Constrained Selection of Moisture Separator Reheater (MSR) Reheat Enthalpy Rise and Low-Pressure Turbine (LPT) Inlet Conditions Under Wet-Steam Penalty Models

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1. Introduction

In pressurized water reactor (PWR) power-conversion systems, a Moisture Separator Reheater (MSR) is installed upstream of the low-pressure (LP) turbine to dry and reheat the wet exhaust steam from the high-pressure turbine; two-stage MSR reheat is standard in modern PWR turbine islands (e.g., APR1400) [1]. The MSR reheating section is typically supplied by high-temperature, high-pressure heating steam bled from the steam-generator (main steam) line and/or from intermediate high-pressure turbine extraction stages, thereby increasing the LP-turbine inlet superheat and suppressing in-path condensation [2,3]. This moisture mitigation improves cycle efficiency and reduces droplet-induced erosion risk in LP stages [2,4].

Since MSR reheating consumes valuable steam, its benefit is not monotonic with heating duty. Diverting a fraction of main steam (or extraction steam) to the MSR reduces the mass flow available for expansion in the turbine train—particularly through the upstream turbine stages—thereby decreasing turbine work [3]. In addition, excessive heating-steam flow and MSR pressure losses can impose a measurable penalty on net electrical output; consequently, the heating-steam/bleed requirement should be kept as low as practical while meeting the turbine wetness limit [4]. These competing effects make it necessary to identify an "efficient" reheat duty that achieves LP-turbine moisture control with minimal steam diversion.

The thermodynamic leverage of a given reheat enthalpy rise also depends strongly on the LP-turbine inlet and outlet pressures because saturation properties (e.g., the two-phase entropy gap and latent enthalpy) vary with pressure; thus, the same added superheat can yield different changes in exhaust quality on a $T-s$ diagram. To explore this dependence at the cycle level, this study employs Cotton/SCC-type empirical correlations that relate representative moisture (wetness) levels to isentropic-efficiency degradation [5,6]. A parametric map is constructed over pressure ratios and inlet pressures representative of publicly available PWR turbine cycles: published PWR design data report LP-turbine inlet pressures near ~ 0.9–1.0 MPa and intermediate LP extraction pressures around 0.23–0.46 MPa (stage-group pressure ratios on the order of ~3–4), while higher extraction pressures for feedwater heating can reach ~1.8–3.1 MPa [7,8]. Accordingly, P_{in} is swept from 100 to 5,000 kPa and PR from 3.0 to 4.5, with the MSR reheat enthalpy rise Δh_{rise} varied from 0 to 200 kJ/kg. The turbine model is implemented in MATLAB, and water/steam properties are obtained from NIST REFPROP [9].

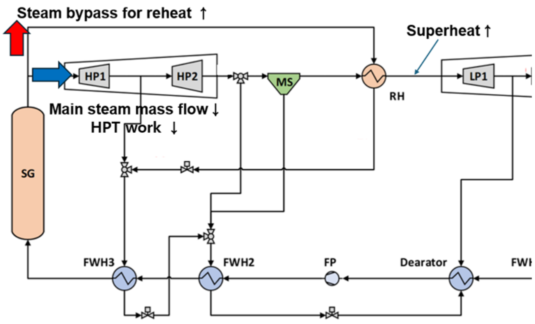

Fig. 1. Schematic diagram of PWR SMR power conversion system with reheater.

2. Methodology

2.1 Wet-steam penalty model and Wilson-line interpretation

In wet-steam turbines, the presence of liquid droplets causes additional aerodynamic and thermodynamic losses, and may accelerate erosion. Cotton reported empirical correlations that relate the reduction of isentropic efficiency to a representative (weighted) wetness level in the condensing region. Such correlations are well suited for conventional condensing stages where the wet region can be represented by quasi-equilibrium wetness metrics. In this work, Cotton's correlation is used to construct a first-order penalty factor that modifies a prescribed dry-steam (baseline) isentropic efficiency. [2]

Under rapid expansions, steam may cross the saturation line while remaining metastable state until nucleation begins. The onset of spontaneous condensation is commonly associated with the Wilson point/line, beyond which non-equilibrium (i.e., supersaturation) losses may appear even when the equilibrium liquid mass fraction is still small.

