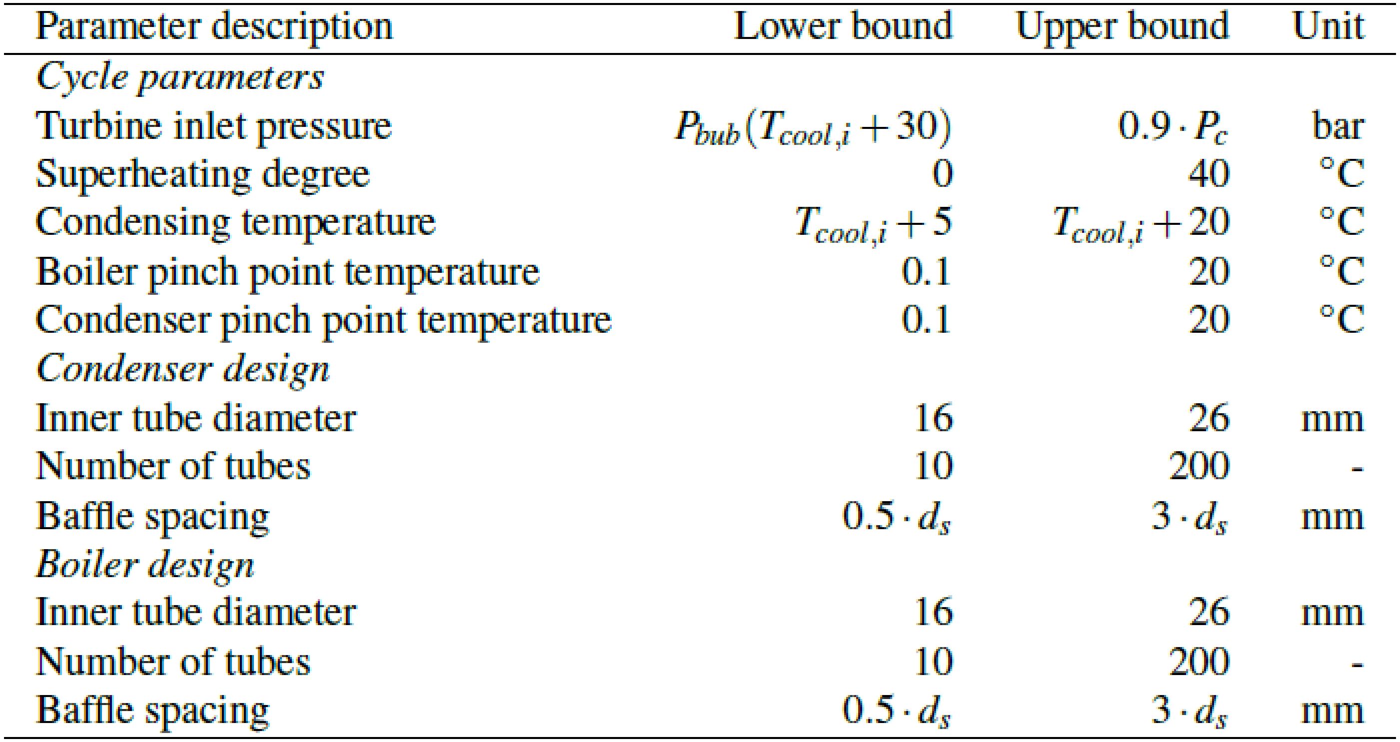
$$C_{turb} = 1.230 \cdot 10^6 \left(\frac{1}{2}\right)^{0.5} \left(\frac{\sqrt{\dot{V}_4}/(\Delta h_{is})^{0.25}}{0.18}\right)^{1.1}$$

- Remaining components, Smith

$$C_E = C_B \left(\frac{Q}{Q_B}\right)^M f_M f_P f_T$$

Component	C_B	Q_B	M	f_M	f_P	f_T	Reference
Heat exchangers	32.8 k$	80 m^2	0.68	1.7	(Smith, 2005)	1	(Smith, 2005)
Pumps	9.48 k$	4 kW	0.55	1	1	1	(Smith, 2005)
Generator	3.7 k$	1000 kW	0.95	1	1	1	(Boehm, 1987)

