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Modeling a Condensing Heat Exchanger in RELAP5-3D

This model has been completely defeatured and is for academic purposes only

Condensing Heat Exchangers in Thermal-Hydraulic Systems

Why this matters?

- Condensing heat exchangers remove steam enthalpy through a coupled sequence of vapor cooling, latent heat release, condensate drainage, and tube-side liquid heating.
- They are used in passive heat-rejection systems, steam-handling trains, industrial energy-recovery systems, and separate-effects facilities that must reproduce plant-like condensation behavior.
- The governing engineering questions are operating-point closure, outlet state, pressure control, drainage stability, and startup conditioning time.

Across these applications, the fundamental problem is to direct just enough steam to the exchanger to satisfy the coolant-side heat demand while preserving a stable shell pressure and drain path.

Passive steam rejection

Condense steam to remove stored energy while maintaining pressure control.

Feedwater conditioning

Reject latent heat to a colder water stream that must reach a target outlet condition.

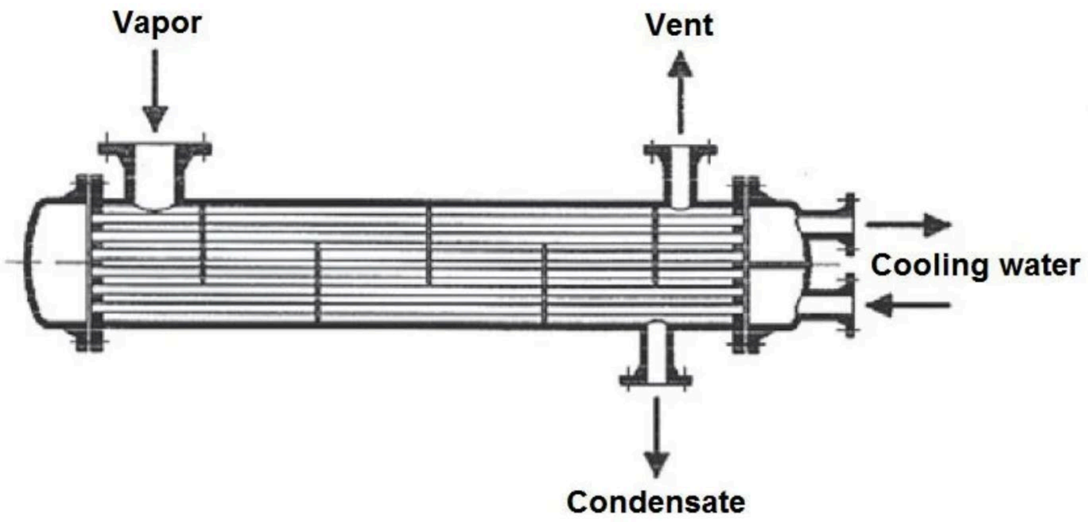
Vent-steam recovery

Drainage and condensation determine how much vapor is retained, recovered, or discharged.

How a Condensing Shell-and-Tube Heat Exchanger Works

Physics Sequence

- Steam enters the shell at high pressure and is driven toward local saturation at the tube wall.
- A condensate film forms on the outside of the tubes; latent heat release dominates the duty through most of the shell.
- Continuous drainage prevents shell flooding and preserves vapor access to the bundle.
- On the tube side, feedwater gains sensible heat; tube metal thermal inertia controls startup lag and low-load settling time.



How is the operating point set?

- Operating point equations:

$$\dot{Q} = \dot{m}_{fw}\,(h_{out} - h_{in})$$

$$\dot{Q} = \dot{m}_s\left(h_{s,in} - h_{s,out}\right)$$

$$\text{Find } \dot{m}_s \text{such that } T_{out,tube} \rightarrow T_{target}$$

- Shell inlet pressure and temperature are set by the operating mode, while shell outlet pressure is set by the downstream drain-side basis.

- Because tube-side demand is fixed, the shell-side steam flow can be tuned monotonically to hit the requested tube outlet temperature.

Two-Fluid Field Equations Solved by RELAP5-3D

Primary Variables

- RELAP5-3D advances eight primary variables: pressure P, phasic internal energies U_g and U_f, void fraction α_g, phasic velocities v_g and v_f, and noncondensable quality X_n
- A condensing exchanger requires separate vapor and liquid balances because phase change, slip, gravity drainage, and sensible cooling do not evolve on a single homogeneous time scale.
- The model is nonhomogeneous and nonequilibrium: the phases share pressure but carry separate mass, momentum, and energy inventories.

$$\frac{\partial(\alpha_k\rho_k)}{\partial t} + \left(\frac{1}{A}\right)\frac{\partial(\alpha_k\rho_k v_k A)}{\partial x} = \Gamma_k\,, \qquad k = g, f$$

$$\alpha_k\rho_k A\frac{\partial v_k}{\partial t} = -\alpha_k A\frac{\partial P}{\partial x} + \alpha_k\rho_k A g_x + F_{w,k} + F_{i,k} + S_{m,k}$$

$$\frac{\partial(\alpha_k\rho_k U_k)}{\partial t} + \frac{\left(\frac{1}{A}\right)\partial(\alpha_k\rho_k U_k v_k A)}{\partial x} + P\frac{\partial\alpha_k}{\partial t} = Q_{w,k} + Q_{i,k} + S_{e,k}$$

$$\Gamma_f = -\Gamma_g\,, \qquad \Gamma_g = \Gamma_{ig} + \Gamma_w\,(+\Gamma_c) \qquad Q = Q_{wg} + Q_{wf}$$

Closure Layers

- Mass closure is obtained through the phase-change rate Γ_g, which is tied to the interface energy partition.
- Momentum closure adds interphase drag, wall drag, form losses, and body-force effects that cannot be resolved directly in a 1-D field equation set.
- Energy closure adds bulk interfacial transfer, near-wall boiling/condensation transfer, and wall heat partitioning between phases.
- This layered structure is why RELAP can represent condensation dynamics without assuming thermal equilibrium or zero slip.

