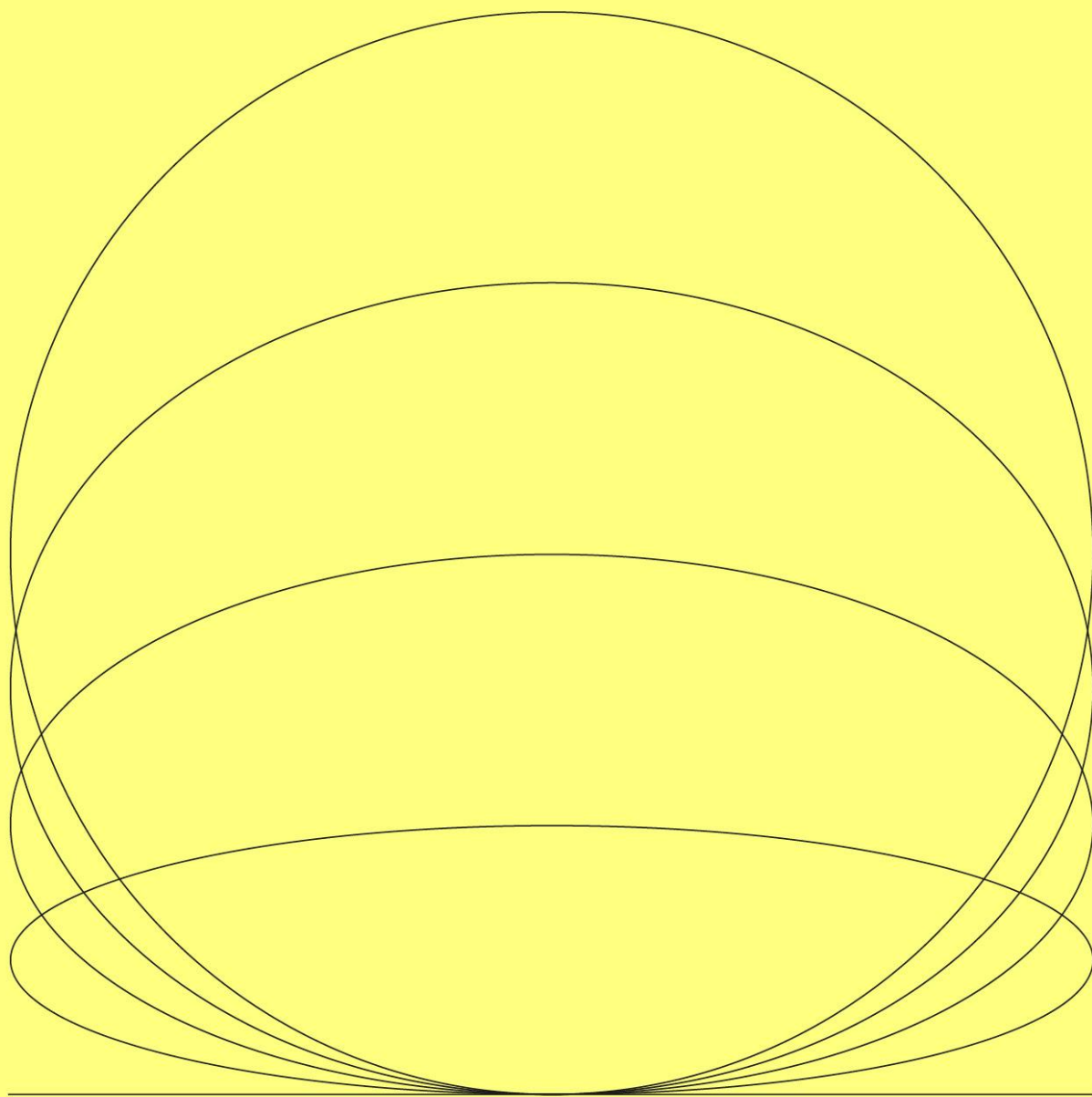




Report

H₂UpScale

Deliverable 3.4

Thermal System Design

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H2UpScale



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	Name	Beneficiary short name	Date (DD-MM-YYYY)
Lead Author(s)	Theo Koch	RWTH	14-09-2026
Contributing Author(s)			
Quality Assurance Reviews the quality of contents in the deliverable	Sören Tinz	FEV	10-09-2026
Approval Confirms that the deliverable conforms to the specific WP expectations/requirements	Federico Ribatti	TNO	15-09-2026

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Table of abbreviations

Abbreviation	Meaning
BOP	Balance of Plant
DC/DC	Direct Current to Direct Current (Converter)
HT	High Temperature (Coolant Loop)
MT	Medium Temperature (Coolant Loop)
NTU	Number of Transfer Units
SciML	Scientific Machine Learning (Julia Software Ecosystem)
UA	Overall Heat-Transfer Coefficient × Area (Thermal Conductance)
WP	Work Package



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1 Introduction

1.1 Context

WP2 established the application and fuel cell system requirements. D2.1 defined the application-specific requirements and supplied the marine operating profiles. D2.2 consolidated the fuel cell system requirements around a generic high power stack model and a modular arrangement based on 175 kW stack modules. In WP3, D3.1 reviewed the state of the art and defined architecture blueprints at 350 kW, 700 kW and 1050 kW. D3.2 provided should-cost functions for the balance-of-plant components.

D3.3 uses these inputs in a dynamic Julia/SciML model to optimise the fuel cell system for efficiency and balance-of-plant cost. The thermal system affects both objectives. Larger heat exchangers, radiators, pumps and fans increase cost. Pump and fan power reduce net system efficiency. Air-path choices also change the thermal load through compressor work and charge-air-cooler duty. The thermal system must therefore be optimised together with the air, hydrogen and electrical systems.

This deliverable, D3.4, presents the thermal part of that combined optimisation. It describes the cooling architecture, component equations and method used to scale the system to the higher power blueprints. Sections 6–9 report the thermal optimisation results.

1.2 Inputs used

The model and its generic component data were developed in cooperation with WP4, WP5, WP6 and WP7. WP4 supports system simulation and scalable stack and balance-of-plant models. WP5 supplies hydrogen-path component information, WP6 supplies air-path component information, and WP7 supports the coolant path and heat exchangers. The reduced-order model is fast enough for repeated full duty cycle simulations inside the optimiser.

The thermal-system model uses:

- the generic stack characteristics and modular power-node definition from D2.2;
- the application requirements and fixed architecture blueprints from D3.1;
- the balance-of-plant should-cost relations from D3.2;
- the road, marine and aviation duty cycles also used in D3.3;
- generic component parameters developed with WP4–WP7; and
- application-specific ambient conditions and operating limits.

The project does not select a specific commercial stack or cooling component. The results will compare blueprint and sizing choices. They will not represent qualification results for a named product.

