Optimal design of supercritical CO_2 (S-CO_2) cycle systems for internal combustion engine (ICE) waste-heat recovery

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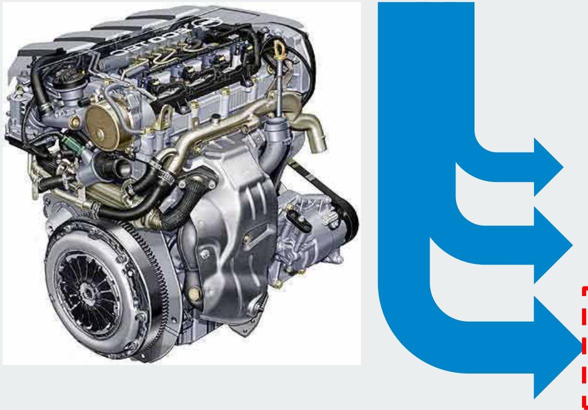
Outline

- Background
- System configuration
- Methodology and models
- Results and discussion
- Conclusions

Background

- Internal combustion engines
  - Vehicle motive-power machines
  - Industrial equipment
  - Small power units



Friction loss (10%)

Power output (30-45%)

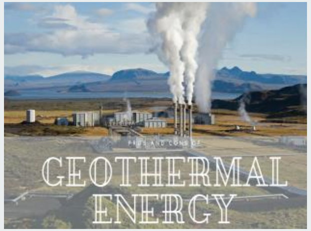
Waste heat, e.g., exhaust gas, jacket water

- **Waste-heat recovery** has been acknowledged as a promising solution to improving ICE thermal efficiency and reducing emissions

Background

- S-CO$_2$ cycle systems have appeared as an effective option for ICE WHR
  - good thermal match with heat source
  - system compactness
  - free from working fluid decomposition
  - Diverse wide-ranging applications

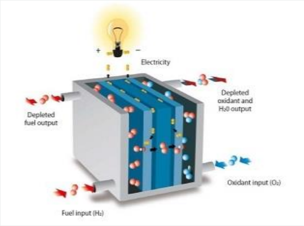
Coal power



Geothermal

CSP

Nuclear



Fuel cell

Background

- Extensive research on $S\text{-}CO_2$ cycle systems for ICE WHR is available
  - Thermodynamic and economic investigations
  - Design and optimisation with **a specific heat-source condition** (oversize/undersize)

- ICE will be operated under frequent part-load conditions, and the $S\text{-}CO_2$ cycle system will be forced to operate under off-design conditions

- $S\text{-}CO_2$ cycle system **off-design performance** needs to be carefully considered and the **design point** needs to be selected carefully

Aim of this paper

- Explore optimal design of S-CO_2 cycle systems for ICE WHR considering heat-source fluctuations and probability of occurrence of various part-load conditions:
  - all possible heat-source conditions selected for separate designs
  - detailed design and off-design performance performed
  - select optimal design scheme from thermodynamic and economic perspectives