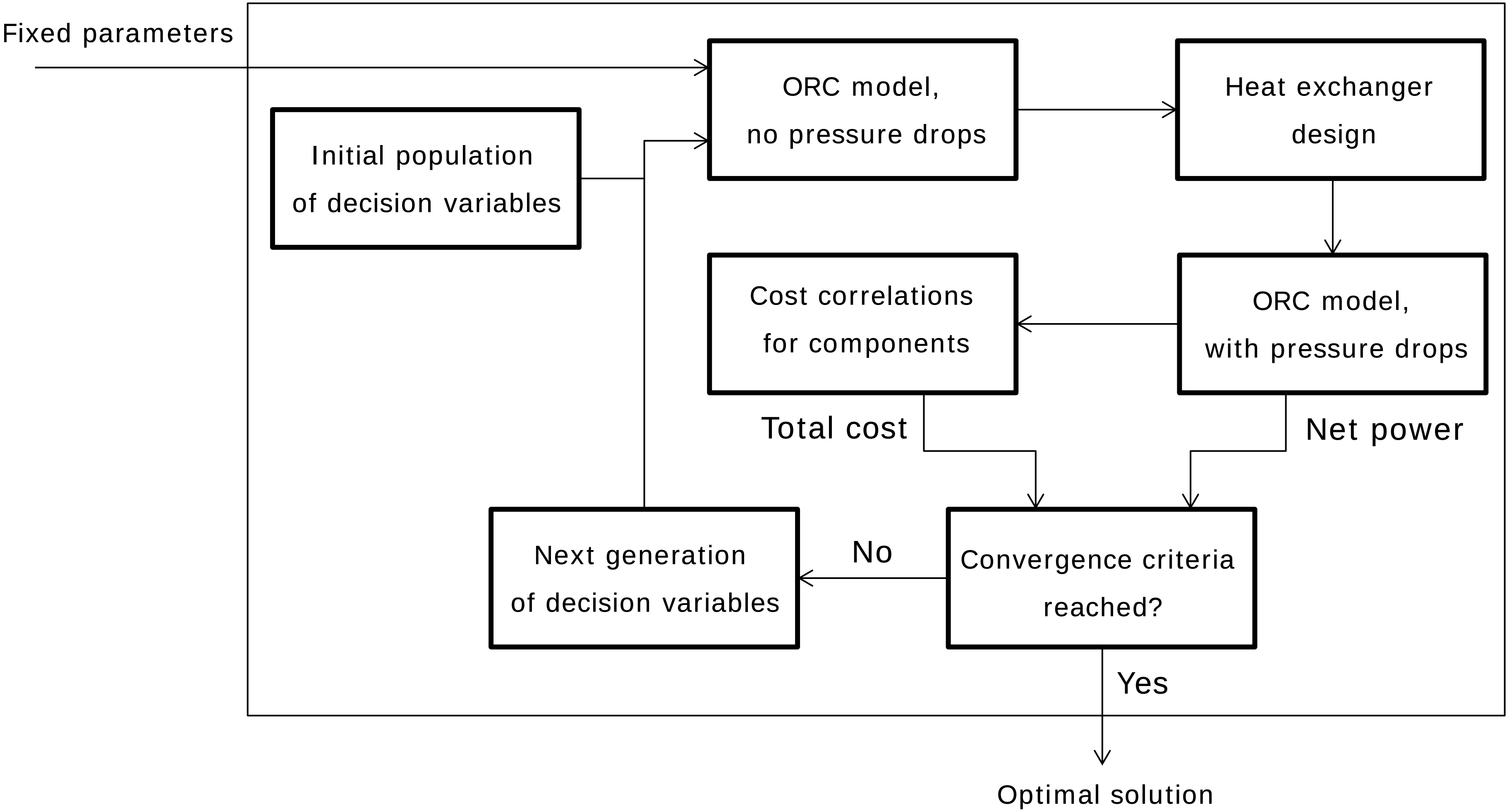
Optimization framework, decision variables