Interfacial Heat and Mass Transfer Closures

Interfacial closure structure

- RELAP separates bulk interfacial transfer from wall-driven interfacial transfer because near-wall boiling or condensation is not governed only by $T_g - T_f$.

$$\Gamma_g = \Gamma_{ig} + \Gamma_w\ (+\Gamma_c)$$

$$Q_{ig} + Q_{if} + \Gamma_{ig}(h_g^* - h_f^*) + \Gamma_w(h_g' - h_f') = Q_{ifB} = H_{if}\left(T_s - T_f\right)$$

$$Q_{igB} = H_{ig}\,(T_s - T_g)$$

Closure layer	Role in the exchanger
Bulk interface	Cools vapor and liquid toward saturation / thermal equilibrium
Near-wall interface	Creates condensate directly at the tube surface
Wall partition	Splits wall heat flux between vapor and liquid energy equations

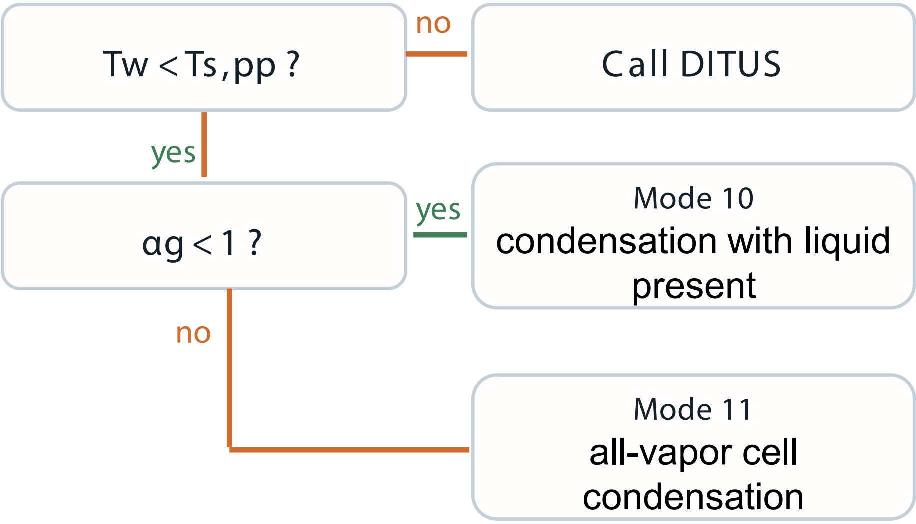
Interfacial closure structure

- The shell-side solution must capture both bulk vapor cooling and near-wall condensation on the tube bundle.
- Condensate generation immediately feeds the shell-side liquid inventory and, through the drain legs, the outlet mass flow and shell outlet quality.
- Because the exchanger outlet is fully condensed in the final cases, the interfacial closure mostly governs how rapidly the shell collapses toward the liquid-dominated outlet state during the transient.
- For a condensing exchanger, these closures determine shell vapor depletion, condensate rate, and the rate at which the tube-side target can be supported.

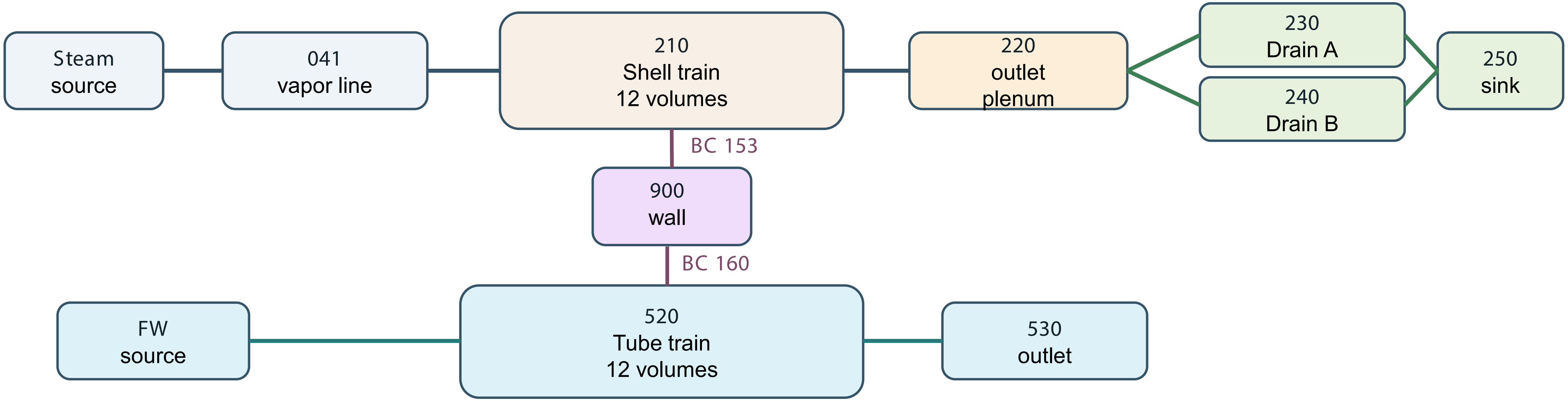
Wall Heat Transfer, Condensation Mode Logic, and Tube Correlations

Wall-to-Fluid Heat Flux Decomposition

- During condensation, the relevant reference is the saturation temperature based on vapor partial pressure rather than total pressure.
- RELAP prints heat-transfer mode numbers in the major edits; modes 10 and 11 correspond to condensation logic, with an additional 20 added if noncondensables are present.
- That partial-pressure dependence is physically important because even small noncondensable fractions can change the condensation driving temperature.

