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INTRODUCTION TO THERMODYNAMICS

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ON SCIENCE AND TECHNOLOGY OF SCWRS

LECTURE SC06

INTRODUCTION TO THERMODYNAMICS

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PREFACE

This lecture is devoted to supercritical "steam" Rankine cycles and consists of three parts: Part 1. Simplified Thermodynamic Cycles; Part 2. SCWRs Cycles; and Part 3. Co-Generation of Hydrogen. All these parts are based on a course of the classical engineering thermodynamics. Therefore, for all general terms, thermodynamics laws and principles, please refer to general engineering thermodynamic textbooks, for examples: Çengel and Boles (2011); Borgnakke and Sontag (2009); Moran and Shapiro (2008). Also, this lecture is based on the material presented in Lecture SC02 Introduction and Historical Development of SCWRS.

PART 1. SIMPLIFIED THERMODYNAMIC SUPERCRITICAL RANKINE CYCLES

1.1. Introduction

Currently there are a number of Generation IV SuperCritical Water-cooled nuclear Reactor (SCWR) concepts under development worldwide. The main objectives for developing and utilizing SCWRs are: 1) Increase gross thermal efficiency of current Nuclear Power Plants (NPPs) from 33 – 35% to approximately 45 – 50%, and 2) Decrease capital and operational costs and, in doing so, decrease electrical-energy costs.

SCW NPPs will have much higher operating parameters compared to those of current NPPs (i.e., pressures of about 25 MPa and outlet temperatures up to 625°C). Additionally, SCWRs will have a simplified flow circuit in which steam generators, steam dryers, steam separators, etc. will be eliminated. Furthermore, SCWRs operating at higher temperatures can facilitate an economical co-generation of hydrogen through thermo-chemical cycles (particularly, the copper-chlorine cycle) or direct high-temperature electrolysis.

To decrease significantly development costs of an SCW NPP, to increase its reliability, and to achieve similar high thermal efficiencies as advanced fossil-fuel steam cycles it should be determined whether SCW NPPs can be designed with a steam-cycle arrangement that closely matches that of mature SuperCritical (SC) fossil-fuel thermal power plants (including their SC turbine technology). The state-of-the-art SC steam cycles in fossil-fuel power plants are designed with a single-steam reheat and regenerative feedwater heating and reach thermal steam-cycle efficiencies up to 54% (i.e., net plant efficiencies of up to 43 - 48% on a Higher Heating Value (HHV) Basis).

Before analyzing actual SC "steam" cycles simplified no-reheat, single-reheat, and double-reheat cycles without heat regeneration and a single-reheat cycle with heat regeneration based on the expected steam parameters of future SCW NPPs will be analyzed first in terms of their thermal efficiencies.

On this basis, several conceptual steam-cycle arrangements of pressure-tube SCWRs, their corresponding T–s diagrams and steam-cycle thermal efficiencies (based on constant isentropic turbine and polytropic pump efficiencies) are presented in this lecture.

Also, one of the main objectives of this lecture is to analyze possible steam-cycle arrangements and to evaluate conceptually their complexity and adaptability to current SCW NPP concepts.

The following analysis provides a thermodynamic comparison of three possible steam-cycle arrangements: 1) No-reheat cycle; 2) Single-reheat cycle; and 3) Double-reheat cycle. Moreover, the thermodynamic benefit of regenerative heating is also presented based on a single-reheat cycle.

All of the steam-cycle arrangements considered here are based on the Rankine cycle, and all of the cycles are based on the main "steam" pressure and temperature of 25 MPa and 625 °C, respectively. Where reheat is present, the reheat temperature was assumed to be 625 °C or 700 °C as indicated. All cycles are based on a condensing pressure of 6.8 kPa. The amount of cycle-heat input (Q) listed in Tables 1 – 4 is based on a cycle with the total useable work ("thermal output") of 1200 MW_{th}.

It should be noted that thermal-cycle efficiencies listed in Tables 1 to 4 are based on the following simplifying assumptions:

- Isentropic turbine efficiency of 88% for all turbine sections (cylinders);
- No mechanical losses (e.g., bearing losses);
- No steam turbine packing leakage or gland steam-system losses;
- No turbine exhaust losses;
- No generator losses;
- 84% polytropic efficiency for all pumps;
- Reactor feed-pump discharge pressure is 127% of turbine throttle pressure;
- Condensate pump discharge pressure is 0.7 MPa (100 psi) above the deaerator pressure;
- Reheat system ΔP is 8% of the cold reheat pressure;
- Turbine extraction piping ΔP is 5% of turbine stage pressure and includes turbine nozzle losses;
- No piping heat losses; and
- No reactor heat losses.

For illustration purposes, the T–s diagrams show irreversible compression and both reversible (dashed line) and irreversible (solid line) expansions. For the purpose of T–s diagrams heat addition takes place at constant pressure (i.e., not all system pressure drops are shown in the T–s diagrams).

1.2. Cycles without Heat Regeneration

1.2.1. No-Reheat Cycle

The most basic steam-cycle configuration that could be used in an SCW NPP is the non-reheat cycle without heat regeneration. A simplified cycle arrangement and the corresponding T–s diagram of a no-reheat cycle are shown in Figs. 1a and 1b, respectively. The corresponding heat-transfer rates of this cycle are listed in Table 1[1].