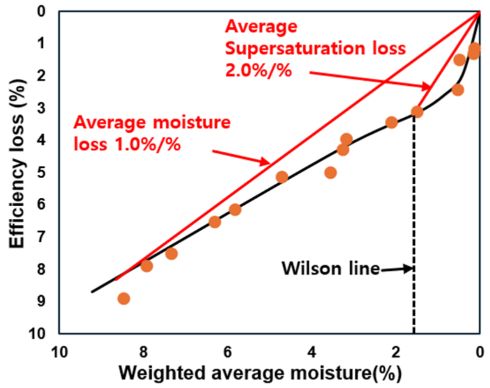

Fig. 2. Empirical relationship between turbine average moisture and isentropic efficiency loss.

2.2 Steam Turbine Modeling Incorporating Wet-Steam (Moisture) Loss Corrections

Figure 2 illustrates the idealized expansion of a low-pressure turbine (LPT) group on a *T–s* diagram, where the expansion in the reheated case proceeds from point 1' to point 2'. Without reheat, the inlet state is assumed to be saturated vapor at the LPT inlet pressure (point 1), and the steam expands directly to the outlet pressure prescribed by the specified pressure ratio, reaching point 2. With reheat, the inlet enthalpy is first increased at (approximately) constant pressure from point 1 to point 1' by an amount Δh_{rise}. This represents the MSR-side superheating, and the subsequent expansion then starts from the reheated state (point 1') and proceeds to point 2'. As can be inferred from the *T–s* trajectories in Fig. 3, the reheat process alters the inlet and exhaust entropies and thus is expected to change the mean wetness (moisture fraction) in the LPT exhaust; therefore, the corresponding wet-steam efficiency degradation is accounted for using Cotton's empirical moisture-loss correlation, consistent with the empirical Spencer–Cotton–Cannon (SCC) performance-correction methodology.

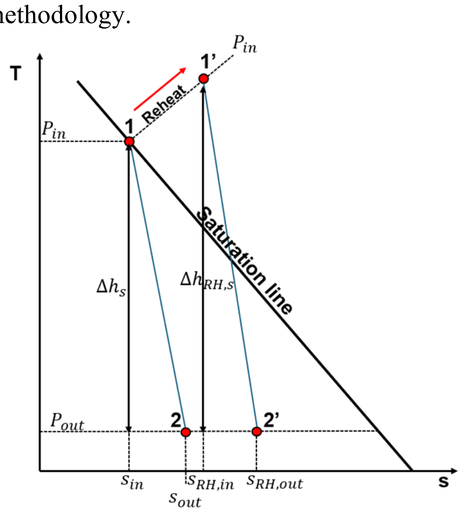

Fig. 3. Schematic T–s diagram for LPT expansion with and without reheat.

For an adiabatic turbine (work-output device), the isentropic efficiency is defined by the ratio of the actual to the isentropic enthalpy drop:

$$\eta \equiv \frac{h_{in} - h_{\text{out}}}{h_{in} - h_{\text{out,s}}} \quad (1)$$

where $h_{\text{out,s}}$ denotes the outlet enthalpy at the same outlet pressure p_{out} under an isentropic expansion ($s_{\text{out,s}} = s_{in}$). Rearranging Eq. (1) gives:

$$h_{\text{out}} = h_{in} - \eta\left(h_{in} - h_{\text{out,s}}\right) \quad (2)$$

Defining the isentropic enthalpy drop:

$$\Delta h_s \equiv h_{in} - h_{\text{out,s}} \quad (3)$$

Then,

$$h_{\text{out}} - h_{\text{out,s}} = (1-\eta)\Delta h_s. \quad (4)$$

From the fundamental thermodynamic relation for enthalpy:

$$dH = T\,dS + V\,dP. \quad (5)$$

It follows that on an isobaric line ($dP = 0$):

$$\left(\frac{\partial h}{\partial s}\right)_P = T. \quad (6)$$

In the saturated two-phase region at p_{out}, entropy is written as;

$$s = s_f + x\,s_{fg}, x = \frac{s - s_f}{s_{fg}}, \quad (7)$$

where x is the vapor quality, s_f is saturated-liquid entropy, and $s_{fg} = s_g - s_f$.