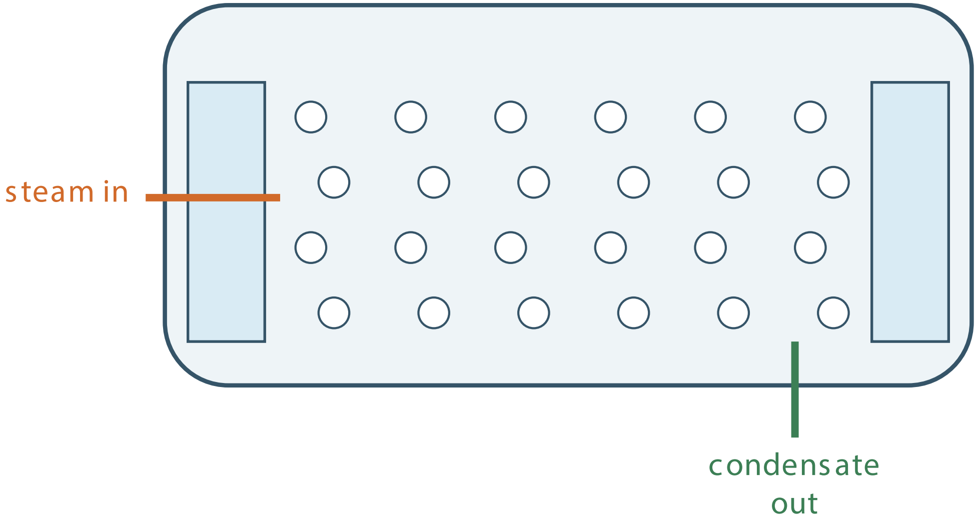


RELAP5-3D Nodalization for the Local HX Model



- The model resolves the local steam inlet line, 12 shell cells, outlet plenum, twin drain legs, 12 tube cells, and a dedicated wall heat structure.
- The wider plant steam split is represented by the tuned 041 inlet flow rather than by a full upstream branch model.

Geometry Basis and Component Representation



Parameter	Value
Shell ID	9 in
Shell length	200 in
Tube count	34 Tubes
Area	180 ft^2

This is a local mechanistic model. The retained detail is the minimum set required to predict the requested shell-side operating conditions with physically meaningful transients.

Component Mapping in RELAP

- Steam inlet and drain lines are represented by 1D pipes and junctions, preserving length, hydraulic diameter, roughness, and local form-loss effects.
- The shell body is homogenized into a one-dimensional shell train with the correct overall inventory, length scale, and exchanger heat-transfer area.
- The tube side is represented by a one-dimensional equivalent path whose total heat-transfer area and hydraulic diameter match the documented exchanger design basis.
- The tube wall is represented by a heat structure so that conjugate storage and delayed startup behavior are captured rather than collapsed into a pure algebraic UA.