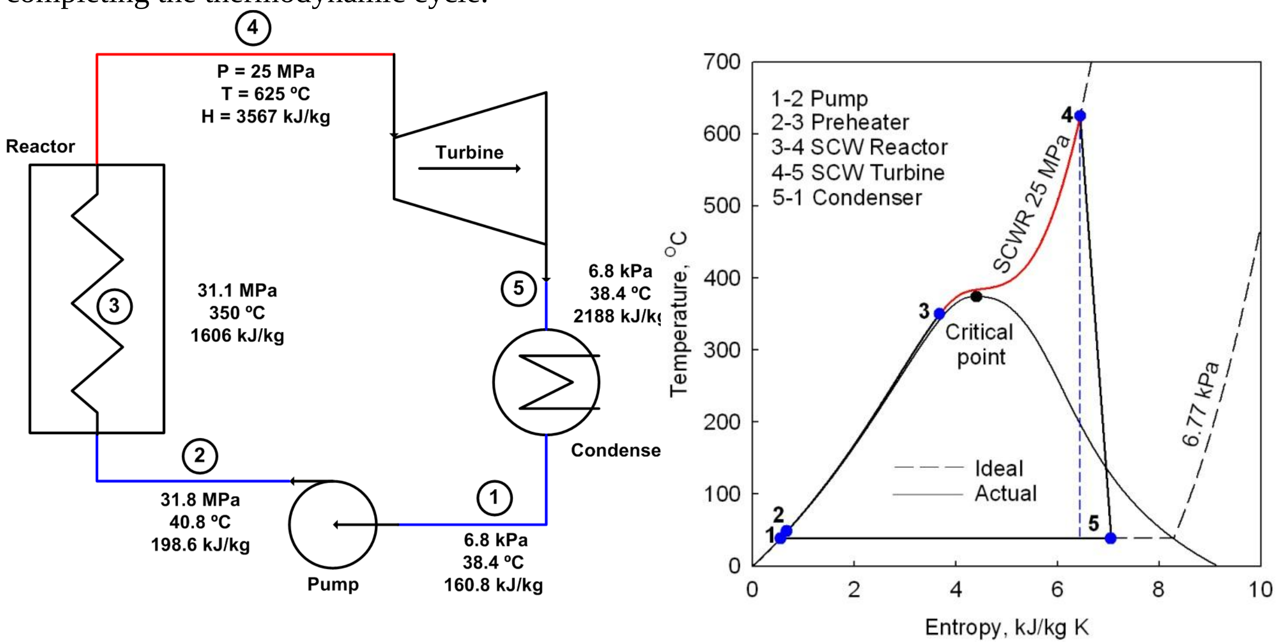
Saturated feedwater (i.e., condensate) from a condenser (Point 1: Pressure 6.8 kPa and temperature 38.4°C) enters a pump, which compresses the condensate (water) to a supercritical operating pressure (Point 2: Pressure 31.8 MPa and temperature 40.8°C). After that, the supercritical water is heated in a preheater (for simplification purposes the preheater is considered to be a part of the reactor) between Points 2 and 3 from 40.8°C to 350°C. The preheater outlet pressure shown in Fig. 1a is only for reference purposes. It is expected that, due to the relatively low flow rates inside the reactor and piping, the pressure drop from the pump discharge to the reactor outlet might be as little as 1 – 2 MPa.

[1] The water/steam properties were calculated using WinSteam 3.1 software, which is based on the 1997 formulations published by the International Association for the Properties of Water and Steam (IAPWS-IF97).

Further on, supercritical water continues to flow through the steam generator (reactor), being heated from 350°C to 625°C (Points 3 – 4).

In the next step, supercritical water (here it is better to say supercritical "steam", because this actually incorrect term suits better in connection with SC turbines) at the pressure of 25 MPa and temperature of 625°C (Point 4) enters the turbine where it expands and produces mechanical work by rotating the turbine shaft connected to an electrical-generator rotor. Inside the turbine, the steam expands and pressure and temperature drop; and saturated steam at a low pressure leaves the turbine at Point 5 (pressure 6.8 kPa and temperature 38.4°C).

The steam is condensed at the pressure of 6.8 kPa and saturation temperature of 38.4°C inside a condenser (Points 5 – 1). The condenser is basically a heat exchanger that is using water from a cooling tower, river or lake as cooling medium. The water (condensate) leaves the condenser as saturated liquid (in reality it would be slightly subcooled) and enters the pump (Point 1), thus completing the thermodynamic cycle.



(a) (b)

Fig. 1: No-Reheat Cycle Layout (a) and Corresponding T–s Diagram (b).

Table 1: No-Reheat Cycle Heat-Transfer Rates.

Main "Steam" Temperature, °C	**625**	
Mass Flow Rate , kg/s	**870**	
Stage	***ΔH*, kJ/kg**	***Q*, MW_{th}**
Pumps	38	33
Preheater	1407	1225
Reactor	1961	1707
Turbine	–1379	–1200
Condenser	–2027	–1765
Efficiency with pump work, %	**40.5**	
Efficiency without pump work, %	**40.9**	

The difference between the energy added to the cycle (energy added in the reactor plus energy added by the pumps) and the heat rejected through the condenser represents the useful work generated by this configuration. The efficiency of the cycle is the ratio of total useful work and heat added to the cycle. As per Table 1, the total thermal energy added to the cycle is 3406 kJ/kg. The energy rejected through the condenser is 2027 kJ/kg. Therefore, the thermal efficiency of the cycle is about 40.5% if the work provided by the pumps is included as a heat source, and 40.9% if the heat added by the pumps is neglected or considered ”free” energy.