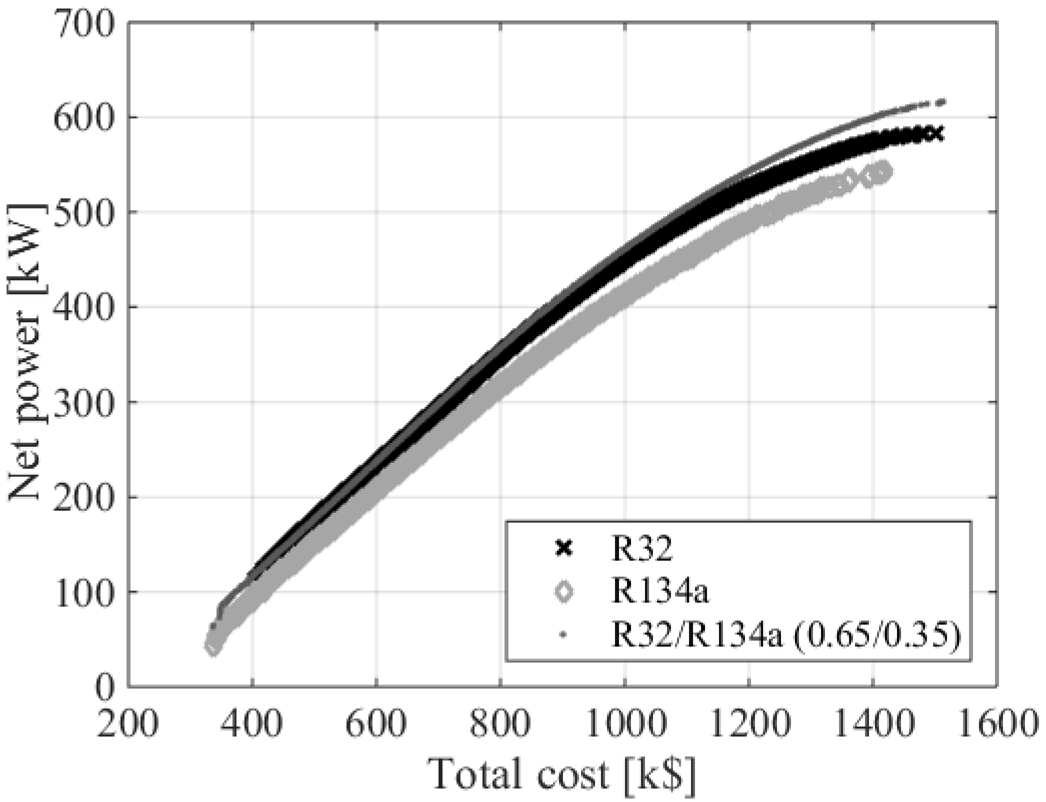
Parameter description	Lower bound	Upper bound	Unit
Cycle parameters			
Turbine inlet pressure	$P_{bub}(T_{cool,i}+30)$	$0.9 \cdot P_c$	bar
Superheating degree	0	40	°C
Condensing temperature	$T_{cool,i}+5$	$T_{cool,i}+20$	°C
Boiler pinch point temperature	0.1	20	°C
Condenser pinch point temperature	0.1	20	°C
Condenser design			
Inner tube diameter	16	26	mm
Number of tubes	10	200	-
Baffle spacing	$0.5 \cdot d_s$	$3 \cdot d_s$	mm
Boiler design			
Inner tube diameter	16	26	mm
Number of tubes	10	200	-
Baffle spacing	$0.5 \cdot d_s$	$3 \cdot d_s$	mm