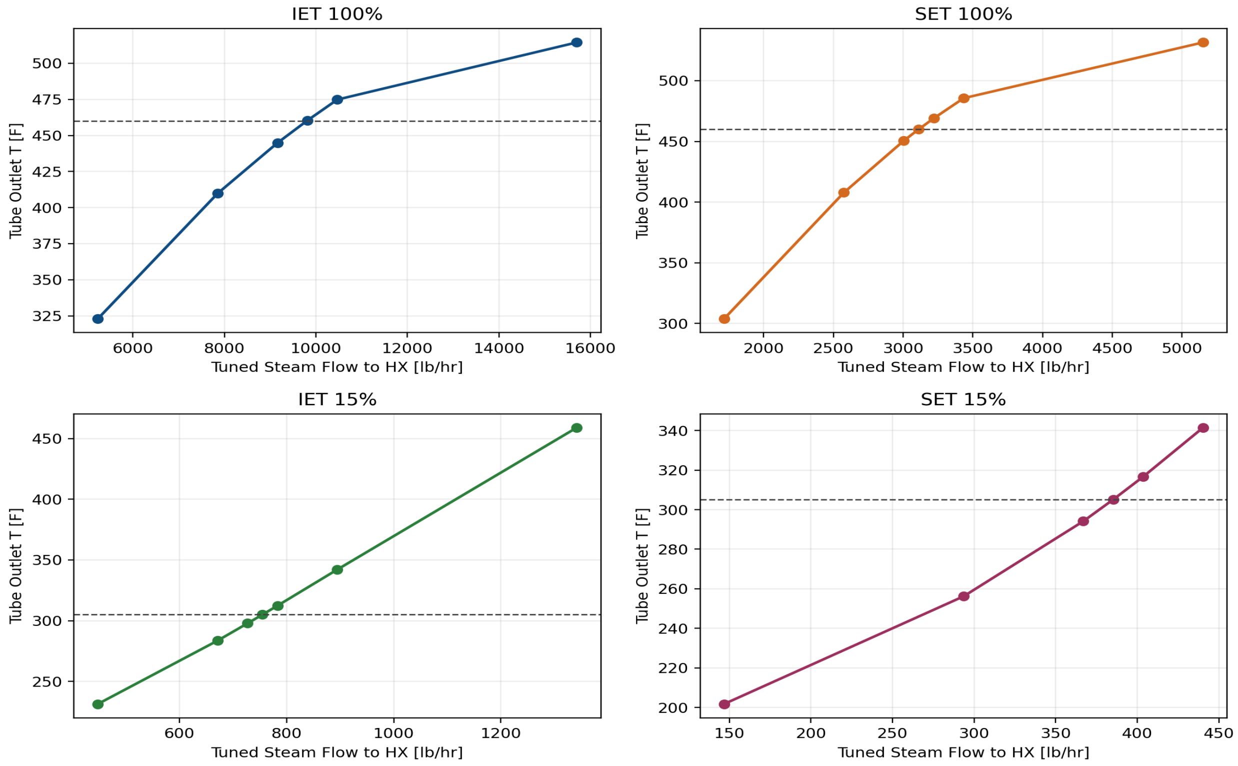
Boundary Conditions by Operating Mode

Mode	Source P [psia]	Source T [F]	FW flow [lb/s]	Tube target [F]	Tuned steam [lb/hr]	Run horizon [s]
IET 100%	900	596	5.0	460	10,000	300
SET 100%	900	596	1.7	460	3,000	300
IET 15%	1000	570	0.6	305	750	600
SET 15%	1000	570	0.2	305	390	1,200

- The feedwater flow is imposed directly for each case, so the shell-side steam flow is the only tuned variable.
- Shell outlet pressure is fixed by the local downstream drain-side basis, which is why shell inlet and outlet pressure remain close while outlet temperature changes substantially.
- Longer low-load horizons are intentional: they allow the wall and shell inventory to finish conditioning before the final output window is evaluated.

Steam-Flow Search Used to Match the Tube-Side Target

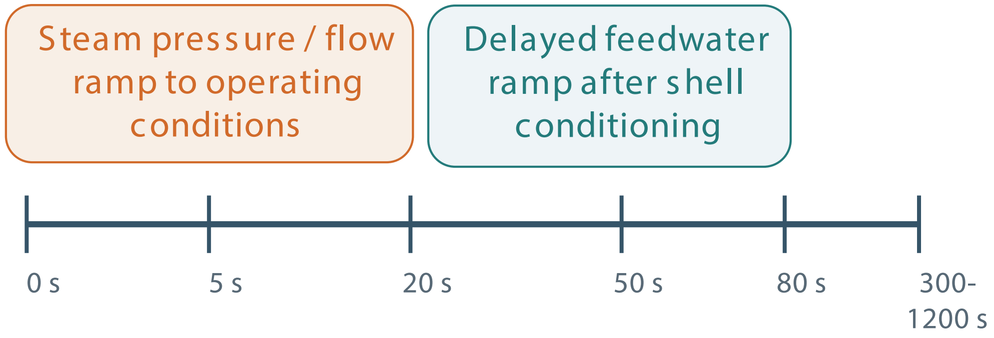
Monotonic Convergence of the Search



- Each case admits a monotonic operating-point search because feedwater flow is fixed.
- High-load cases require substantially more steam to reach the 460 F target than low-load cases need to reach 305 F.
- Search termination required both target satisfaction and a non-oscillatory tail window.