The advantages of this no-reheat cycle as part of an SCW NPP are:

1) A relatively simple general SCW NPP layout that lowers capital costs associated with the design, construction and operation of an NPP; and
2) A relatively simple SCWR design, which can be a Pressure Tube (PT) or Pressure Vessel (PV) types.

However, there are currently no SC turbines operating without a reheat stage. This is mainly due to economic reasons. Also, it should be noted that a high main “steam” pressure in no-reheat turbines results in high turbine exhaust moisture and would require technical means to prevent moisture induced erosion in the low-pressure stages of the turbine. Also, the thermal efficiency of a no-reheat SCWR NPP might not be high enough to be competitive with the efficiencies of current SC or combined-cycle thermal power plants.

1.2.2. Single-Reheat Cycle

It is well known that steam reheat increases the thermal efficiency of the cycle Çengel and Boles (2011); Borgnakke and Sontag (2009); Moran and Shapiro (2008)). Moreover, steam reheat reduces the amount of moisture in the last stages of the turbine, therefore eliminating the need for moisture removal equipment. In general, incorporation of a single reheat in modern power plants improves the cycle efficiency by 2 to 4%, compared to the no-reheat cycle, by increasing the average temperature at which heat is added to the steam. For that reason, the vast majority of SC thermal power plants operating around the world employ SC “steam” generators and turbines with a single-reheat cycle (for details, see Lecture SC02 or Pioro and Duffey (2007)).

A single-reheat steam-cycle arrangement is shown in Fig. 2a. The corresponding T–s diagram and heat-transfer rates are shown in Fig. 2b and Table 2, respectively. Compared to the no-reheat cycle the single-reheat cycle uses two turbine sections or cylinders: a High-Pressure (HP) turbine and an Intermediate- (IP) or/and a Low-Pressure (LP) turbine(s) (Mokry et al., 2008). Moreover, a steam-reheat is added to the cycle.

The SC “steam” from a reactor is expanded inside the HP turbine from the supercritical pressure of 25 MPa and temperature of 625°C (Point 4) to an intermediate pressure of 5.3 MPa and temperature of 382°C (Point 5). The subcritical-pressure steam leaves the HP turbine in a superheated state and is sent back to the reactor where it is re-heated from 382°C to 625°C (Point 6). After the reactor reheat, the superheated reheat steam is expanded in the IP/LP turbine(s) to a sub-atmospheric pressure of 6.8 kPa and temperature of 38.4°C (Point 7), at which it is condensed in the condenser.

For this arrangement, the total heat energy added to the cycle is approximately 3986 kJ/kg and the energy rejected through the condenser is 2289 kJ/kg. Thus, the thermal efficiency of the cycle is about 42.6% if the work provided by the pumps is included as a heat source, and 43.0% if the heat added by the pumps is neglected or considered ”free” energy.

If a higher steam-reheat temperature is achieved, for example 700°C, the thermal efficiency would rise by about 0.9%. However, this increase in temperature may be very expensive accounting on special materials to be used inside the reactor. Therefore, on the current level of technology, this option may not be viable in spite of the increased thermal efficiency.

In order to maximize the thermal-cycle efficiency of the SCW NPPs it would be beneficial to include nuclear steam reheat, similar to the reheat cycle of fossil power plants. This nuclear steam reheat is easier to implement inside PT reactors compared to PV reactors. Currently, the development challenges are to minimize the higher costs of materials needed at higher temperatures, and to enhance safety and performance margins despite increased pressures, while retaining the economic advantages.

Also, if the steam reheat is adopted at lower pressures, even higher temperatures (limited only by materials corrosion rates) could allow direct thermochemical production of hydrogen, due to increased reaction rates, which could be utilized in fuel cells, hydrogen vehicles and as a part of chemical processing or hydrocarbon upgrading.

The advantages of the single-reheat cycle are:

1) Higher thermal efficiency, which corresponds to that of the current SC thermal power plants.
2) High reliability due to proven state-of-the-art SC turbine technology; and
3) Reduced development costs based on a wide variety of SC turbines manufactured by various companies worldwide.

However, the major disadvantage is that significant changes are required to the reactor design due to addition of the nuclear steam reheat at lower pressures. Also, the NPP layout is more complex compared to that of the no-reheat cycle.