1.3 Structure

Section 2 describes the thermal architecture and dynamic model. Section 3 gives the component equations, including the NTU–effectiveness heat-exchanger models and the efficiency-based pump and fan models. Section 4 explains how the thermal system scales to the higher-power nodes. Section 5 describes its

integration in the D3.3 optimiser. Sections 6–9 report the application results, and Section 10 concludes the deliverable.

2 Thermal system model and architecture

2.1 Model overview and boundary

The model contains two liquid-coolant temperature levels (Figure 1):

- a high-temperature (HT) loop for the fuel cell stack and gas-conditioning heat exchangers; and
- a medium-temperature (MT) loop for the fuel cell power electronics and electrically driven auxiliaries.

Both loops reject heat to ambient air through a shared front-end radiator pack. The MT radiator is upstream and receives ambient air. Its warmed outlet air then passes through the HT radiator. One aggregate pull-through fan moves air through both cores.

Air is the sink for every application, including the marine columns. The project agreed to keep liquid-to-air rejection throughout, so that one radiator-pack architecture serves all six columns rather than each application carrying its own. The marine HT margins reported in §7 and §8 are therefore set against a 298.15 K air sink; a seawater sink would relax them.

The thermal boundary includes stack heat, charge-air-cooler heat, hydrogen conditioning, DC/DC-converter losses, compressor-machine losses, fan-machine losses and coolant-pump power. The traction inverter, traction motor, cabin heating system and vehicle thermal loads are outside the current fuel cell-system model.

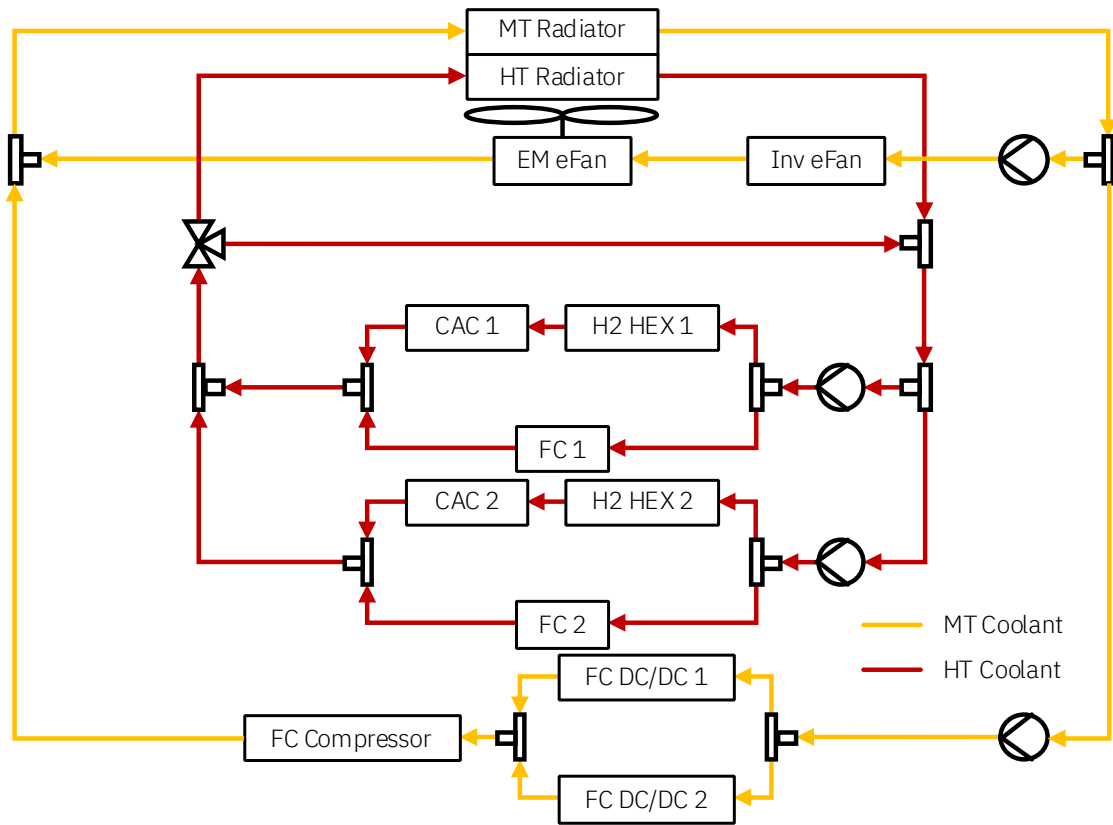


Figure 1. Modelled thermal system of the 350 kW fuel cell system.

2.2 High-temperature loop

The HT pump supplies parallel coolant branches. One branch serves each physical stack module. Parallel rather than series stack cooling supplies every module with coolant at the same inlet temperature, which gives better temperature uniformity and faster dynamic response (D3.1). An auxiliary branch serves each air and hydrogen train, with the hydrogen heat exchanger followed by the charge-air cooler on the coolant side. Placing the charge-air cooler and hydrogen heat exchanger in a branch parallel to the stack means the stack does not receive coolant already heated by these components. The branches merge before the shared HT radiator. One branch per stack module and per air and hydrogen train keeps the loop modular: the higher-power blueprints add branches on the same headers instead of requiring a new loop layout (§4.2). Separate module pumps, as in the reference schematic, additionally allow per-module flow balancing, staged pump operation at part load and continued operation at reduced power after a pump failure.

The stack supplies the dominant heat load. The hydrogen heat exchanger can add heat to or remove heat from the coolant, depending on the hydrogen inlet temperature. The charge-air cooler removes heat from compressed cathode air and adds it to the HT coolant. Pump shaft power is also returned to the loop as heat.

A radiator bypass controls over-cooling at low load. At higher load, the fan flow increases until the required heat rejection is reached or the fan rating is saturated.

2.3 Medium-temperature loop

The MT loop contains two hydraulically parallel branches:

- branch A cools the DC/DC converter and compressor electric machine; and
- branch B cools the radiator-fan motor and inverter.

Each branch has its own pump representation. The branch flows merge before the MT radiator. Converter losses, compressor-machine losses and fan-machine losses are added to the lumped MT coolant node. The aerodynamic work of the radiator fan remains in the exhaust airstream and is not added to the coolant.

The MT loop uses a radiator bypass to control coolant temperature without reducing the cold-plate flow below its useful range. This separates cold-plate cooling from radiator heat-rejection control.

2.4 Shared radiator air path

The radiators are arranged in series on the air side:

ambient air → MT radiator → HT radiator → pull-through fan → ambient

The same air mass flow passes through both cores. The HT radiator therefore sees the MT radiator outlet temperature rather than the ambient temperature. Air-side pressure drops from both cores, the grille, ducting, shroud and outlet are summed when fan power is calculated. For the road application, recovered ram pressure reduces the static pressure rise required from the fan.