Initialization Strategy and Core Modeling Hypotheses

Staged Startup

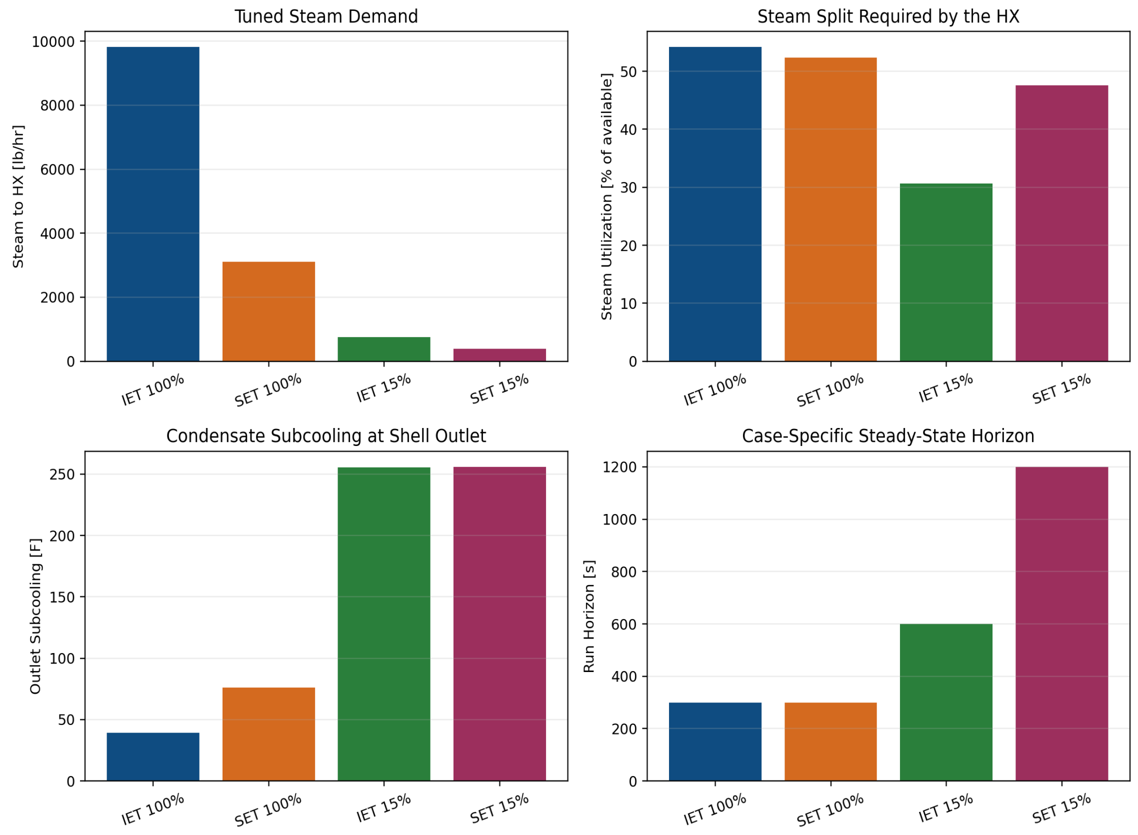


The steam-first startup was introduced specifically to eliminate artificial startup shocks and the oscillatory low-flow behavior seen in earlier surrogate formulations.

Principal Hypotheses

- The current model boundary is local to the exchanger; upstream valve motion is represented implicitly through the tuned steam inlet flow.
- Shell-side multidimensional crossflow and baffle leakage are homogenized into an equivalent one-dimensional shell path with the documented total area and inventory.
- Noncondensable-gas accumulation is excluded from the base model so that the acceptance outputs reflect pure steam-water condensation behavior.
- Steady reporting requires both target temperature closure and a flat tail window in the shell and tube temperature histories.

Cross-Case Regime Comparison



- High-load modes use the largest fraction of available steam and settle most quickly.
- The two 15% modes are deeply subcooled at the shell outlet because condensation completes and the liquid continues to cool against the feedwater demand.
- The run horizon itself becomes a physical indicator: long low-load settling reflects wall and inventory thermal storage, not unresolved oscillation.

Physical Interpretation Across Load Levels

100% Load Family

- At 100% load, larger steam demand and stronger tube-side heating keep the outlet condensate relatively warm and shorten the settling time.
- The exchanger is still fully condensing at the outlet, but the condensate does not cool nearly as far below saturation as in the low-load modes.

15% Load Family

- At 15% load, the exchanger still condenses all incoming steam but needs much longer to thermally condition the wall and inventory.
- Once condensation is complete, the outlet liquid continues to cool strongly, producing the largest outlet subcooling values in the study.

Outputs for the Four Required Operating Modes

Mode	Pin [psia]	$\dot{m}$shell [lb/s]	Pout [psia]	Tout [F]	xout [-]	Steam to HX [lb/hr]
IET 100%	900	2.75	898	495	0.000	10,000
SET 100%	900	0.90	899	455	0.000	2,000
IET 15%	1000	0.20	999	290	0.000	750
SET 15%	1000	0.10	999	290	0.000	385

- Outlet quality is effectively zero in all four cases, indicating complete condensation by the shell outlet location.
- The local shell train exhibits only small pressure drop, so the dominant mode-to-mode differences are in required steam flow and outlet condensate subcooling.

Model Credibility and Remaining Limitations

Captured Well

- Shell-side condensation duty and outlet thermodynamic state.
- Tube-side heating to the target outlet condition at fixed feedwater flow.
- Wall thermal inertia and long low-load conditioning transients.

Reduced-Order Assumptions

- Upstream valve motion and plant-level branch splitting are represented implicitly through the tuned inlet flow.
- Shell-side multidimensional crossflow and baffle leakage are homogenized into a one-dimensional equivalent shell path.
- Noncondensable-gas accumulation is outside the current base-model scope.

Best Next Validation Targets

- Compare outlet temperature and condensate rate against test data once available.
- Benchmark shell pressure drop and drainage using a higher-fidelity shell-side model or empirical exchanger correlation set.
- If valve sizing becomes primary, introduce explicit upstream valve characteristics into the coupled model.

Main Scientific Conclusions

- All four production cases reached stable, non-oscillatory operating points with complete condensation by the shell outlet and a smooth tail-window approach to the final solution.
- The exchanger behaves as a strong thermal sink with only modest local shell-side pressure drop, so the dominant differences among operating modes are in required steam flow and outlet subcooling.
- High-load cases converge faster and retain warmer outlet condensate, whereas low-load cases require longer wall-conditioning time and leave the shell outlet deeply subcooled.

Modeling Recommendations and Next Analytical Steps

- Use the present deck as the baseline for shell-side operating-condition prediction.
- If future work shifts toward valve sizing or system controls, connect the local exchanger model to explicit upstream valve characteristics and branch hydraulics rather than forcing those dynamics into the present operating-point deck.
- Validate shell outlet temperature, condensate removal rate, and startup conditioning time against test data before increasing shell-side geometric detail.
- Only add higher-order shell-side detail, noncondensable transport, or broader system coupling when the next engineering decision actually requires those phenomena.