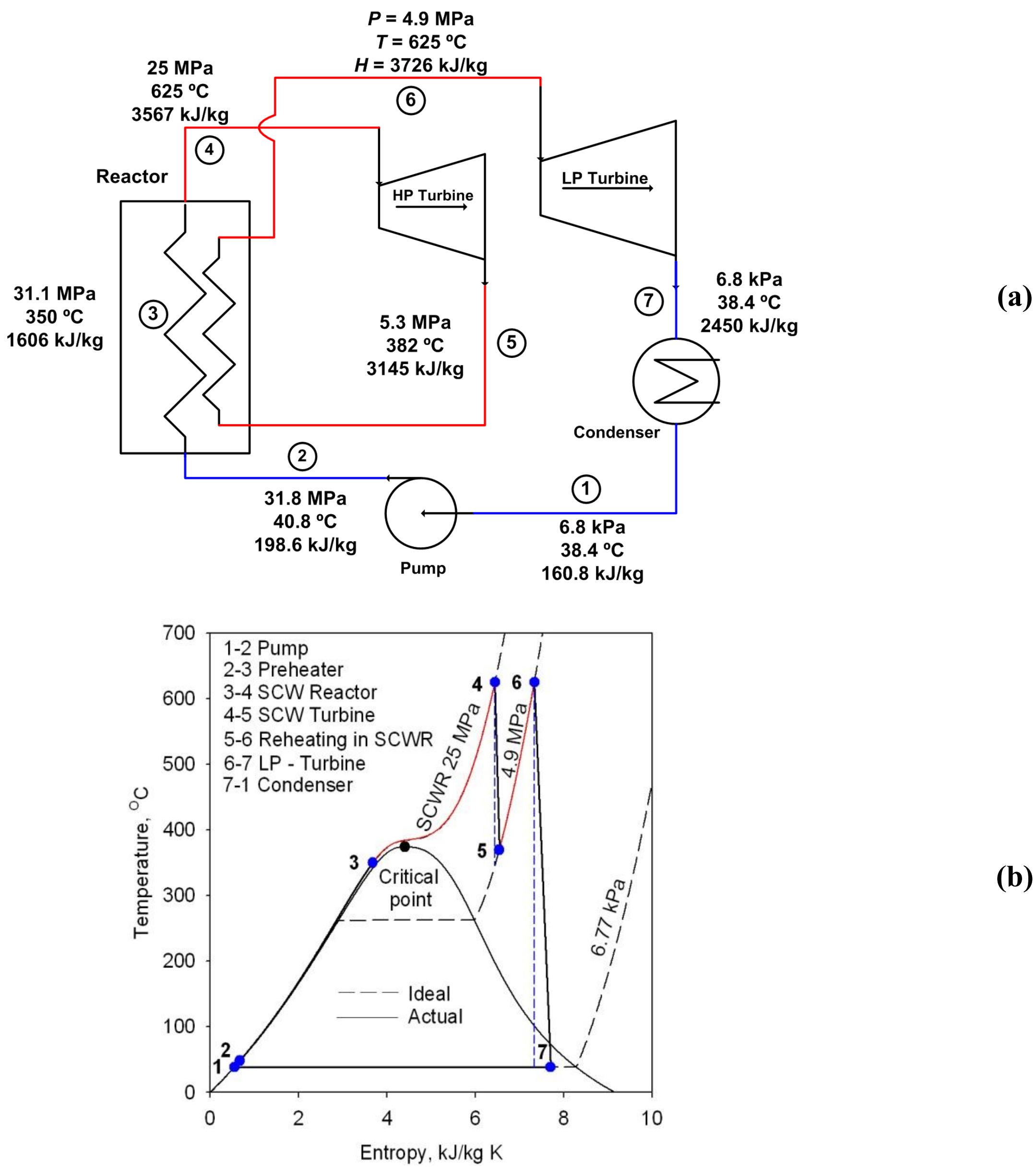


Figure 2: Single-Reheat Cycle Layout (a) and Corresponding T–s Diagram (b).

Table 2: Single-Reheat Cycle Heat-Transfer Rates.

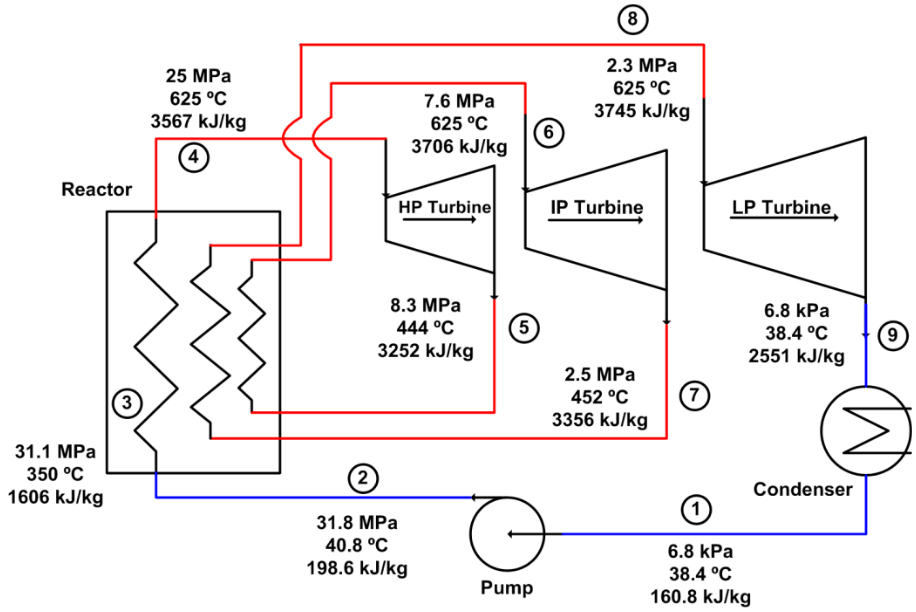
Steam Temperature Main/Reheat, °C	**625 / 625**		**625 / 700**	
Mass Flow Rate, kg/s	**707**		**659**	
Stage	**ΔH, kJ/kg**	**Q, MW$_{th}$**	**ΔH, kJ/kg**	**Q, MW$_{th}$**
Pump	38	27	38	25
Preheater	1407	994	1407	928
Reactor	1961	1385	1961	1292
HP Turbine	–421	–298	–441	–291
Reheater	580	410	775	511
LP Turbine	–1277	–902	–1380	–910
Condenser	–2288	–1617	–2360	–1556
Efficiency with pump work, %	**42.6**		**43.5**	
Efficiency without pump work, %	**43.0**		**43.9**	