Optimization framework

Genetic algorithm with: population size: 500, generations: 30000

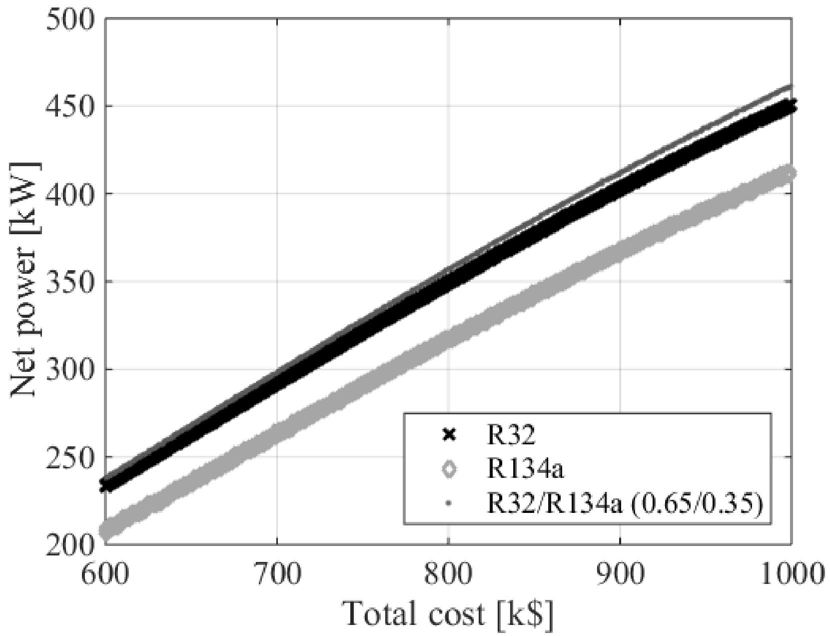


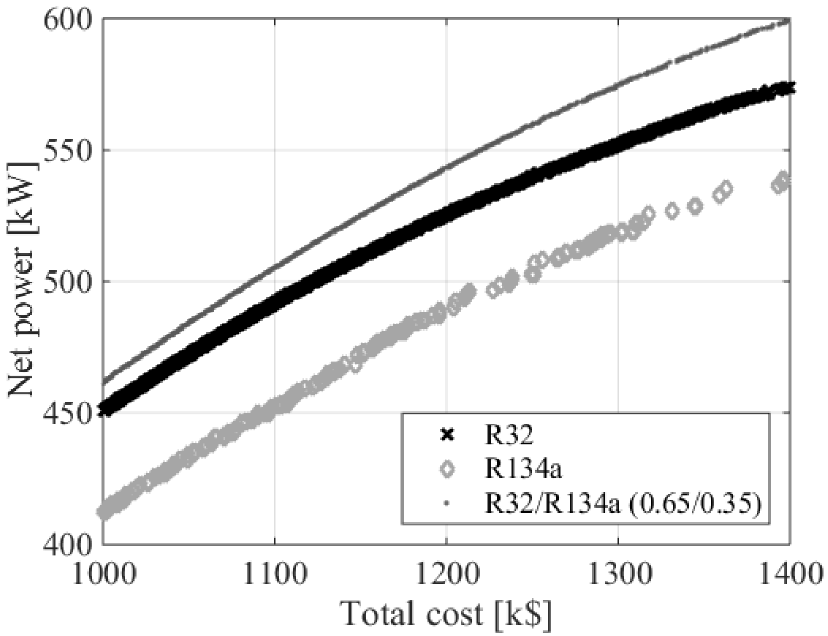
Results, net power vs. cost of components



Results, net power vs. cost of components

- At 800 k$, the mixture achieves 2.1 % higher net power than R32
- At 1200 k$, the mixture achieves 3.4 % higher net power than R32
- For an optimization of net power only the difference is 13.6 %