This shared air path couples the coolant loops. A change in MT heat rejection changes the HT radiator inlet-air temperature, while a change in either core changes the pressure drop and fan power of the complete pack.

2.5 Lumped dynamic energy balances

Each coolant loop is represented by one well-mixed temperature state. For loop (j), the energy balance is

$$C_{th,j} \frac{dT_j}{dt} = \sum \dot{Q}_{source,j} + \dot{W}_{pump,j} - \dot{Q}_{radiator,j}.$$

where $C_{th,j} = m_j c_{p,j}$ is the lumped thermal capacitance. Heat sources and heat rejection are evaluated at every integration step of the complete fuel cell system model.

The HT balance contains stack heat and the net heat transferred from the hydrogen heat exchanger and charge-air cooler. The MT balance contains the electrical losses of the DC/DC converter, compressor machine and fan machine. Coolant-pump shaft power is dissipated within the closed liquid circuit and is added to the relevant loop balance.

The single-node representation captures mission-scale heat accumulation and cool-down. It does not resolve the temperature rise through an individual component, pipe transport delay or temperature differences between parallel branches. Consequently, the temperature limits are checked against the lumped loop

temperature, which excludes the coolant temperature rise across the stack. The local stack outlet temperature is higher, so the reported temperature margins are screening values. More detailed modelling, resolving stack inlet and outlet temperatures with separate coolant nodes, is left for future work.

2.6 Thermal control

The thermal controller uses algebraic, load-following commands:

- HT and MT pump flow increase with heat load and remain between minimum and design flow;
- the shared fan flow is selected to meet HT heat-rejection demand, subject to its rated flow;
- the HT bypass limits over-cooling when minimum fan flow already rejects more heat than required; and
- the MT bypass controls the lower-temperature loop because the MT radiator has no independent air-flow actuator.

These controls do not add solver states. A real pump or fan speed controller could later replace the commanded-flow relations without changing the loop energy balances.

2.7 Main assumptions

Table 1 summarises the main assumptions of the thermal system model.

Table 1. Main assumptions made.

Assumption	Treatment in the model
Stack temperature	Isothermal stack; aggregate heat generation is calculated, but local temperature is not resolved
Coolant loops	One well-mixed temperature state per loop
Coolant properties	Constant density and specific heat within each application model
Branch distribution	Equal flow division between identical parallel components
Pipes and headers	Adiabatic and without separate thermal mass
Component pressure drop	Reference pressure drop scaled by a mass-flow power law
Radiator pack	One functional MT core, one functional HT core and one aggregate fan pack per system
Heat exchangers	Steady NTU–effectiveness relation evaluated at each dynamic time step



Assumption

Treatment in the model

Pumps and fan

Commanded flow; no shaft-speed/flow operating map

Faults and degradation

Not represented

3 Component modelling

3.1 NTU-effectiveness method

The radiators, charge-air cooler and hydrogen heat exchanger use the NTU-effectiveness method. The heat-capacity rates are

$$C_h = \dot{m}_h c_{p,h}, \quad C_c = \dot{m}_c c_{p,c}.$$

$$C_{\min} = \min(C_h, C_c), \quad C_{\max} = \max(C_h, C_c), \quad C_r = \frac{C_{\min}}{C_{\max}}.$$

The number of transfer units and heat flow are

$$NTU = \frac{UA}{C_{\min}}.$$

$$\dot{Q} = \varepsilon C_{\min} (T_{h,in} - T_{c,in}).$$

where effectiveness ε depends on the flow arrangement. Outlet temperatures follow from the energy balance of each stream.

For the charge-air cooler and hydrogen heat exchanger, conductance varies with both stream flows:

$$UA = UA_{\text{ref}} \left(\frac{\dot{m}_1}{\dot{m}_{1,\text{ref}}} \right)^a \left(\frac{\dot{m}_2}{\dot{m}_{2,\text{ref}}} \right)^b.$$

In the radiator model, air-side and coolant-side conductances are scaled separately and combined as thermal resistances in series. This matches the two convection resistances of the radiator core.

3.2 Radiators

The MT and HT radiators use a cross-flow, both-fluids-unmixed correlation:

$$\varepsilon = 1 - \exp \left[\frac{NTU^{0.22}}{C_r} (\exp(-C_r NTU^{0.78}) - 1) \right].$$

For C_r close to zero, the limiting relation $\varepsilon = 1 - \exp(-NTU)$ is used. Effectiveness is restricted to zero to one. Radiator heat flow is cooling-only. If the coolant is colder than the air, radiator heat rejection is set to zero.

Radiator rating sets the reference (UA). Conductance is related to core depth using a fixed frontal area, free-flow ratio, fin-surface density and air/coolant heat-transfer resistances. A larger rating therefore increases

both conductance and depth. Its air-side pressure drop also rises, so greater radiator conductance comes with a fan-power penalty.

Coolant-side and air-side pressure drops follow

$$\Delta p = \Delta p_{\text{ref}} \left(\frac{\dot{m}}{\dot{m}_{\text{ref}}} \right)^n.$$

The air-side value is corrected for temperature because the HT core receives warmer, less-dense air from the MT core.

3.3 Charge air cooler

The charge air cooler transfers heat from compressed cathode air to the HT coolant. It uses the same cross-flow effectiveness relation as the radiators. Instantaneous gas and coolant capacity rates determine its effectiveness and heat flow.

The model only cools the cathode air. When the inlet gas is colder than the coolant, heat transfer is set to zero. Gas-side flow uses the common compressible-restriction model from the air path. Coolant-side pressure drop follows the reference-flow power law.

3.4 Hydrogen heat exchanger

The hydrogen heat exchanger uses a counter-flow relation:

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C_r)]}{1 - C_r \exp[-NTU(1 - C_r)]}.$$

For C_r close to one, the limiting relation $\varepsilon = NTU/(1 + NTU)$ is used.

Unlike the radiators and charge-air cooler, this model is bidirectional. Hydrogen colder than the HT coolant is heated and removes heat from the coolant. Hydrogen hotter than the coolant adds heat to the loop. This formulation can represent the larger hydrogen-conditioning duty expected for the aviation blueprint once its boundary conditions are confirmed.

3.5 Fuel cell stack heat

The isothermal stack model calculates the aggregate heat transferred to the HT coolant from the difference between thermoneutral and operating voltage:

$$\dot{Q}_{\text{stack}} = n_{\text{cell}} I (V_{\text{th}} - V_{\text{cell}}).$$

where n_{cell} is the represented cell count, I is stack current, V_{th} is thermoneutral voltage and V_{cell} is modelled cell voltage. Operating points that reduce cell voltage increase heat generation at the same current. Stack-coolant pressure drop is calculated separately from coolant flow.