Idaho National Laboratory

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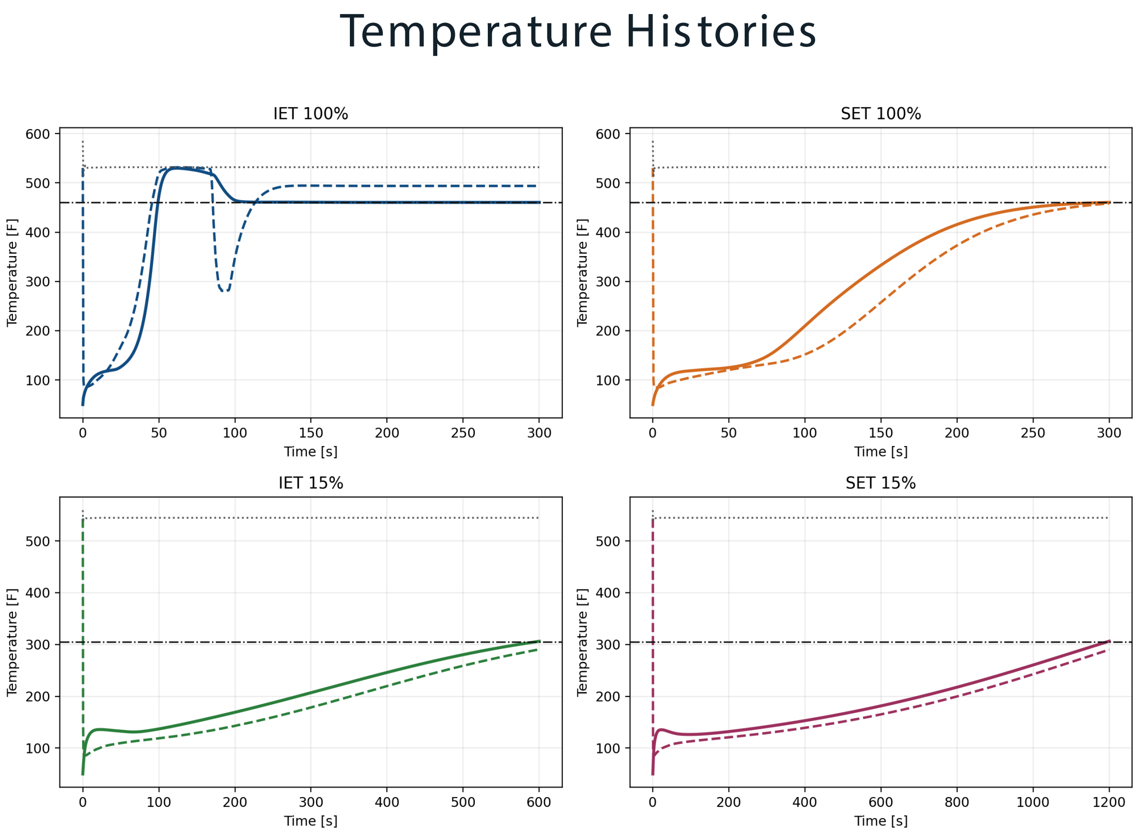
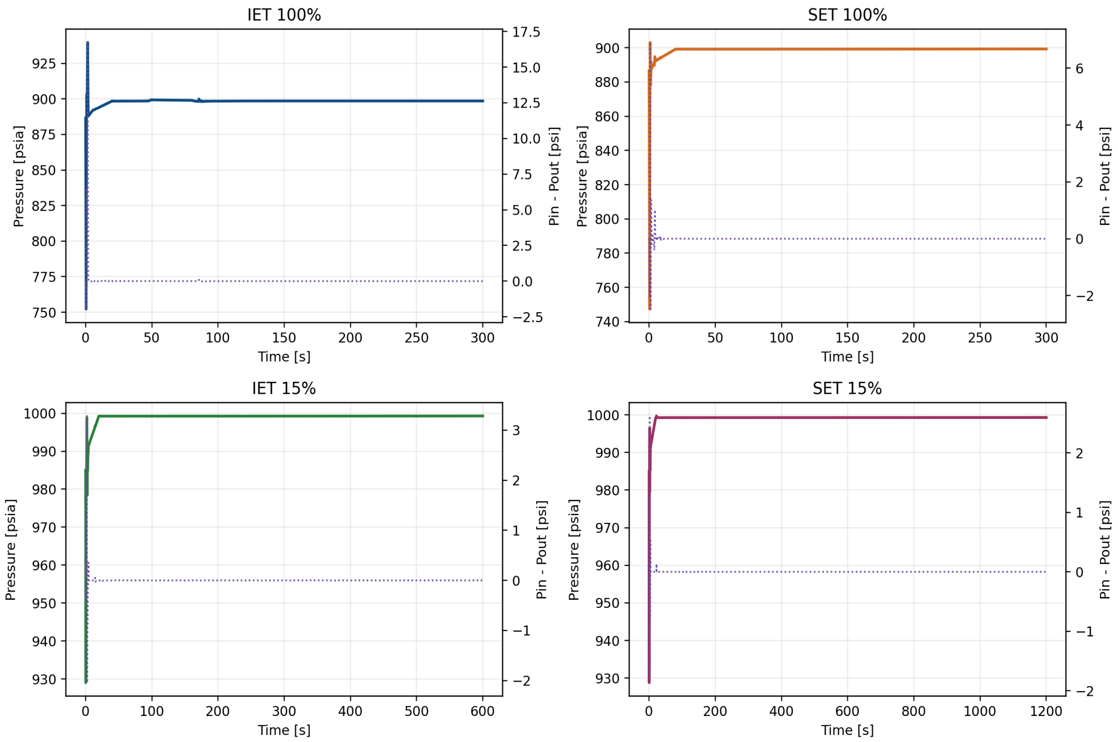
Thermal Response of the Shell and Tube Sides