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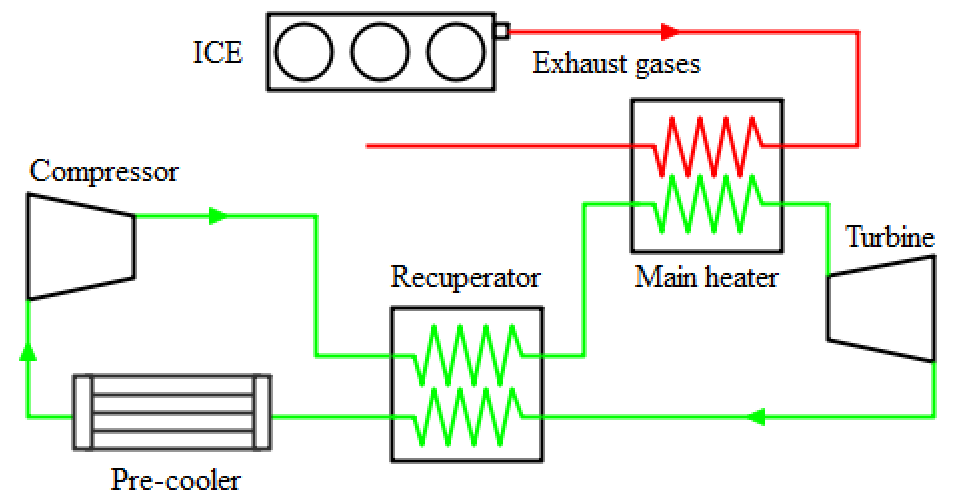
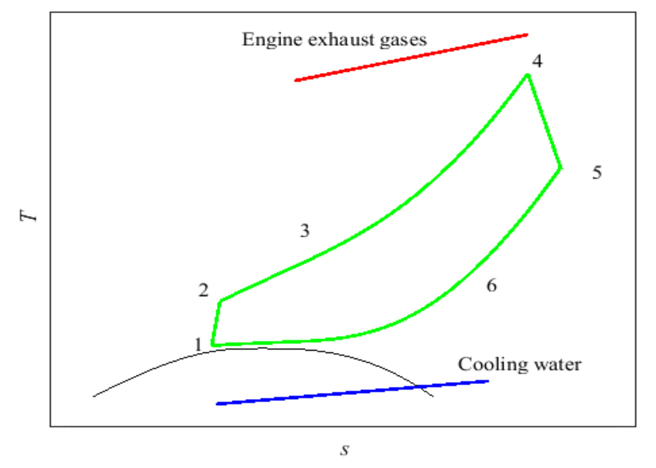
System configuration





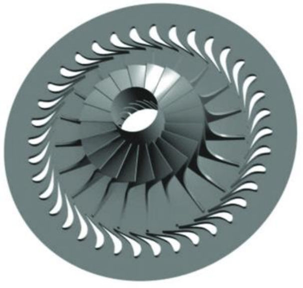
Component models

- Radial-inflow turbine model:
  - 1-D model based on mean-line method
  - loss models to capture losses relating to incidence, friction and leakage
  - all geometric and aerodynamic parameters optimised to achieve highest efficiency

$$\eta_{\mathrm{u}} = 2\bar{u}_1 \cdot \left(\varphi \cdot \sqrt{1-\Omega} \cdot \cos\alpha_1 - \bar{D}_2^2 \cdot \bar{u}_1 + \bar{D}_2 \cdot \psi \cdot \cos\beta_2 \cdot \left[\Omega + \varphi^2 \cdot (1-\Omega) - 2\varphi \cdot \sqrt{1-\Omega} \cdot \bar{u}_1 \cos\alpha_1 + \bar{D}_2^2 \cdot \bar{u}_1^2 \right]^{1/2} \right)$$

$$\zeta_{\mathrm{I}} = \frac{\left[w_1 \cdot \sin\left(\beta_1 - \beta_{1,\mathrm{opt}} \right) \right]^2}{2 \cdot \Delta h_{\mathrm{s}}} \qquad \zeta_{\mathrm{f}} = \frac{K_{\mathrm{f}} \cdot \frac{(\rho_1 + \rho_2)}{2} \cdot u_1^3 \cdot \left(\frac{D_1}{2} \right)^2}{2 \cdot m \cdot w_2^2 \cdot \Delta h_{\mathrm{s}}}$$

$$\zeta_{\mathrm{l}} = \frac{w_1^3 \cdot N_{\mathrm{rotor}}}{8\pi \cdot \Delta h_{\mathrm{s}}} \cdot \sqrt{\left[0.4 \cdot 0.004 \cdot K_{\mathrm{c1}} + 0.75 \cdot 0.00023 \cdot K_{\mathrm{c2}} - 0.3 \cdot \left(0.0004 \cdot 0.00023 \cdot K_{\mathrm{c1}} \cdot K_{\mathrm{c2}} \right) \right]}$$



[1] Glassman AJ. Computer program for design analysis of radial-inflow turbines. NASA Technical Note 1976.
[2] Da Lio L, Manente G, Lazzaretto A. A mean-line model to predict the design efficiency of radial inflow turbines in organic Rankine cycle (ORC) systems. Appl Energy 2017:205:187-209.