3.6 Coolant pumps

Pump hydraulic and shaft power are

$$\dot{W}_{\text{hyd}} = \Delta p \frac{\dot{m}}{\rho}, \quad \dot{W}_{\text{shaft}} = \frac{\dot{W}_{\text{hyd}}}{\eta_{\text{pump}}}.$$

The reduced-order model uses a constant hydraulic efficiency. The electric motor and inverter use either a constant combined efficiency or an attached scaled efficiency map. Electrical bus input is the shaft power divided by the machine efficiency.

Coolant flow is commanded rather than solved from a pump curve. The pump must overcome the radiator pressure drop and the largest pressure drop among parallel branches. Pressure drops from parallel branches are therefore not added together.

3.7 Radiator fan

The radiator fan must overcome both radiator cores and the installation:

$$\Delta p_{fan} = \max(\Delta p_{MT} + \Delta p_{HT} + \Delta p_{installation} - \Delta p_{ram}, 0).$$

Fan shaft power and bus input are

$$\dot{W}_{fan,shaft} = \frac{\dot{V}_{air}\Delta p_{fan}}{\eta_{fan}}, \quad \dot{W}_{fan,bus} = \frac{\dot{W}_{fan,shaft}}{\eta_{motor+inverter}}.$$

The fan aerodynamic efficiency and motor/inverter efficiency are given by a scaled generic electric-machine map. Air density is corrected using fan inlet temperature. The electrical loss is added to the MT coolant loop, while aerodynamic work leaves with the exhaust air.

3.8 Electrical loss sources on the MT loop

The DC/DC converter uses a constant conversion efficiency:

$$\dot{Q}_{DC/DC} = P_{in} - P_{out} = P_{in}(1 - \eta_{DC/DC}).$$

The compressor electric machine and inverter use a scaled efficiency map. Where a turbine is present, turbine shaft power first offsets compressor demand. The remaining machine loss is added to the MT loop.

These relations keep the electrical and thermal balances consistent. Every liquid-cooled electrical loss becomes a heat source in the appropriate loop.

3.9 Model parameterisation

The relations of §2 and §3 are stated with symbols rather than numbers. The values behind them are collected here so that a reader can reproduce a heat balance or check a margin without reading the source. Where a quantity is scaled rather than fixed, Table 2 gives the value for one 350 kW group and §4.3 gives the scaling rule.

Table 2. Numerical parameters of the thermal model.

Group	Parameter	Value	Unit
Radiator pack	Shared frontal area	1.3	m ²
Radiator pack	Minimum free-flow ratio	0.78	-
Radiator pack	Air-side surface density	1,200	m ² /m ³



Group	Parameter	Value	Unit
Radiator pack	Colburn factor at Re ~ 1,365	0.0115	
Radiator pack	Fanning friction factor at Re ~ 1,365	0.044	
Radiator pack	Overall fin efficiency	0.94	-
Radiator pack	Coolant-side UA per air-side area	555.6	W/(K m ²)
Radiator pack	Reference pack air flow	9.85	kg/s
Radiator pack	Air-side share of thermal resistance	0.807	-
Radiator pack	Rating-to-UA reference temperature difference	25.533	K
Radiator pack	Maximum core depth, HT / MT	0.120 / 0.060	m
Radiator pack	Installation pressure drop at reference air flow	150	Pa
Radiator pack	Ram-pressure recovery, road column only	0.35	-
Radiator pack	Reference air state for density correction	298.15 K, 101,325 Pa	
Radiator fan	Bus draw buying the reference air flow	11.77	kW
Radiator fan	Aerodynamic efficiency	0.45	-
Radiator fan	Motor and inverter efficiency	0.94	-
Radiator fan	Mean core air temperature, upstream / downstream	302.74 / 324.54	K
Radiator fan	Fan inlet air temperature reference	341.73	K
Coolant pumps	Hydraulic efficiency, both loops	0.65	-
Coolant pumps	HT pump rating reference	2.5	kW
Coolant pumps	MT pump rating reference, per branch	0.15	kW
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Group	Parameter	Value	Unit
Coolant pumps	MT loop design temperature rise	7.0	K
Pressure drops	Stack coolant channel, reference	15	kPa
Pressure drops	HT radiator coolant channel, reference	15	kPa
Pressure drops	HT loop plumbing, reference	40	kPa
Pressure drops	MT loop plumbing, reference	20	kPa
Pressure drops	MT branch-A cold plates, reference	50	kPa
Pressure drops	MT branch-B cold plates, reference	30	kPa
Pressure drops	Coolant-side mass-flow exponent	1.75	-
Loop states	HT loop volume per 350 kW group	16.5	L
Loop states	MT loop volume per 350 kW group	8.0	L
Loop states	HT reference temperature	353.15	K
Loop states	MT reference temperature	330.0	K
Loop states	HT lumped capacitance at reference, per group	59.6	kJ/K
Loop states	MT lumped capacitance at reference, per group	28.5	kJ/K
Coolant	Ethylene-glycol mass fraction, both loops	0.52	-
Coolant	Density over 263.15 to 358.15 K	1,085.1 to 1,036.2	kg/m ³
Coolant	Specific heat over 263.15 to 358.15 K	3,062 to 3,492	J/(kg K)
Coolant	Dynamic viscosity over 263.15 to 358.15 K	12.96 to 0.92	mPa s
Coolant	Thermal conductivity over 263.15 to 358.15 K	0.346 to 0.397	W/(m K)
Limits	HT coolant temperature limit	358.15	K
Limits	MT coolant temperature limit	343.15	K
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Group	Parameter	Value	Unit
Limits	Ambient sink, road and marine columns	298.15 K, 101.3 kPa	

The two lumped capacitances are derived rather than set: each is the loop volume multiplied by coolant density and specific heat at that loop's reference temperature. Radiator reference conductance is not tabulated because it is not a constant; it follows from the searched rating through the rating-to-UA convention above, and the implied core depth follows from the pack geometry (§3.2). Parameters of the stack, air path and hydrogen path are defined in D3.3 rather than here.

4 Scaling to higher power nodes

4.1 Scaling principle

The 350 kW road system is the source architecture. Higher-power systems repeat stack modules, electrical groups, air and hydrogen paths according to the fixed D3.1 blueprint counts. Repeated components are placed behind shared coolant headers and shared heat-rejection equipment.

Scaling is based on the heat and flow generated by the complete higher-power system. A system repeated at identical temperatures, flow rates and ambient condition would require radiator duty in proportion to power. The 700 kW and 1050 kW ratings nevertheless differ from simple multiples of the 350 kW ratings, because each column has its own blueprint train counts, mission and part-load operating points, and the optimiser sizes each column separately.