- The steam-first startup conditions the shell before cold feedwater demand reaches full strength.
- High-load cases approach target faster because the wall thermal inertia is smaller relative to the through-flow enthalpy transport.
- Low-load cases remain slow but smooth, which is a physical conditioning effect rather than numerical noise.

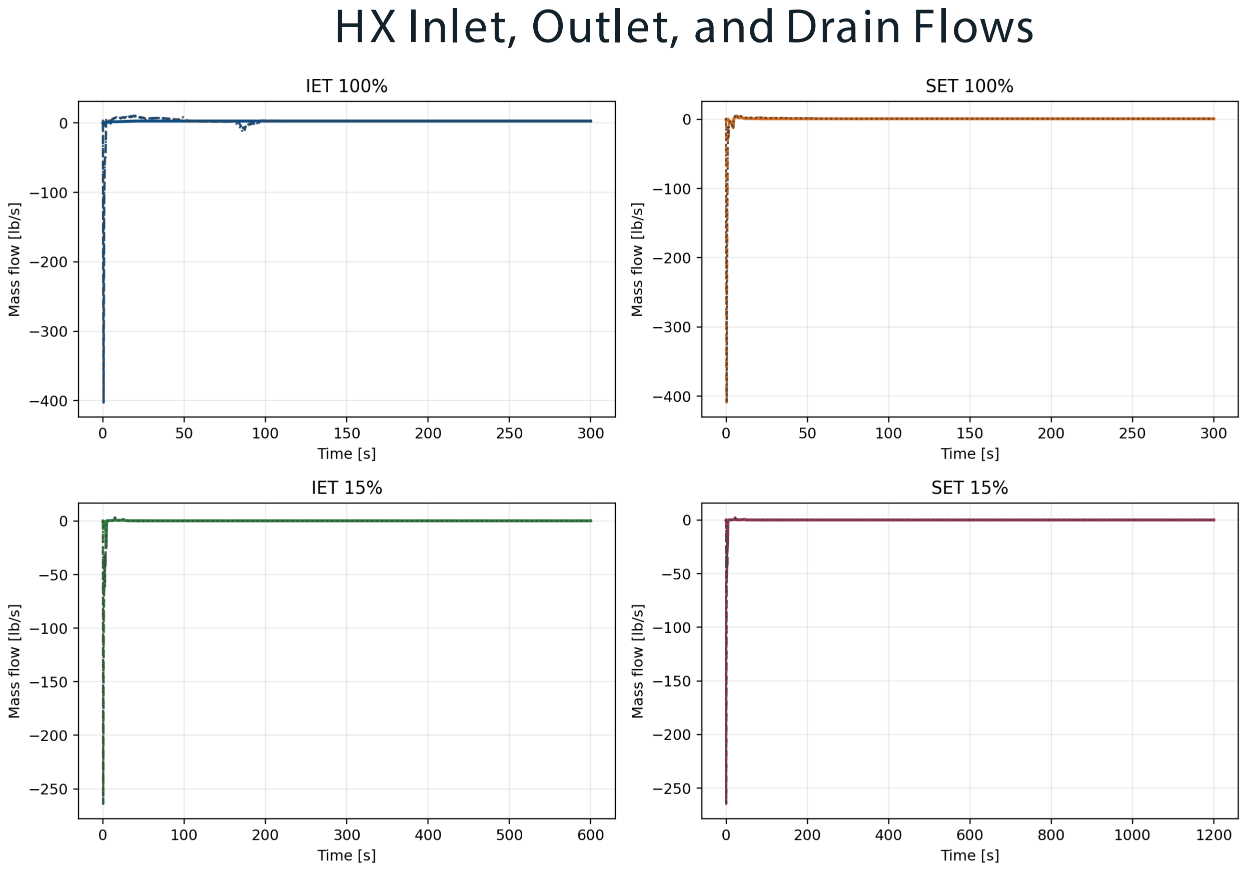
Shell-Side Pressure Response

Pressure and Local ΔP Histories



- Shell inlet and outlet pressure remain close because the exchanger acts mainly as a thermal sink rather than a strong throttling device.
- Pressure stays high while outlet temperature drops because condensation removes enthalpy much more strongly than the local shell path removes pressure.
- This is why outlet state, not shell pressure loss, is the dominant indicator of exchanger duty in the present operating modes.