Component models

- Heat exchanger model:
  - shell and tube heat exchangers selected

Shell side (exhaust gases): $\alpha_{\mathrm{s}} = \alpha_{\mathrm{i}} \frac{c_{\mathrm{p,s}} G_{\mathrm{s}}}{\mathrm{Pr}_{\mathrm{s}}^{2/3}} \left(\frac{\mu_{\mathrm{s}}}{\mu_{\mathrm{w}}} \right)^{0.14} j_{\mathrm{c}} j_{\mathrm{l}} j_{\mathrm{b}} j_{\mathrm{s}} j_{\mathrm{r}}$

Tube side (CO_2 working fluid): $\alpha_{\mathrm{t}} = \frac{\lambda}{d_{\mathrm{i}}} \frac{(f/8)\,\mathrm{Re}\,\mathrm{Pr}}{\left[12.7(f/8)^{0.5}(\mathrm{Pr}^{2/3}-1)+1.07\right]} \cdot \left(\frac{\bar{c}_{\mathrm{p}}}{c_{\mathrm{p\,bulk}}}\right)^{0.35} \cdot \left(\frac{\lambda_{\mathrm{bulk}}}{\lambda_{\mathrm{wall}}}\right)^{-0.33} \cdot \left(\frac{\mu_{\mathrm{bulk}}}{\mu_{\mathrm{wall}}}\right)^{0.11}$

Overall Heat transfer coefficient: $\frac{1}{U} = \frac{1}{\alpha_{\mathrm{t}}} \cdot \frac{d_{\mathrm{o}}}{d_{\mathrm{i}}} + r_{\mathrm{ft}} \cdot \frac{d_{\mathrm{o}}}{d_{\mathrm{i}}} + \frac{\delta_{\mathrm{w}}}{\lambda_{\mathrm{w}}} \cdot \frac{d_{\mathrm{o}}}{d_{\mathrm{m}}} + r_{\mathrm{fs}} + \frac{1}{\alpha_{\mathrm{s}}}$

[1] K. Thulukkanam. Heat exchanger design handbook. Second Edition. CRC Press, 2013.

Cost models

- Module costing technique used to calculate bare module cost of each component, and chemical engineering plant cost index used to obtain system capital cost:

$$C_{\mathrm{BM}} = C_{\mathrm{p}}^{0} F_{\mathrm{BM}} = C_{\mathrm{p}}^{0}\left(B_1 + B_2 F_{\mathrm{M}} F_{\mathrm{P}}\right)$$

$$\log\left(C_{\mathrm{p}}^{0}\right) = K_1 + K_2 \log\left(X_i\right) + K_3\left[\log\left(X_i\right)\right]^2$$

$$\log\left(F_{\mathrm{p}}^{0}\right) = C_1 + C_2 \log\left(p_i\right) + C_3\left[\log\left(p_i\right)\right]^2$$

$$C = \sum_{\mathrm{i}} C_{\mathrm{BM}} \frac{CEPCI_{2017}}{CEPCI_{2001}}$$

- Levelised cost of electricity (*LCOE*) used for economic performance evaluations:

$$LCOE = \frac{C + \sum_{n=1}^{N} \frac{C_{\mathrm{O\&M}}}{(1+i)^n}}{\sum_{n=1}^{N} \frac{P}{(1+i)^n}}$$