4.2 Thermal count rules

The following rules are used for every blueprint:

- each physical stack module has one HT coolant source branch;
- each air and hydrogen path has one HT auxiliary branch containing its hydrogen heat exchanger and charge-air cooler;
- the number of DC/DC cold plates follows the DC/DC-converter count;
- the number of compressor-machine cold plates follows the air-train count;
- equal flow division is assumed within a bank of identical parallel branches;
- all HT branches merge before one functional HT radiator;
- all MT branches merge before one functional MT radiator;
- each complete system has one aggregate shared fan pack.

Source counts divide flow and distribute heat. They do not multiply the number of functional radiators or fan packs.

4.3 Component scaling

The stack heat of repeated identical modules is summed. Charge-air-cooler and hydrogen-heat-exchanger duties are calculated for each train and then summed at the loop level. Electrical losses are calculated from the actual power handled by each converter or machine.

Reference (UA), pressure drop, thermal mass and flow authority scale with the purchased component rating. Radiator rating also changes implied core depth and air-side pressure drop. Pump and fan ratings therefore interact with radiator sizing rather than acting as independent multipliers.

The dynamic model retains one HT and one MT temperature state at every power node. This symmetry reduction is suitable for aggregate sizing when parallel branches operate alike. Explicit branch states would be required to study maldistribution, mismatch, isolation or faults.



4.4 Blueprint application

Table 3 lists the stack modules, electrical groups, air and hydrogen trains and shared thermal equipment of each blueprint.

Table 3. Blueprint overview.

Application	Stack modules	Electrical groups / DC/DC	Air and hydrogen trains	Shared thermal equipment
350 kW road	2	1	1	1 HT radiator, 1 MT radiator, 1 fan pack
700 kW marine, modular blueprint	4	2	2	1 HT radiator, 1 MT radiator, 1 fan pack
700 kW marine, common-rail blueprint	4	2	1	1 HT radiator, 1 MT radiator, 1 fan pack
1050 kW marine, modular blueprint	6	3	3	1 HT radiator, 1 MT radiator, 1 fan pack
1050 kW marine, common-rail blueprint	6	2	2	1 HT radiator, 1 MT radiator, 1 fan pack
1050 kW aviation	6	3	3	1 HT radiator, 1 MT radiator, 1 fan pack

The modular and common-rail marine blueprints can produce different thermal loads at the same system power. They use different train counts, component ratings and part-load operating points. The optimiser evaluates each blueprint separately.

The aviation blueprint uses the same count structure as the modular 1050 kW marine blueprint, but its boundary conditions differ. Altitude changes air density, compressor operation, radiator capacity and fan power. Cryogenic hydrogen also changes the hydrogen heat-exchanger duty. The aviation thermal system must be re-evaluated with its own mission and ambient profile; it cannot be inferred from the marine result.



5 Integration in the D3.3 optimiser

5.1 Combined fuel cell and thermal optimisation

D3.3 and D3.4 use one combined optimisation (Figure 2). The D3.3 optimiser proposes a candidate design, builds the corresponding fuel cell and thermal system model, runs the complete duty cycle and returns efficiency, cost and constraint outcomes.

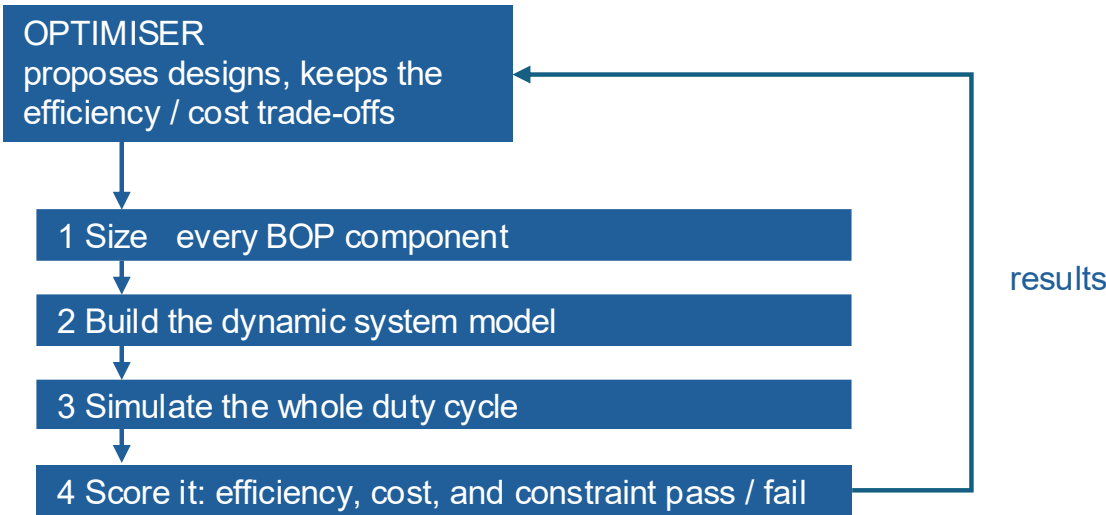


Figure 2. Optimisation framework overview.

Five entries in the D3.3 design vector directly size the thermal system:

Variable	Meaning	Effect in the model
pump_HT_kW	HT coolant pump rating	HT coolant flow authority and pump cost
pump_MT_kW	MT coolant pump rating, per branch	MT branch flow authority and pump cost
fan_kW	Radiator fan pack rating	Maximum shared air flow, parasitic power and cost
radHT_kW	HT radiator rating	HT (UA), core depth, pressure drop, thermal mass and cost
radMT_kW	MT radiator rating	MT (UA), core depth, pressure drop, thermal mass and cost

Two further D3.3 variables strongly affect the thermal result:

- cathode stoichiometry changes cathode mass flow, compressor power and charge air cooler duty;

- cathode backpressure changes compressor work, stack voltage and stack heat.

The charge air cooler and hydrogen heat exchanger ratings are derived from the simulated duties rather than searched independently.

5.2 Candidate evaluation

Each candidate passes through the following sequence:

1. The optimiser supplies all energetic and thermal design variables.
2. Component maps, ratings, (UA) values, flow limits and controller limits are set for that candidate.
3. The coupled model is integrated over the full application duty cycle.
4. The simulation records heat duties, coolant temperatures, radiator rejection, pump and fan power, component utilisation and temperature margin.
5. The candidate is checked against electrical, thermal and numerical limits.

Cycle efficiency and D3.2 balance-of-plant should cost are returned to the multi-objective optimiser.

5.3 Objectives and thermal constraints

The two objectives are those defined in D3.3:

- maximise cycle efficiency, calculated from net electrical energy and fresh hydrogen lower-heating-value energy;
- minimise the D3.2 balance-of-plant should-cost.