1.2.3. Double-Reheat Cycle

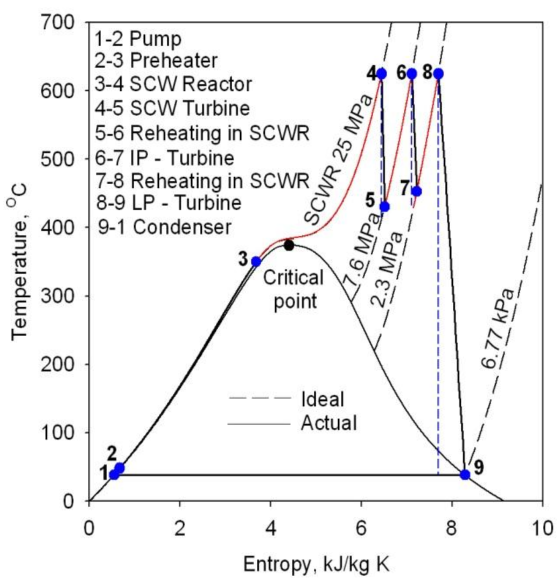
In general, every time when steam in-between turbine stages is re-heated, the thermal efficiency of the cycle is improved. However, the additional reheat also increases capital costs of equipment. It has been determined in the past that the use of more than two reheat stages is not practical in SC thermal power plants, because the theoretical improvement in the thermal efficiency from the second reheat is approximately only half of that which resulted from a single reheat. The fact that there are only a few double-reheat turbines currently operating around the world (Mokry et al., 2008) appears to support the finding that there is a diminishing return for every reheat stage added.

The double-reheat cycle arrangement and the corresponding T–s diagram are shown in Figs. 3a and 3b, respectively. The corresponding heat-transfer rates are listed in Table 3. In comparison to the single-reheat cycle (for details, see Section 1.2.2) the double-reheat cycle has three turbine sections: 1) HP turbine; 2) IP turbine(s); and 3) LP turbine(s). Moreover, the second steam reheat is added to the reactor.

The SC "steam" from the reactor is expanded inside the HP turbine from the supercritical pressure of 25 MPa and temperature of 625°C (Point 4) to an intermediate pressure of 7.6 MPa and temperature of 444°C (Point 5). The subcritical pressure steam leaves the HP turbine in a superheated state and is sent back to the reactor where it is heated from 444°C to 625°C (Point 6). After the reactor reheater, the superheated steam is expanded in the IP turbine(s) to a pressure of 2.3 MPa and temperature of 452°C (Point 7). The subcritical-pressure steam leaves the IP turbine in a superheated state and is sent back to the reactor where it is heated from 452°C to 625°C (Point 8). After the second reactor reheater, the superheated steam is expanded in the LP turbine(s) to sub-atmospheric pressure of 6.8 kPa and temperature of 38.4°C (Point 9), and condensed in the condenser.



(a)



(b)

Fig. 3: Double-Reheat Cycle Layout (a) and Corresponding T–s Diagram (b).

Table 3: Double-Reheat Cycle Heat-Transfer Rates.

Steam Temperature Main/Reheat/Reheat, °C	**625 / 625 / 625**	
Mass Flow Rate, kg/s	**646**	
Stage	***ΔH*, J/kg**	***Q*, MW_{th}**
Pump	38	24
Preheater	1407	909
Reactor	1961	1267
HP Turbine	–314	–203
Reheater 1	453	293
IP Turbine	–349	–226
Reheater 2	389	251
LP Turbine	–1194	–771
Condenser	–2390	–1544
Efficiency with pump work, %	**43.7**	
Efficiency without pump work, %	**44.1**	

For this cycle, the total heat energy added to the cycle is 4248 kJ/kg and the energy rejected through the condenser is 2390 kJ/kg. Thus, the thermal efficiency of the cycle is about 43.7% if the work provided by the pumps is included as a heat source, and 44.1% if the heat added by the pumps is neglected or considered ”free” energy.

The only advantage of the double-reheat cycle is the higher thermal efficiency compared to that of the no-reheat and single-reheat cycles. However, the reactor-core design of a double-reheat steam cycle is even more challenging. Also, capital costs of introducing another turbine and reactor stage should be carefully weighed against the benefits of the increased thermal efficiency.

1.3. Cycle with Heat Regeneration

It has been well established (Çengel and Boles (2011); Borgnakke and Sontag (2009); Moran and Shapiro (2008)) that an increased in feedwater temperature leads to improvement in the cycle efficiency if heat from within the steam cycle is used to preheat the feedwater. A practical regeneration process can be accomplished by extracting steam from the turbine at various points. Although this steam could have been expanded inside the turbine to produce more useful work, it is used to preheat the feedwater instead.