Combining Eqs. (6)–(7) yields, along the outlet isobar in the two-phase region,

$$\frac{dx}{dh} = \frac{1}{T\,s_{fg}}. \quad (8)$$

Using outlet saturation properties $\left(T_{\text{out}}, s_{fg,\text{out}}\right)$gives the linear relation:

$$x_{\text{out}} - x_{\text{out,s}} \approx \frac{h_{\text{out}} - h_{\text{out,s}}}{T_{\text{out}}\,s_{fg,\text{out}}}. \quad (9)$$

Substituting Eq. (4) into Eq. (9);

$$x_{\text{out}} = x_{\text{out,s}} + \frac{(1-\eta)\Delta h_s}{T_{\text{out}}\,s_{fg,\text{out}}}, \quad (10)$$

where $x_{\text{out,s}} = \frac{s_{in} - s_{f,\text{out}}}{s_{fg,\text{out}}}$. (11)

Finally, the latent enthalpy at p_{out} is $h_{fg,out} = T_{out}\, s_{fg,out}$ (latent heat equals $T\Delta s$at phase change).

Define $h_{vap} \equiv h_{fg,out} = T_{out} s_{fg,out}$ and rewrite Eq. (10) in enthalpy form.

Moisture effects are commonly incorporated by multiplying a "dry" efficiency by a moisture correction (e.g., SCC-type corrections in wet steam regions). In this work, the effective efficiency is modeled as:

$$\eta = \phi_{bss} \ \ \eta_{base}\ , (12)$$

and ϕ_{bss} is taken as the (empirical) Cotton-type piecewise function of outlet quality:

$$\phi_{bss} = \begin{cases} 0.5\, x_{out} + 0.468 & x_{out} < 0.984 \\ x_{out} & 0.984 < x_{out} < 1 \end{cases} \quad (13)$$

Using $x_{out,s} = (h_{out,s} - h_{f,out})/h_{vap}$ and $\Delta h_s = h_{in} - h_{out,s}$, Eq. (10) can be rearranged into explicit x_{out} expressions.

$$x_{out} = \frac{(h_{out,s} - h_{f,out}) + (1 - 0.468\,\eta_{base})\Delta h_s}{h_{vap} + 0.5\,\eta_{base}\,\Delta h_s} \quad (14a)$$

for $x_{out} < 0.984$ ($\phi_{bss} = 0.5x_{out} + 0.468$

$$x_{out} = \frac{(h_{out,s} - h_{f,out}) + (1 - 0.468\,\eta_{base})\Delta h_s}{h_{vap} + 0.5\,\eta_{base}\,\Delta h_s} \quad (14b)$$

for $0.984 < x_{out} < 1$ ($\phi_{bss} = x_{out}$).

All saturation properties $(T_{out},\ h_{f,out},\ s_{f,out},\ s_{fg,out},\ h_{vap})$ are evaluated at p_{out}, and $h_{out,s}$is evaluated at $(p_{out}, s = s_{in})$.

After reheat (state 1 → 3), let the turbine inlet state be $(h_{RH,in},\ s_{RH,in})$. The same derivation applies by replacing $(h_{in}, s_{in}) \rightarrow (h_{RH,in}, s_{RH,in})$in Eqs. (11)–(14), i.e.,

$$\Delta h_{s,RH} \equiv h_{RH,in} - h_{out,s,RH}, s_{out,s,RH} = s_{RH,in}\ . (15)$$