Heat source conditions

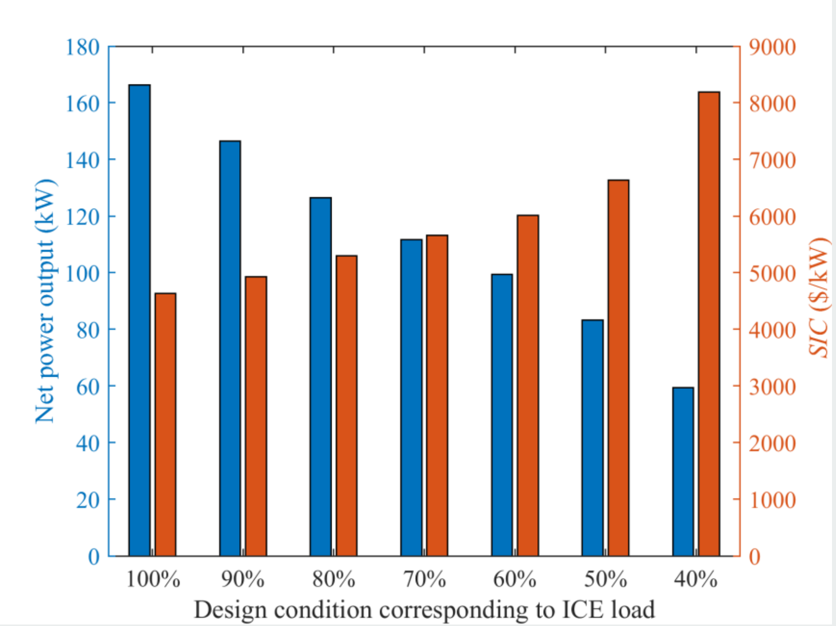
- ICE with rated power of 1 MW is selected; performance below:

Engine load	100%	90%	80%	70%	60%	50%	40%
Temperature (°C)	540	532	530	527	525	515	470
Mass flow rate (kg/s)	1.56	1.41	1.23	1.10	0.99	0.86	0.72

- Two cases considered to represent variations in the ICE operation/conditions:
  - same probability of occurrence for all possible conditions (equal-weighted scenario)
  - different weights for all conditions reported in Ref. [1], with 20.6%, 18.3%, 16.2%, 14.1%, 12.5%, 10.6% and 7.7% (different-weighted scenario), respectively

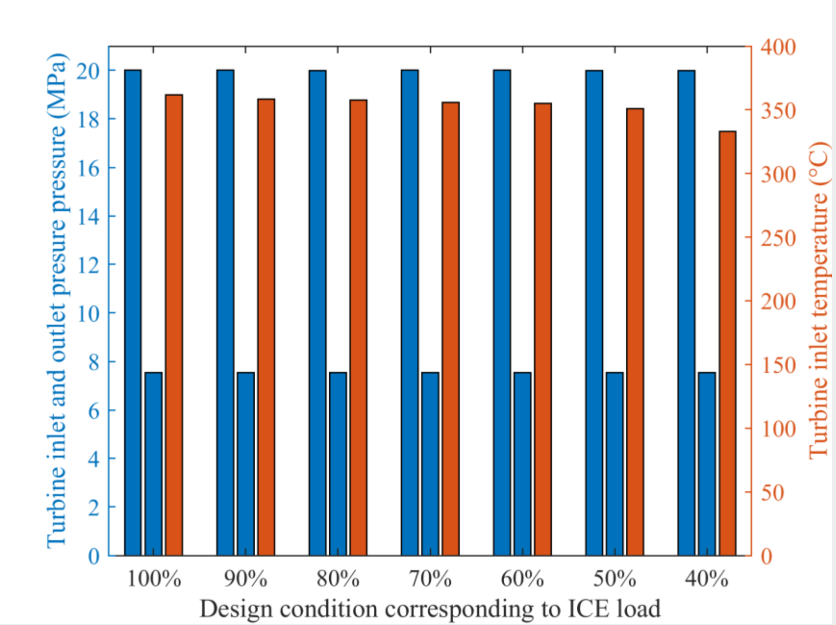
[1] Li X, Shu G, Tian H. Integrating off-design performance in designing CO2 power cycle systems for engine waste heat recovery. Energy Convers Manag 2019;201:112146.