The thermal system contributes through component cost and parasitic power. It is also subject to hard admissibility checks:

- maximum HT coolant temperature;
- maximum MT coolant temperature;
- pump and fan demand within installed ratings;
- radiator and heat-exchanger operation within modelled capacity;
- physically valid temperatures, flows and effectiveness values; and
- successful completion of the dynamic solve.

The temperature thresholds are engineering screening limits. Passing them shows that a candidate is admissible within this comparative model. It does not constitute component qualification.

5.4 Thermal outputs

The report roles are the low-cost, compromise and high-efficiency designs selected in D3.3 (§3.8) from the admissible Pareto set of each column. For each confirmed report-role design, chapters 6 to 9 report:

- peak HT and MT heat source and peak radiator heat rejection;
- maximum HT and MT coolant temperature;
- temperature margin to the assumed limits;
- radiator, charge-air-cooler and hydrogen-heat-exchanger ratings;
- pump and fan ratings and peak power;
- component utilisation;
- thermal-system should-cost and its share of balance-of-plant should-cost.

The same output set is used for every blueprint so that power-node and architecture comparisons remain consistent.

Three further quantities need the time-resolved solution rather than its extrema, and the retained confirmation artefacts do not carry it: the HT and MT heat-source ledgers split by contributor, cycle-integrated radiator rejection together with pump and fan cycle energy, and minimum and time-dependent coolant temperatures. Emitting the trace from the confirmation stage would supply all three from one re-run, and that run batches with the MT radiator bound re-run of §10.

6 350 kW road platform

The 350 kW road column solves the 4,503 s road mission at a 34.7 % mean load. One HT loop, one MT loop and one shared radiator pack serve both stack modules, so the differences between the confirmed designs (Table 4) come from the five thermal sizing variables rather than from architecture.

Table 4. Thermal results for the confirmed 350 kW road designs.

Quantity	Low-cost	Compromise	High-efficiency	Unit
Peak HT heat source	348.5	347.2	341.3	kW
Peak HT radiator rejection	350.1	348.8	342.4	kW
Peak MT heat source	10.6	10.6	10.5	kW
Peak MT radiator rejection	10.3	10.2	10.2	kW
Peak charge-air-cooler duty	20.9	20.9	21.0	kW
Peak H ₂ heat-exchanger duty	4.02	4.08	4.22	kW



Quantity	Low-cost	Compromise	High-efficiency	Unit
Maximum HT coolant temperature	355.40	355.51	355.63	K
HT margin to 358.15 K	2.75	2.64	2.52	K
Maximum MT coolant temperature	328.93	328.87	328.80	K
MT margin to 343.15 K	14.22	14.28	14.35	K
HT radiator rating	365.70	396.73	470.15	kW
HT radiator utilisation	0.953	0.875	0.726	-
MT radiator rating	35.23	34.58	40.52	kW
MT radiator utilisation	0.302	0.306	0.259	-
Fan rating	20.58	17.56	16.72	kW
Fan utilisation	0.988	0.991	0.865	-
HT pump rating / utilisation	0.55 / 0.634	0.63 / 0.621	0.52 / 0.605	kW
MT pump rating, both branches / utilisation	0.05 / 0.318	0.06 / 0.311	0.05 / 0.316	kW
Thermal should-cost	1,620	1,630	1,689	€
Thermal share of BOP should-cost	43.9	44.0	44.7	%

Utilisation is the mission peak divided by the installed rating. Radiator ratings are UA-equivalents at a 25.5 K reference approach, so a rating states conductance rather than heat-rejection capacity at mission conditions. Thermal should-cost covers both radiators, both coolant pumps, the fan pack, the charge-air cooler and the hydrogen heat exchanger.

The HT loop binds and the MT loop does not. HT margins run 2.52–2.75 K against the 358.15 K limit while MT margins run 14.22–14.35 K against 343.15 K, about five times more room. The MT radiator carries a 10.5 kW peak duty on a core rated 34.6–40.5 kW and never exceeds 31 % of that rating. The lower bound on radMT_kW sets that figure rather than the search. A 34 kW bound corresponds to an 8 mm core at the shared 1.3 m² frontal area, about as thin as a louvered-fin core is built, and the compromise design sits 1.7 % above it. A 10.5 kW duty against a 34 kW floor caps utilisation near 31 % whatever the optimiser does.

The three roles trade radiator core against fan work. The low-cost design buys a 365.7 kW HT core, runs it at 95.3 %, and pays for the shortfall with a 20.6 kW fan held at 98.8 % of its rating. The high-efficiency design buys a 470.2 kW core, runs it at 72.6 %, and needs only a 16.7 kW fan at 86.5 %. A deeper core also raises air-side pressure drop (§3.2), so this is a genuine trade rather than a free gain; on this mission the added conductance wins.

Two of the three roles leave under 1.5 % of fan rating unused at the mission peak. A fan sized that closely carries no reserve for a fouled core or an ambient temperature above the modelled sink, and it would be the only thermal component on this column that the mission drives to its limit. The coolant pumps are the opposite case: the HT pump runs at 60–63 % of rating and the MT pump at about 31 %.

Peak HT heat source falls from 372.6 kW on the flagship baseline to between 341.3 and 348.5 kW on the optimised roles, because raising cell efficiency results in reduced stack heat ingress into the coolant. Thermal hardware is 43.9–44.7 % of balance-of-plant should-cost here, the highest share of any column in this deliverable.

7 700 kW maritime platforms

The 700 kW marine column runs both D3.1 blueprints on the same 7,080 s supplied mission at a 57 % mean load. The modular blueprint carries two air trains, one per electrical group; the common-rail blueprint feeds both groups from one larger train. Both share a single HT radiator, a single MT radiator and one fan pack, so the comparison tests whether consolidating the air path moves the thermal result (Table 5).

Table 5. Thermal results for the confirmed 700 kW marine designs.