Substituting the reheated inlet state into the moisture-corrected formulation yields the following closed-form expressions for the reheated outlet quality $x_{out,RH}$:

$$x_{out} = \frac{h_{out,s} - h_{out,f} + \{1 - (0.5x_{RH,in} - 0.016)\}\eta_{base}\,\Delta h_s}{h_{vap} + 0.5\eta_{base}\,\Delta h_s}$$

for ($x_{out} < 0.984$), (16)

$$x_{out} = \frac{h_{out,s} - h_{out,f} + (2 - x_{RH,in})\eta_{base}\,\Delta h_s}{h_{vap} + 0.5\eta_{base}\,\Delta h_s}.$$

for ($0.984 < x_{out} < 1$). (17)

The benefit of reheat is quantified by the efficiency gain,

$$\eta_{gain} = \eta_{RH} - \eta_{nonRH}\,,\ (18)$$

which measures the efficiency increase attributable to reheat, and by the outlet quality gain,

$$x_{out,gain} = x_{out,RH} - x_{out,nonRH}\,, (19)$$

which quantifies the corresponding improvement in exhaust steam quality (i.e., reduced wetness) due to reheat.

2.3 Parametric Study Matrix

To quantify how the turbine inlet pressure, turbine pressure ratio, and the reheat-induced enthalpy rise affect the LPT expansion characteristics, a parametric study was conducted and the investigated cases are summarized in Table I. The baseline turbine isentropic efficiency under dry (moisture-free) conditions was fixed at $\eta_{base} = 0.90$, which is within the commonly reported range for modern, properly designed steam turbines. Reheat was modeled as a constant-pressure enthalpy increase, Δh_{rise}, varied from 0 to 200 kJ/kg. The LPT inlet pressure $P_{in}$was swept from 100 to 5,000 kPa, and simulations were performed for turbine pressure ratios PR = 3.0, 3.5, 4.0, 4.5.

Table I: . Parameter space and base assumptions for the LPT-group parametric study.

Parameter	Value
LPT inlet pressure [kPa]	100–5,000
Pressure ratio $[P_{in}/P_{out}]$	3.0, 3.5, 4.0, 4.5
Δh_{rise} [kJ/kg] (enthalpy rise via reheat)	0–200
η_{base} (Baseline isentropic efficiency)	0.90

3. Results

Fig. 4 summarizes the calculated maps of (left) isentropic efficiency gain and (right) outlet quality gain as functions of LPT inlet pressure and reheat enthalpy rise for four representative pressure ratios. The gain values are defined relative to the non-reheated (saturated inlet) case at the same inlet pressure and pressure ratio. For each pressure ratio, the red solid curve denotes the locus of maximum efficiency gain in the (Δh_{rise}, P_{in}) plane, and the red dashed curve indicates the Wilson-line boundary, used here as an interpretive marker for the onset of non-equilibrium condensation [4].