Aside from improving the cycle efficiency, regeneration provides a convenient way of a deaerating the feedwater to prevent corrosion in various components of the plant. Another benefit is the minimization of the volume flow rate of steam at the final stages of the turbine.

The number of feedwater heaters is usually determined based on an economical evaluation. From a thermodynamic point of view, an infinite number of heaters would maximize the cycle efficiency. However, this is not possible in the practical world. The optimum number of heaters for a modern SC plant has been determined between 8 and 10. They are classified as LP heaters,

Deaerators (mixing chambers) and HP heaters depending on the operating pressure and the type of heat exchanger used.
As previously mentioned, the exact point where extraction steam is taken from plays an important role in the efficiency of the thermal cycle and also depends on the turbine design.

It is difficult to show the benefit of the regenerative-feedwater heating in a T–s diagram, because the flow rates at the various points in the cycle are not the same and hence, the areas within the T–s diagram do not represent the total work and heat rejection of the cycle. For the same reason, it is not meaningful to show the heat sources and sinks in Table 4 on the basis of energy per unit mass (kJ/kg). Table 4 is based on a total useful heat of 1200 MW_{th}.

The arrangement analyzed below involves three feedwater heaters: one LP heater, one deaerator and one HP heater (for details, see Fig. 4 and Table 4). Steam is extracted from the HP turbine and used to heat the feedwater flowing through the Heat Exchanger (HX) 3 (closed-type feedwater heater). The LP turbine supplies extraction steam for the LP feedwater heater (HX1). Also, a fraction of the steam exhausted from the HP turbine is diverted to heat the water in the deaerator (HX2).

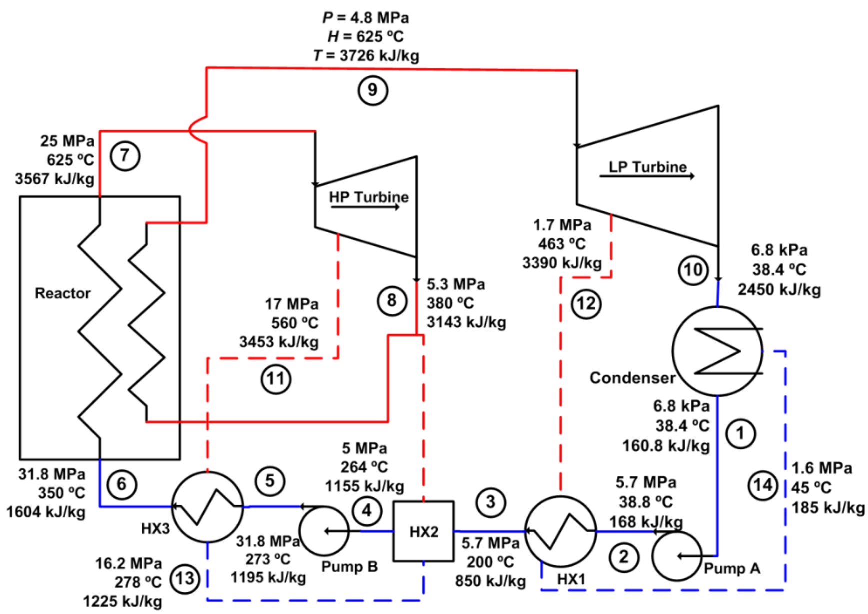


Fig. 4: Single-Reheat Cycle with Heat Regeneration through Single Deaerator and Two Feedwater Heaters.

Table 4 summarizes the heat transfer rates and thermal efficiency associated with this cycle (see Fig. 4). The total heat added to the cycle is approximately 2564 MW_{th}. At the same time, the heat rejected in the condenser is approximately 1364 MW_{th}. Thus, the thermal efficiency is about 46.8% if the work provided by the pumps is included as a heat source, and 47.7% if the heat added by the pumps is neglected or considered ”free” energy.

As can be seen from Tables 2 and 4 the thermal efficiencies of the regenerative single-reheat cycles are significantly higher than those for the single-reheat cycle without regeneration. The optimum number of feedwater heaters would have to be determined through a cost-benefit analysis.

Table 4: Mass-Flow and Heat-Transfer Rates through System Components.

Steam Temperature Main / Reheat, °C	**625 / 625**	
Stage	***m*, kg/s**	**Q, MW$_{th}$**
Pumps (total)	N/A	48
Feedwater Heater (HX1)	754	514
Deaerator	754	230
Feedwater Heater (HX2)	1058	433
Reactor	1058	2076
HP Turbine	1058	–388
Reheater	864	440
LP Turbine	754	–812
Condenser	594	–1364
Efficiency with pump work, %	**46.8**	
Efficiency without pump work, %	**47.7**	

The regenerative cycle presented in this section represent a viable basis for a future SCW NPP. The optimum number of regenerative heaters should be investigated – it is likely going to be higher (see Fig. 5) than what was presented herein for illustration of efficiency trends (for details, see Duffey et al., 2008a,b). However, it is clear that the calculated thermal efficiencies with regenerative heating are significantly higher than those of currently operating NPPs. The increase in efficiency is high enough to justify the increased costs associated with the design, construction and operation of this new type SCW NPPs.