Quantity	Modular low-cost	Modular compromise	Common-rail low-cost	Common-rail compromise	Unit
Peak HT heat source	790.9	714.1	757.6	734.6	kW
Peak HT radiator rejection	792.4	715.5	758.6	735.7	kW
Peak MT heat source	22.4	21.5	22.2	21.9	kW
Peak MT radiator rejection	21.6	21.2	21.6	21.5	kW
Peak charge-air-cooler duty	33.1	42.7	41.4	41.3	kW
Peak H ₂ heat-exchanger duty	16.08	15.97	10.06	10.61	kW
Maximum HT coolant temperature	355.45	355.40	355.41	355.48	K
HT margin to 358.15 K	2.70	2.75	2.74	2.67	K
Maximum MT coolant temperature	329.86	329.45	329.64	329.48	K



Quantity	Modular low-cost	Modular compromise	Common-rail low-cost	Common-rail compromise	Unit
MT margin to 343.15 K	13.29	13.70	13.51	13.67	K
HT radiator rating	808.54	844.92	796.10	872.86	kW
HT radiator utilisation	0.978	0.845	0.952	0.842	
MT radiator rating	72.27	82.97	76.87	72.26	kW
MT radiator utilisation	0.309	0.259	0.289	0.303	
Fan rating	51.47	88.14	67.87	68.83	kW
Fan utilisation	0.999	0.393	0.750	0.581	
HT pump rating / utilisation	1.90 / 0.811	/ 1.48 / 0.657	1.10 / 0.763	1.10 / 0.711	kW
MT pump rating, both branches / utilisation	0.63 / 0.148	/ 0.65 / 0.147	0.23 / 0.167	0.22 / 0.168	kW
Thermal should-cost	2,741	2,864	2,614	2,683	€
Thermal share of BOP should-cost	37.5	38.5	38.1	38.4	%

Consolidating the air path does not move the thermal bottleneck. At the compromise role the two blueprints differ by 0.08 K in peak HT temperature and by 0.03 K in peak MT temperature, against HT margins of 2.75 K and 2.67 K. The HT loop binds on all four designs and the MT loop holds 13.29–13.70 K, the same five-to-one split the 350 kW column shows.

The common rail is the cheaper thermal system, by €181 at the compromise role, but it does not reject less heat. Its peak HT source is 20.5 kW higher than the modular blueprint's at that role while its cycle efficiency is 0.52 points better. Peak rejection and cycle efficiency are set at different points of the mission and do not order the same way.

The hydrogen heat exchanger separates the two blueprints more than any other thermal component: its peak duty is 15.97 kW on the modular blueprint against 10.61 kW on the common rail. Both are small against an HT source of roughly 720 kW, so the difference barely moves the loop balance.

Fan sizing again varies more than anything else. The modular low-cost design holds its 51.5 kW fan at 99.9 % of rating at the mission peak, while its own compromise role buys a 88.1 kW fan and uses 39.3 % of it. The optimiser is not converging on a fan size; it is buying whatever the chosen radiator core leaves it to move, which is what §4.3 predicts when pump and fan ratings interact with radiator sizing rather than scaling independently.



8 1050 kW maritime platforms

The 1050 kW marine column is the continuous-rating case of this deliverable. Its 2,560 s mission holds 89 % of nameplate almost throughout, so the cooling pack has little low-load recovery to fall back on. Both D3.1 blueprints are reported in Table 6: a modular blueprint with three air trains and three electrical groups, and a common rail with two of each.

Table 6. Thermal results for the confirmed 1050 kW marine designs.

Quantity	Modular low-cost	Modular compromise	Common-rail low-cost	Common-rail compromise	Unit
Peak HT heat source	1,105.4	1,058.4	1,105.4	1,039.1	kW
Peak HT radiator rejection	1,103.7	1,050.6	1,099.7	1,030.4	kW
Peak MT heat source	33.6	33.1	33.0	33.2	kW
Peak MT radiator rejection	33.6	33.1	33.0	33.2	kW
Peak charge-air-cooler duty	77.4	84.6	71.8	78.7	kW
Peak H ₂ heat-exchanger duty	16.39	16.50	14.89	14.00	kW
Maximum HT coolant temperature	355.17	355.35	355.15	355.33	K
HT margin to 358.15 K	2.98	2.80	3.00	2.82	K
Maximum MT coolant temperature	330.22	329.57	330.12	329.62	K
MT margin to 343.15 K	12.93	13.58	13.03	13.53	K
HT radiator rating	1,199.96	1,415.23	1,199.08	1,385.71	kW
HT radiator utilisation	0.921	0.748	0.922	0.750	
MT radiator rating	144.65	102.99	119.44	118.18	kW



Quantity	Modular low-cost	Modular compromise	Common-rail low-cost	Common-rail compromise	Unit
MT radiator utilisation	0.233	0.321	0.277	0.281	
Fan rating	78.31	145.48	92.00	102.45	kW
Fan utilisation	0.796	0.298	0.713	0.412	
HT pump rating / utilisation	2.44 / 0.693	2.47 / 0.630	1.85 / 0.703	2.43 / 0.605	kW
MT pump rating, both branches / utilisation	0.73 / 0.161	0.37 / 0.176	0.44 / 0.163	0.47 / 0.171	kW
Thermal should-cost	3,680	3,863	3,436	3,731	€
Thermal share of BOP should-cost	33.0	33.2	34.3	35.1	%

The two blueprints reach nearly the same thermal answer. At the compromise role their HT margins differ by 0.02 K and their HT radiators by 2.1 %; at the low-cost role both converge on a 1,199 kW HT core at 92 % utilisation and the same 1,105.4 kW peak HT source. The modular blueprint spreads that duty over three air trains and the common rail over two, and the shared cooling pack absorbs the difference.

This column carries the widest HT margins in the deliverable, 2.80–3.00 K, and the narrowest marine MT margins, 12.93–13.58 K. The HT margin is bought rather than given. The HT cores here are the largest relative to duty of any column, running at 75–92 % against 84–98 % at 700 kW, and the optimiser spends the money because the mission gives it no low-load stretch in which a smaller core could recover.

A third role was confirmed on the modular blueprint. Its high-efficiency design buys the largest HT radiator in the deliverable at 1,481 kW, runs it at 70.5 %, and returns an HT margin of 2.67 K for €4.0 k of thermal hardware. That is €133 more than the compromise role for 0.13 K less HT margin, because the efficiency it buys comes from cathode stoichiometry rather than from cooling.

The MT radiator is again the least used component, at 23–32 % of rating on a 33 kW duty, and again the bound rather than the duty sets the number. radMT_kW is a system-level variable, so its 34 kW lower bound scales to 102 kW at this node while the radiator frontal area stays at 1.3 m². The bound therefore buys core depth, 8 mm at 350 kW against 23.9 mm here, and the compromise design sits 1.0 % above it at 24.1 mm. The duty needs roughly 1,050 W/K, near 6 mm of core, so the unscaled 350 kW bound would already cover the 1050 kW MT loop. Surplus MT core is charged twice on this architecture: once when it is purchased, and again when its depth obstructs the shared frontal area ahead of the binding HT core.

9 1050 kW aviation platform

The aviation column shares the count structure of the modular 1050 kW marine blueprint but not its boundary conditions. Its 3,360 s sortie climbs to 7,620 m, where ambient pressure falls to about 37 % of the sea-level value and ambient temperature falls to 238.7 K. Thin, cold air changes radiator capacity and fan power in opposite directions, and cryogenic hydrogen adds a commanded heat sink to the MT loop. The two admissible report roles are reported in Table 7; the compromise design is inadmissible on the air-handling machine torque envelope, an electrical gate rather than a thermal one, and all three pass both temperature limits.