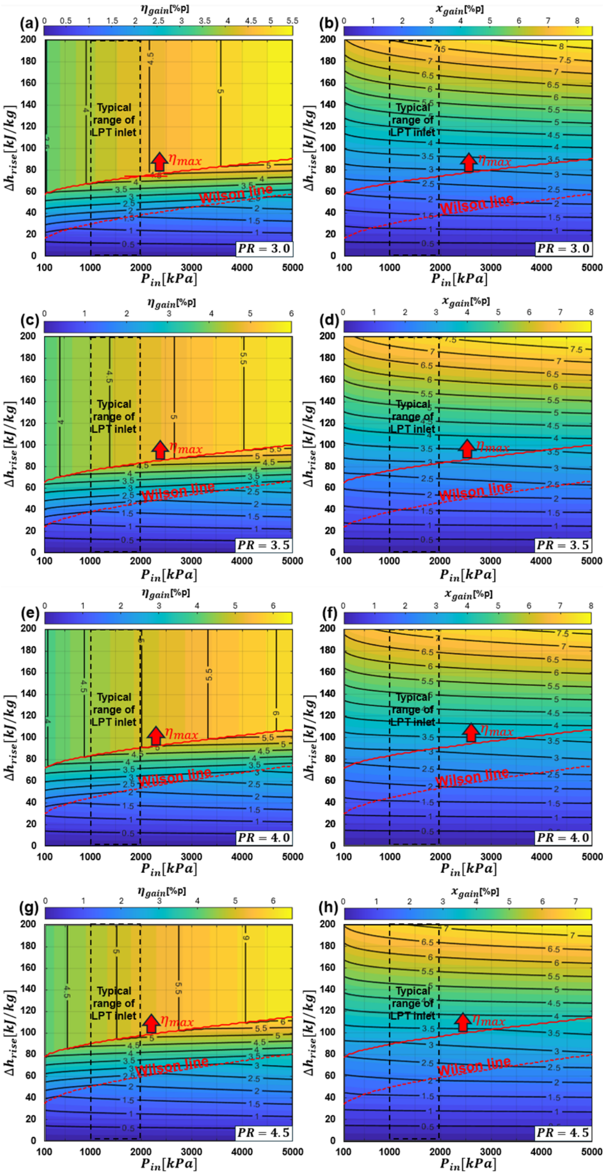

Fig. 4. Contour maps of isentropic efficiency gain (left) and outlet quality gain (right) versus LPT inlet pressure and reheat enthalpy rise for pressure ratios PR = 3.0, 3.5, 4.0, 4.5.

Three consistent trends emerge across the parameter space. First, the efficiency-gain contours are widely spaced under the Wilson line, indicating that efficiency improvement driven by reheat process is not particularly effective at low Δh_{rise}. In contrast, the contours tighten markedly between the Wilson line and the maximum-efficiency ridge, demonstrating that reheat becomes most effective when it shifts the expansion trajectory from a strongly wet region toward the metastable limit (Wilson line), thereby reducing the mean wetness level and the associated empirical moisture-loss penalty.

Second, the outlet-quality gain shows the opposite behavior: quality-gain contours are dense below the Wilson line and become sparse above it. This implies that a moderate reheat duty can significantly increase outlet quality before the expansion approaches the metastable boundary, whereas additional reheat beyond that point yields diminishing returns in terms of further moisture reduction.

Third, the relative positions of the Wilson line and the maximum-gain locus depend strongly on inlet pressure. At lower inlet pressures, the two curves are separated by a wider gap in Δh_{rise}; thus, the operating point can cross into the region of rapid efficiency improvement with a smaller enthalpy rise, but the maximum achievable efficiency gain is modest. At higher inlet pressures, a larger enthalpy rise is required to approach the Wilson-line region, yet the attainable peak efficiency gain increases.

Increasing the pressure ratio shifts both the Wilson line and the maximum-gain locus toward larger Δh_{rise} and increases both efficiency gain and quality gain. A larger pressure ratio drives the isentropic expansion deeper into the two-phase region, so reheat has more leverage to mitigate wetness and recover efficiency. From a design perspective, this suggests that optimal MSR duty cannot be specified independently of the targeted LPT pressure ratio and inlet pressure.

Overall, Fig. 4 highlights a trade-off region in which moisture mitigation (quality gain) begins to saturate while efficiency gain continues to increase. This region is particularly relevant for nuclear turbine design since it provides a quantitative basis to select the reheat duty that satisfies blade-protection constraints while approaching the point of maximum efficiency benefit.

Fig. 5 is obtained by superimposing black dotted curves on the contour maps of Fig. 4a,b. These dotted curves represent the reheat enthalpy rise Δh_{rise} that is achievable as a function of the mass-flow-rate ratio between the hot side and cold side of the reheater, computed using the standard effectiveness–NTU (ε–NTU) method. Specifically, the reheater hot side is assumed to be saturated steam at 5800 kPa, and the heat-exchanger effectiveness is fixed at $\varepsilon = 0.9$. For each hot-to-cold flow ratio $\dot{m}_h/\dot{m}_c$ (where $\dot{m}_h$ is the reheater bypass/heating-steam flow and $\dot{m}_c$is the LPT inlet flow), the ε–NTU formulation provides the heat transfer rate $\dot{Q} = \varepsilon C_{\mathrm{min}}(T_{h,\mathrm{in}} - T_{c,\mathrm{in}})$, which is then converted into a cold-side specific enthalpy increase through $\Delta h_{\mathrm{rise}} = \dot{Q}/\dot{m}_c$. This construction links the cycle-level "required Δh_{rise}" directly to a physically interpretable "required bypass ratio," consistent with the fact that PWR turbine plants route a portion of steam to the MSR to reheat the main-cycle steam.