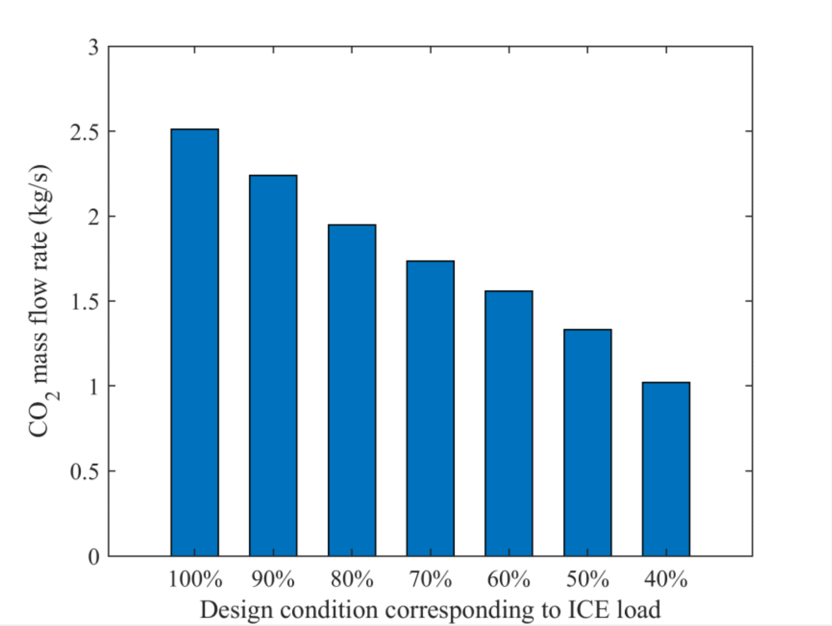
Separate designs



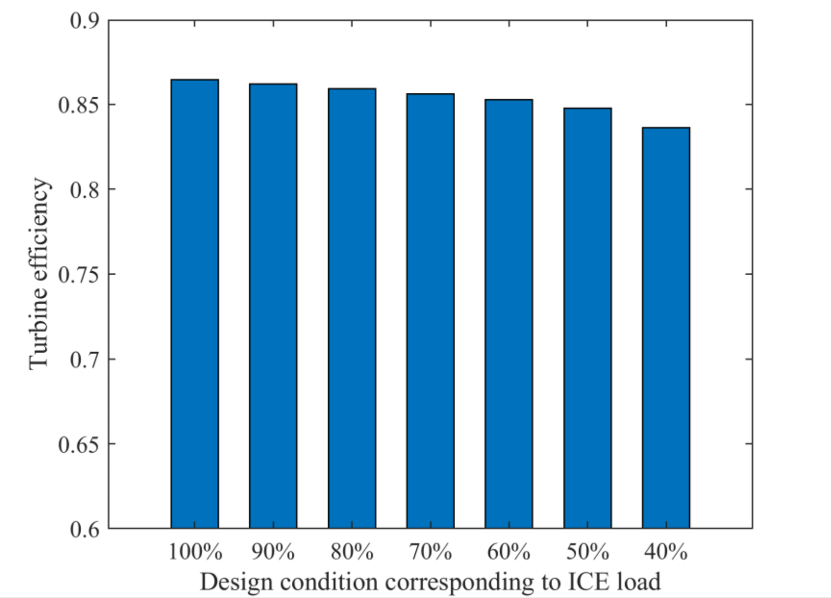
- Cycle and turbine design parameters are optimised simultaneously to achieve maximum power output
- Performance of optimal designs (from a thermodynamic perspective) are closely related to given heat-source conditions