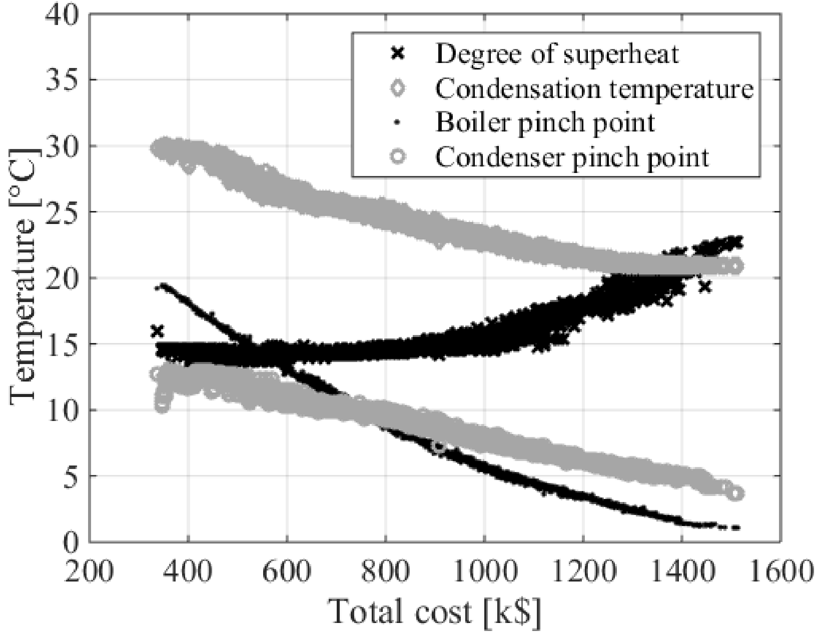




Results, optimized temperatures

- Condensation temperature and boiler pinch point decrease with increasing cost
- The condenser pinch point is constrained by the condensation temperature
- The degree of superheat increases with increasing cost

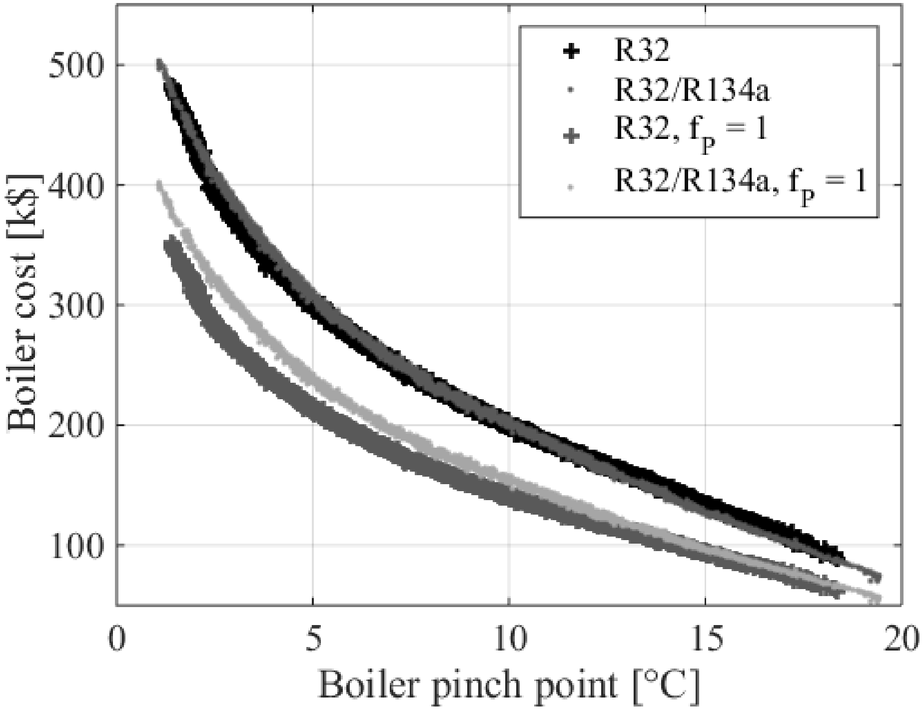
Fluid: R32/R134a



- Temperature glide of condensation is 5.3 ^{o}C, while the cooling water temperature increase ranges from 5.7 ^{o}C to 14.5 ^{o}C

Results, boiler costs and pressure correction factor

- For R32 the boiler pressure ranges from 39 to 45 bar
- For R32/R134a the boiler pressure ranges from 28 to 35 bar



Conclusion

- Higher performance (2-4 %) for R32/R134a compared to R32
- The performance benefit of the mixture is higher (13.8 %) when the comparison is based solely on net power output
- Lower pressures in the heat exchangers for R32/R134a compared to R32 reduces the negative effects of heat transfer degradation
- The cooling water temperature increase was found to be higher than the temperature glide of condensation

Acknowledgements

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Innovation Fund Denmark
RESEARCH, TECHNOLOGY & GROWTH

Thank you for your attention!