If the design objective is to improve the turbine exhaust quality by 1.5 percentage points (i.e., $\Delta x_{\mathrm{out}} = +0.015$), the most bypass-efficient selection is the point where the Wilson-line boundary intersects the +1.5%pquality-gain contour in Fig. 4b. This intersection (approximately$P_{\mathrm{in}} \approx 2000$kPa, orange circle in Fig. 5b) achieves the target moisture reduction with the minimum reheater bypass, since wetness mitigation becomes most responsive as the expansion trajectory approaches the metastable condensation limit represented by the Wilson line.

After satisfying the moisture-protection target, any further increase in reheater bypass intended to enhance turbine efficiency should be chosen below the upper limit (best-efficiency line) indicated in Fig. 5a. In this region, increasing the reheater-bypass (heating-steam) flow leads to a rapid rise in the predicted efficiency gain, but it simultaneously reduces the turbine work available in other sections of the turbine train due to the associated mass-flow redistribution; therefore, these offsetting work reductions must be monitored as the bypass is increased. Accordingly, further bypass for efficiency improvement should be selected only within the area below the upper-limit (best-efficiency) line, where the efficiency gain accelerates while avoiding operating points in which the bypass-induced work penalties in other turbine stages become dominant.

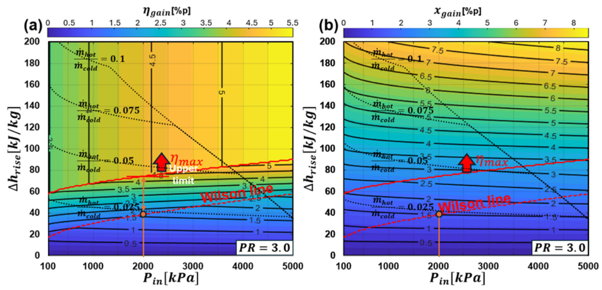

Fig. 5 Contour-map-based selection of P_{in} and Δh_{rise} to achieve a 1.5%p increase in LPT outlet quality

4. Conclusions

A parametric cycle-level map was developed to evaluate how MSR-type reheat at the inlet of an LPT group affects both effective isentropic efficiency and outlet quality when wet-steam penalties are considered. Key conclusions are as follows:

• Reheat provides the largest efficiency benefit when it reduces expansion wetness between the Wilson-line boundary and best efficiency locus.

• Outlet quality improves rapidly with moderate reheat in the sub-Wilson region, but additional reheat beyond the Wilson-line region yields diminishing returns in moisture mitigation.

• Inlet pressure trades reheat duty against achievable peak gain. A lower LPT inlet pressure allows the Wilson-line region to be reached with a smaller enthalpy rise Δh_{rise}, whereas a higher inlet pressure can produce a larger peak efficiency gain but generally requires a larger reheat duty to realize that benefit.

• Higher pressure ratio increases both efficiency gain and quality gain and shifts the optimal enthalpy-rise range upward; therefore, reheat design should be coordinated with the intended LPT pressure ratio.

Accordingly, the selection of the reheater duty Δh_{rise} and the LPT inlet conditions (P_{in}, PR)should be treated as a constrained design problem: the preferred operating/design point is the best-efficiency locus (i.e., the region of maximum η_{gain}) subject to the requirement that the post-reheat in-turbine dryness fraction (quality) remains within the turbine's allowable moisture limit. In other words, (P_{in}, PR, Δh_{rise})should be chosen so that the reheat-shifted expansion trajectory achieves the highest efficiency improvement while maintaining an acceptable wetness margin for blade durability and avoiding excessive condensation-related penalties.

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