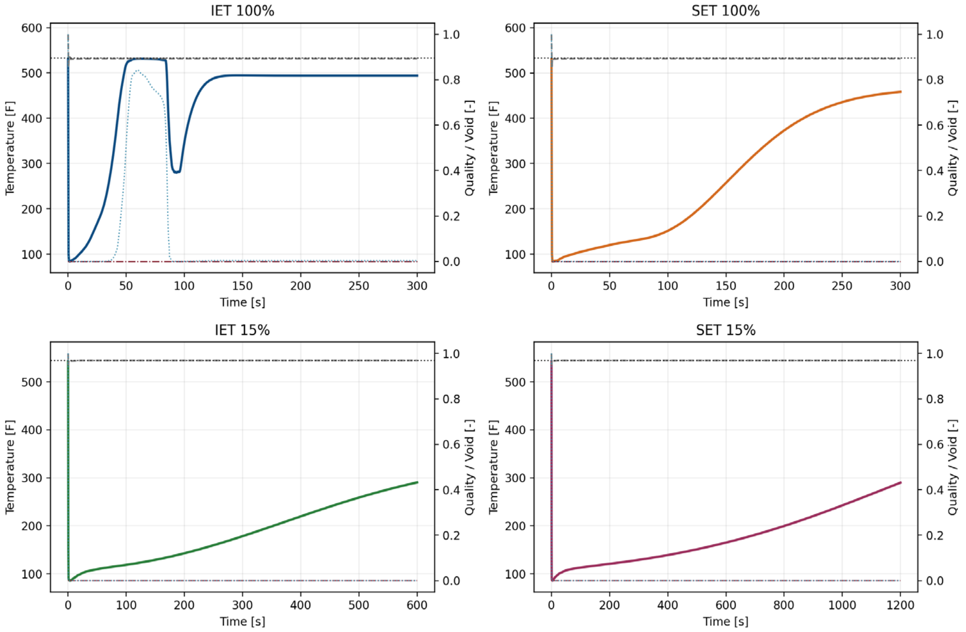
HX Inlet, Outlet, and Drain Flows



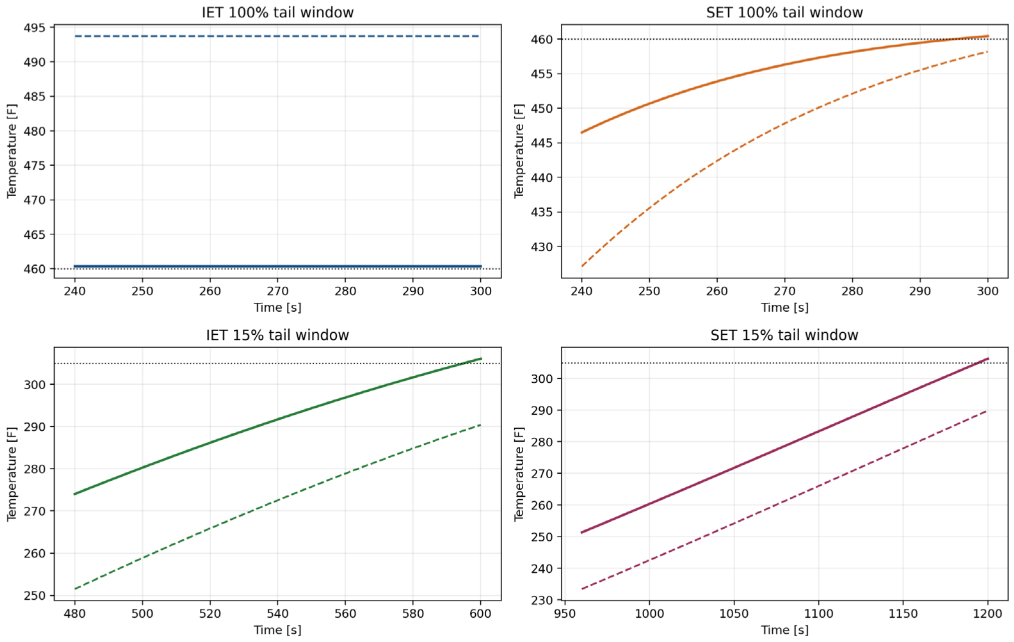
- No persistent sign changes or tail-window reversals remain in the production cases.
- The fixed-feedwater / tuned-steam formulation removes the underdetermined flow feedback that drove spurious oscillations in the earlier reduced-order models.
- Drain flow smoothly approaches the final condensate removal rate in every accepted run.

Outlet-State Evolution and Final Tail-Window Behavior

Shell Outlet State



Tail-Window Zoom



The outlet quality collapses to essentially zero in all cases, and the tail-window zoom confirms a smooth approach to the reported steady operating point.

Semi-Implicit Time Advancement and Linearized New-Time Solve

Difference Form and Linear Solve

- Semi-implicit time advancement (the implemented scheme keeps selected new-time terms implicit, but linear in the unknown state, so the advancement can be solved by direct matrix inversion):

$$A(\Delta t)x^{\mathrm{n+1}}$$
$$= b(x^{\mathrm{n}})x^{\mathrm{n+1}}$$
$$= x^{\mathrm{n}} + \delta x J(x^{\mathrm{n}}, \Delta t)\delta x$$
$$= -R(x^{\mathrm{n}})\left(\frac{M}{\Delta t}\right)(x^{\mathrm{n+1}} - x^{\mathrm{n}}) + L_{imp}(x^{\mathrm{n+1}}) + L_{exp}(x^{\mathrm{n}})$$

Algorithmic flow for HX transients

- New-time terms are chosen so the update matrix remains linear in the unknown state, allowing direct inversion with the default BPLU solver.
- RELAP distinguishes two-phase to two-phase, appearance, and disappearance cases during each update, which is important for startup condensation problems.
- The method also supports implicit hydrodynamic and heat-structure coupling, so wall temperature and fluid state can advance in one enlarged system when needed.