1.4. Conclusions

The following conclusions can be made:

1) Four simplified SCW NPP thermodynamic cycles (main "steam" parameters – pressure of 25 MPa and temperature of 625°C; reheat steam parameters – pressure of 2.3 – 7.6 MPa and temperature of 625°C (700°C) and the corresponding arrangements have been investigated in terms of their thermal efficiency, assuming isentropic turbine efficiencies of 88% for all turbine sections and 84% polytropic efficiency for all pumps: (I) cycles without heat regeneration: (1) no-reheat cycle – 40.5% thermal efficiency; (2) single-reheat cycle – 42.6% thermal efficiency; and (3) double-reheat cycle – 43.7% thermal efficiency; and (II) cycles with heat regeneration: (4) single-reheat cycle with three feedwater heaters – 46.8% thermal efficiency. And
2) Based on the abovementioned analysis the single-reheat cycle with heat regeneration and the corresponding arrangement appears to be the most advantageous as a basis for an SCW NPP.

PART 2. SCWRS RANKINE CYCLES

2.1. Review of Existing SC Turbines at Coal-Fired Thermal Power Plants

SC-"steam" turbines of medium and large capacities (450 – 1200 MW_{el}) have been used very successfully at many coal-fired power plants for more than fifty years. Their steam-cycle thermal efficiencies have reached nearly 54%, which is equivalent to a net-plant efficiency of approximately 43 – 48% on an HHV basis.

An analysis of SC-turbine data (see Lecture SC02) showed that:

- The vast majority of the modern and upcoming SC turbines are single-reheat-cycle turbines;
- Major "steam" inlet parameters of these turbines are: The main or primary SC "steam" – P = 24 – 25 MPa and T = 540 – 600°C; and the reheat or secondary subcritical-pressure steam – P = 3 – 5 MPa and T = 540 – 620°C. And
- Only very few double-reheat-cycle turbines were manufactured so far. The market demand for double-reheat turbines disappeared due to economic reasons after the first few units were built.

Based on the analysis of SC coal-fired power plants data the following two major conclusions can be drawn:

1) Direct single-reheat regenerative thermodynamic cycle (Rankine cycle) is the basic cycle for the vast majority of modern SC coal-fired thermal power plants. And
2) Current level of inlet parameters for SC turbines are: The main or primary SC "steam" – P = 25 – 30 MPa and T = 600 – 625°C; and the reheat or secondary steam – P = 3 – 7 MPa and T = 600 – 625°C.

Therefore, the best option for SCWR development is to match at least the current SC turbine parameters. Only in this case, we will have significant savings in development costs, resources and time. However, SCWRs are not SC steam generators and some special requirements will apply.

2.2. Possible Thermodynamic Cycles for SCWRs

In general, we have a solid basis for SCWR development:

- Pressurized Water Reactor's (PWR's) technology (current pressures up to 16 MPa);
- Boiling Water Reactor's (BWR's) once-through or direct cycle;
- Supercritical "steam" turbines and other equipment from coal-fired power plants;
- Materials, which can withstand high pressures and temperatures, and aggressive medium such as supercritical water; And
- Experience of nuclear steam reheat at several BWR NPPs in Russia and USA (Pioro et al., 2010a; Saltanov et al., 2010)

However, it should be admitted that the major problem for the SCWR development is reliability of materials at high pressures and temperatures, aggressive medium such as supercritical water plus high neutron flux. To resolve this problem SCW loops must be designed and built in experimental reactors with neutron-flux levels, thermalhydraulic and other parameters corresponding to that of SCWRs.
In general, the following thermodynamic cycles can be used in SCW NPPs (for more details, see (Duffey et al., 2008c; Naidin et al., 2009; Pioro et al., 2010b and 2008):

1) Direct single-reheat regenerative thermodynamic cycle (Rankine cycle) (Fig. 4), which is the basic cycle for the vast majority of modern SC thermal coal-fired power plants.
2) In-direct single-reheat regenerative thermodynamic cycle (Fig. 5).
3) Direct no-reheat regenerative thermodynamic cycle (Fig. 6).
4) In-direct no-reheat regenerative thermodynamic cycle (Fig. 7). And
5) Dual regenerative thermodynamic cycles (Figs. 8 and 9).