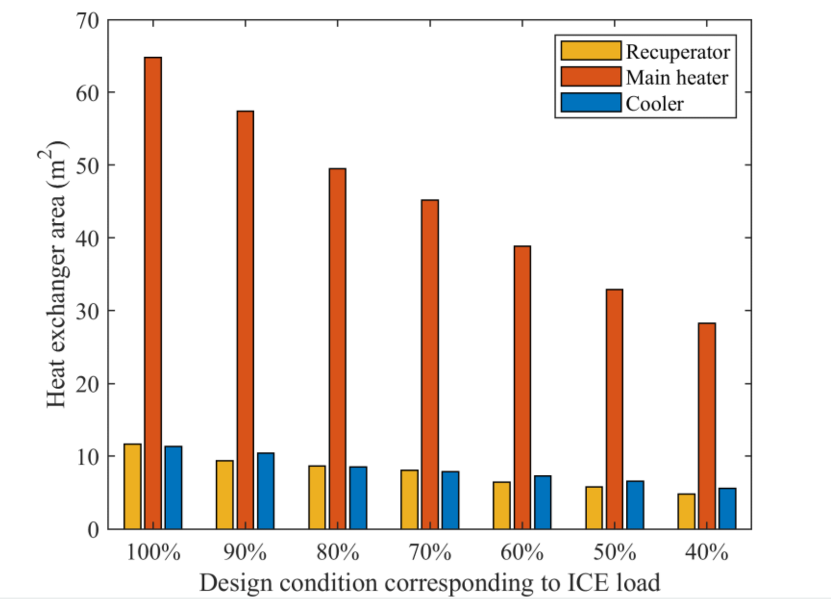
Separate designs





Separate designs





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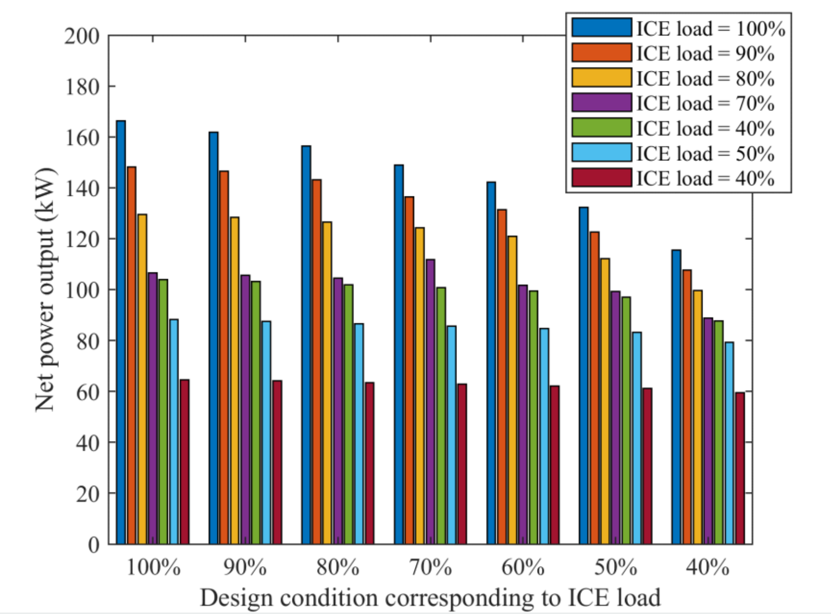
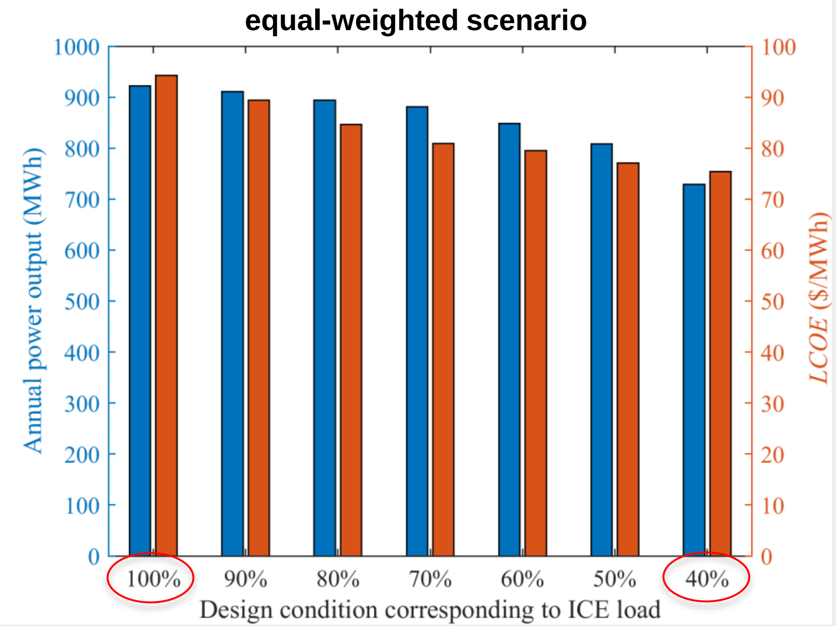
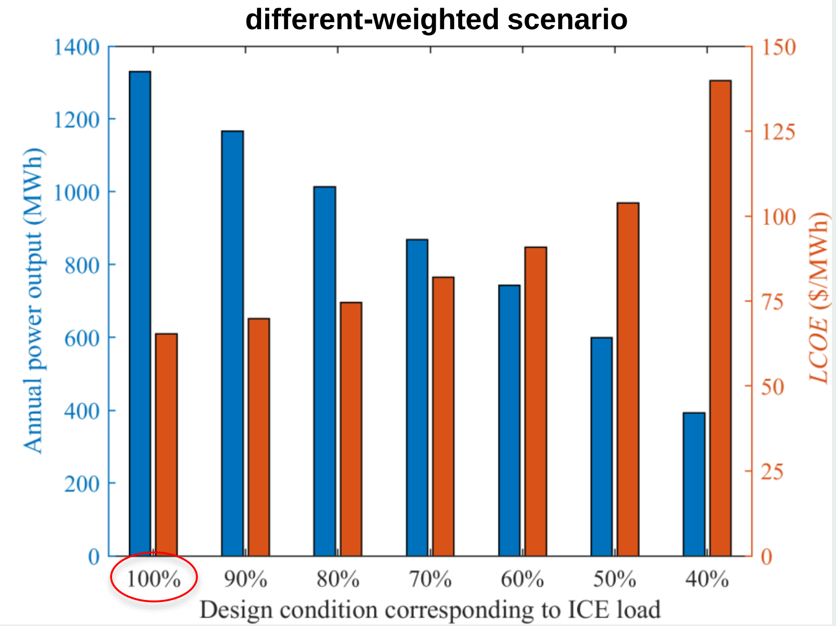
Off-design performance



- System operating parameters at off-design conditions are optimised to achieve the maximum power output
- Power output decreases for all designs when ICE load reduces as the heat input to S-CO_2 cycle system decreases
- Design scheme for rated condition (100% ICE load) provides higher net power output under most heat-source conditions

Off-design performance





Conclusions

- Design corresponding to ICE rated load (100%) provides maximum net power output of 166 kW and corresponding lowest *SIC* of 4630 $/kW
- Design scheme for ICE rated load provides a higher power output under most conditions
- For equal-weighted scenario:
  - design for rated load ICE condition is optimal from a thermodynamic perspective and maximum annual power output reaches 922 MWh
  - for 40% ICE load condition, lowest *LCOE* of 75 $/MWh
- For different-weighted scenario:
  - design for rated load condition is optimal with a maximum annual power output of 1330 MWh and lowest *LCOE* of 65 $/MWh

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Final remarks on future work

- Develop and integrate detailed compressor design and off-design models
- Consider more system configurations including utilisation of the jacket water heat
- Optimise design conditions (heat source temperature and mass flow rate)

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