Table 7. Thermal results for the confirmed 1050 kW aviation designs.

Quantity	Low-cost	High-efficiency	Unit
Peak HT heat source	966.3	963.0	kW
Peak HT radiator rejection	959.5	955.4	kW
Peak MT heat source	30.8	31.0	kW
Peak MT radiator rejection	24.1	24.4	kW
Peak charge-air-cooler duty	55.3	56.1	kW
Peak H ₂ heat-exchanger duty	10.62	11.31	kW
Maximum HT coolant temperature	356.28	356.23	K
HT margin to 358.15 K	1.87	1.92	K
Maximum MT coolant temperature	327.01	327.11	K
MT margin to 343.15 K	16.14	16.04	K
HT radiator rating	1,082.73	1,111.16	kW
HT radiator utilisation	0.893	0.867	
MT radiator rating	108.60	108.95	kW
MT radiator utilisation	0.283	0.284	
Fan rating	47.05	61.72	kW
Fan utilisation	0.949	0.690	
HT pump rating / utilisation	1.71 / 0.534	1.66 / 0.531	kW



Quantity	Low-cost	High-efficiency	Unit
MT pump rating, both branches / utilisation	0.31 / 0.153	0.72 / 0.144	kW
Thermal should-cost	3,317	3,410	€
Thermal share of BOP should-cost	31.3	30.3	%

This column holds both extremes of the deliverable. Its HT margins, 1.87 and 1.92 K, are the smallest anywhere in D3.3 and D3.4; its MT margins, 16.14 and 16.04 K, are the largest. The HT result is the harder one to buy back: peak HT source is 966 kW, below the 1,058 kW of the modular marine blueprint at the same power node, yet the aviation HT loop still runs about 1 K hotter.

The MT asymmetry has a specific cause. Peak MT rejection is 24.1 kW against a 30.8 kW MT source, so roughly 6.7 kW of MT heat never reaches the radiator. The cryogenic fuel vaporiser is modelled as a commanded sink on that loop and absorbs the difference. Every other column in this deliverable rejects essentially all of its MT source through the core, and the aviation MT margin should be read as a consequence of carrying liquid hydrogen rather than as spare cooling capacity.

Fan utilisation reaches 94.9 % on the low-cost role. Together with the 350 kW road roles and the modular 700 kW low-cost design, that makes four confirmed designs in this deliverable whose fan has under 6 % of its rating left at the mission peak. The high-efficiency role buys a 61.7 kW fan instead and uses 69.0 % of it, for €93 more thermal hardware and 0.05 K more HT margin.

Thermal hardware is 30.3–31.3 % of balance-of-plant should-cost here, the lowest share of any column. The trend across the deliverable is monotonic with power node, from 43.9–44.7 % at 350 kW to 37.5–38.8 % at 700 kW and 30.3–35.1 % at 1050 kW, because one radiator pack, one fan pack and two pumps serve a growing number of stack modules while the air and hydrogen trains multiply with the count rules of §4.2.

10 Conclusion

This deliverable defines the thermal-system model used in the H₂UpScale system optimisation. The architecture uses separate HT and MT liquid-coolant loops with one shared, serial airflow radiator pack. Heat exchangers are represented with NTU effectiveness relations. Pumps, the radiator fan and electrical loss sources use reduced-order pressure-drop and efficiency models. Two lumped coolant temperature states retain the mission-scale thermal dynamics needed for component sizing.

The scaling method repeats stack, air-path, hydrogen-path and electrical components according to the D3.1 blueprint counts while retaining one shared HT radiator, one shared MT radiator and one aggregate fan pack per system. Component ratings are determined from simulated aggregate heat and flow, not from a fixed multiple of the 350 kW design.

The thermal model is integrated directly in the D3.3 multi-objective optimiser. This captures the two-way coupling between thermal sizing, system efficiency and balance-of-plant cost. It also enforces coolant-temperature and component capacity limits during the search.

The high-temperature loop binds on every column and the medium-temperature loop does not. Across the confirmed report roles, HT margins to the 358.15 K limit run 1.87...3.00 K while MT margins to 343.15 K run 12.93...16.14 K. The MT radiator runs between 20 % and 32 % of rating on every column, but that is the

imposed lower sizing bound showing through rather than an optimisation result. radMT_kW scales with node power while the radiator frontal area does not, so its 34 kW bound at 350 kW becomes 102 kW at 1050 kW and buys core depth from 8 mm to 23.9 mm. Duty over bound is 31...32 % at every node, which is the ceiling the search can reach, and several report roles sit within 2 % of it. The MT loop of this architecture carries excess capacity; re-running the columns with a substantially lower MT radiator bound is the first item of further thermal work.

Radiator core and fan work trade against one another, and the trade is real in both directions. A deeper core raises conductance and air-side pressure drop together (§3.2), yet on every mission here the added conductance wins: the 350 kW high-efficiency role buys a 470 kW HT core, runs it at 73 % and needs a 16.7 kW fan, while its low-cost role buys 366 kW, runs it at 95 % and holds a 20.6 kW fan at 98.8 % of rating. Four confirmed designs across the deliverable leave under 6 % of fan rating unused at their mission peak, which is the tightest recurring sizing result in this work.

Thermal hardware is a falling share of balance-of-plant should-cost as the power node grows: 43.9...44.7 % at 350 kW, 37.5...38.8 % at 700 kW and 30.3...35.1 % at 1050 kW. One radiator pack, one fan pack and two coolant pumps serve a growing number of stack modules while the air and hydrogen trains multiply with the count rules of §4.2.

Consolidating the air path changes cost and duty but not the thermal bottleneck. At 700 kW the two blueprints differ by 0.08 K in peak HT temperature at the compromise role, and at 1050 kW by 0.02 K, with HT radiator ratings within 2.1 %. Whether a marine system runs one large air train or several smaller ones is not a thermal decision at this level of the model.

The aviation column is the exception on both loops. It holds the smallest HT margins in the deliverable, 1.87 K and 1.92 K on its two admissible roles, and the largest MT margins, 16.04 K and 16.14 K. The MT figure should not be read as spare capacity: the cryogenic fuel vaporiser is a commanded sink on that loop and removes about 6.7 kW of a 30.8 kW MT source before it reaches the radiator.

Two limits qualify these results. The confirmation artefacts retain cycle extrema and peak duties rather than the time-resolved solution, so the heat-source ledgers, cycle integrals and temperature histories of §5.4 are deferred; emitting the trace from the confirmation stage would supply them, and batches with the MT radiator bound re-run into a single pass. The hydrogen heat exchanger reports utilisation between 3.2 and 4.8 because its rating is a flow-throughput proxy rather than a heat-transfer capacity; as in D3.3 it is a diagnostic here and not a capacity gate.