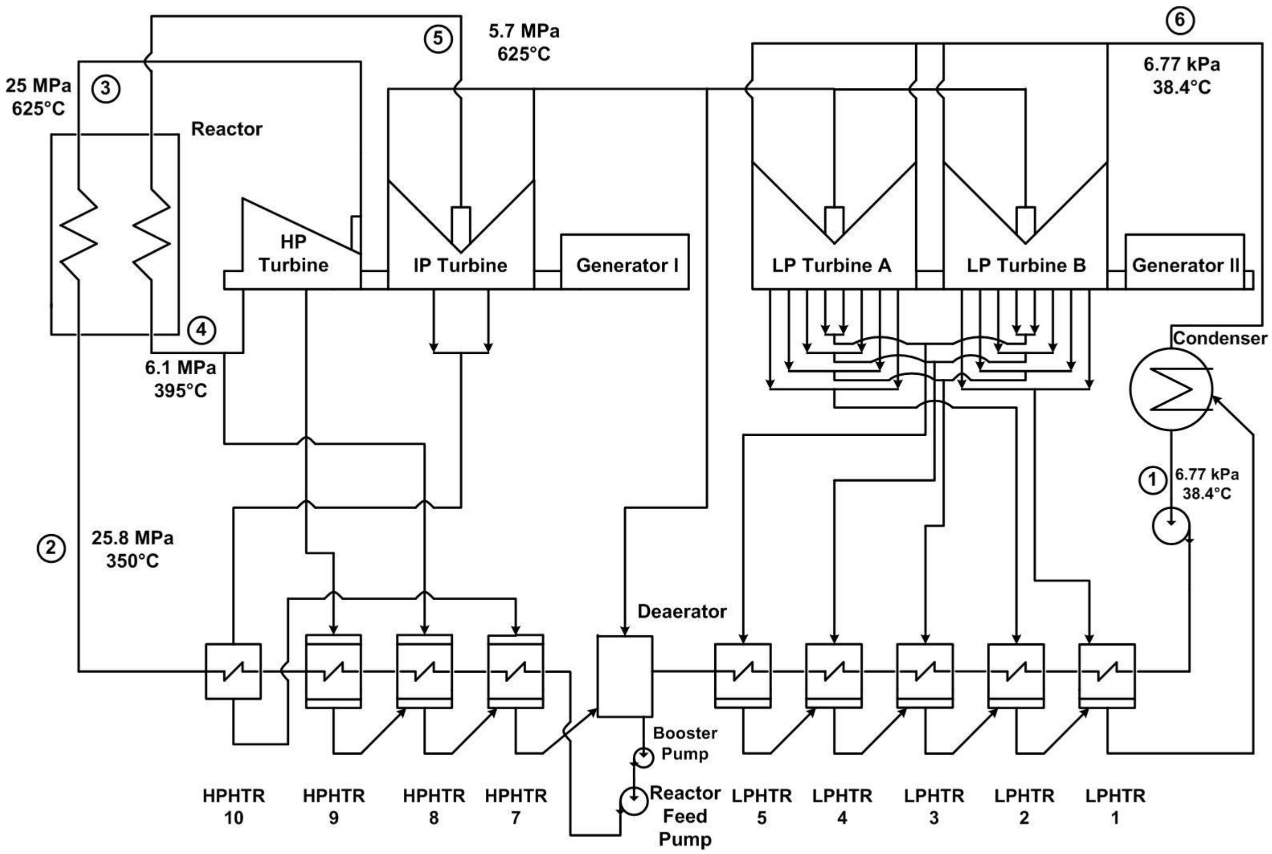


Figure 4. Schematic of single-reheat regenerative thermodynamic cycle for 1200 MW_{el} PT SCW NPP (Naidin et al., 2009).

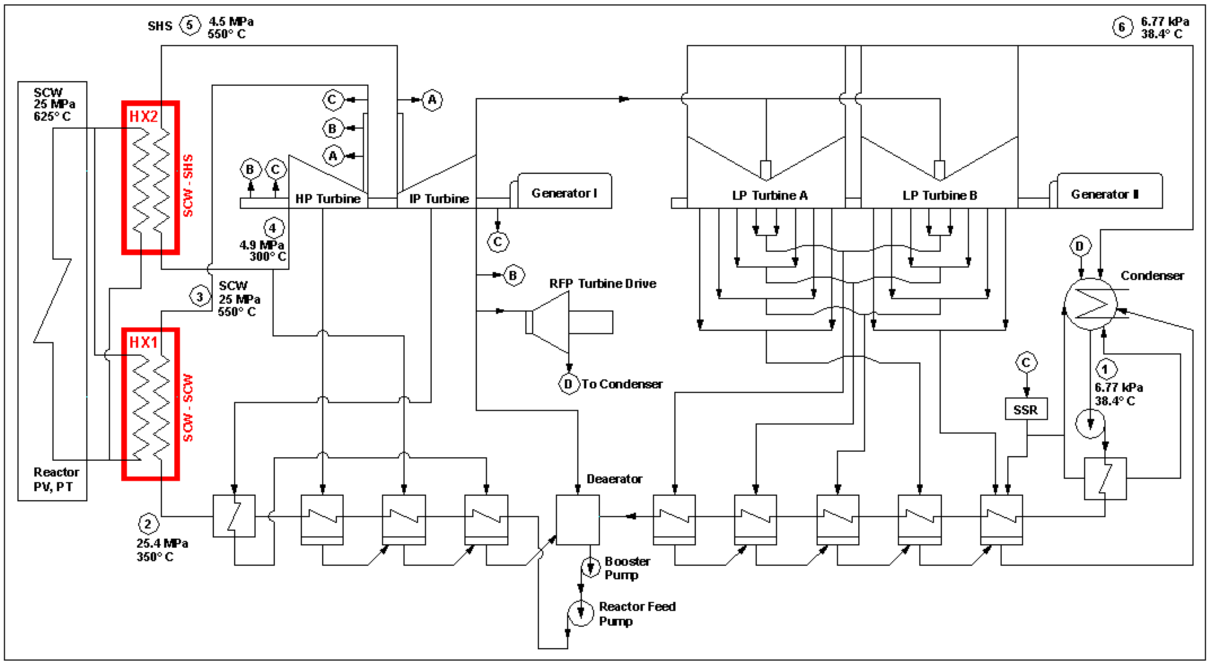


Figure 5. Schematic of in-direct single-reheat regenerative thermodynamic cycle for 1200 MW$_{el}$ PV or PT SCW NPP (Thind et al., 2010).

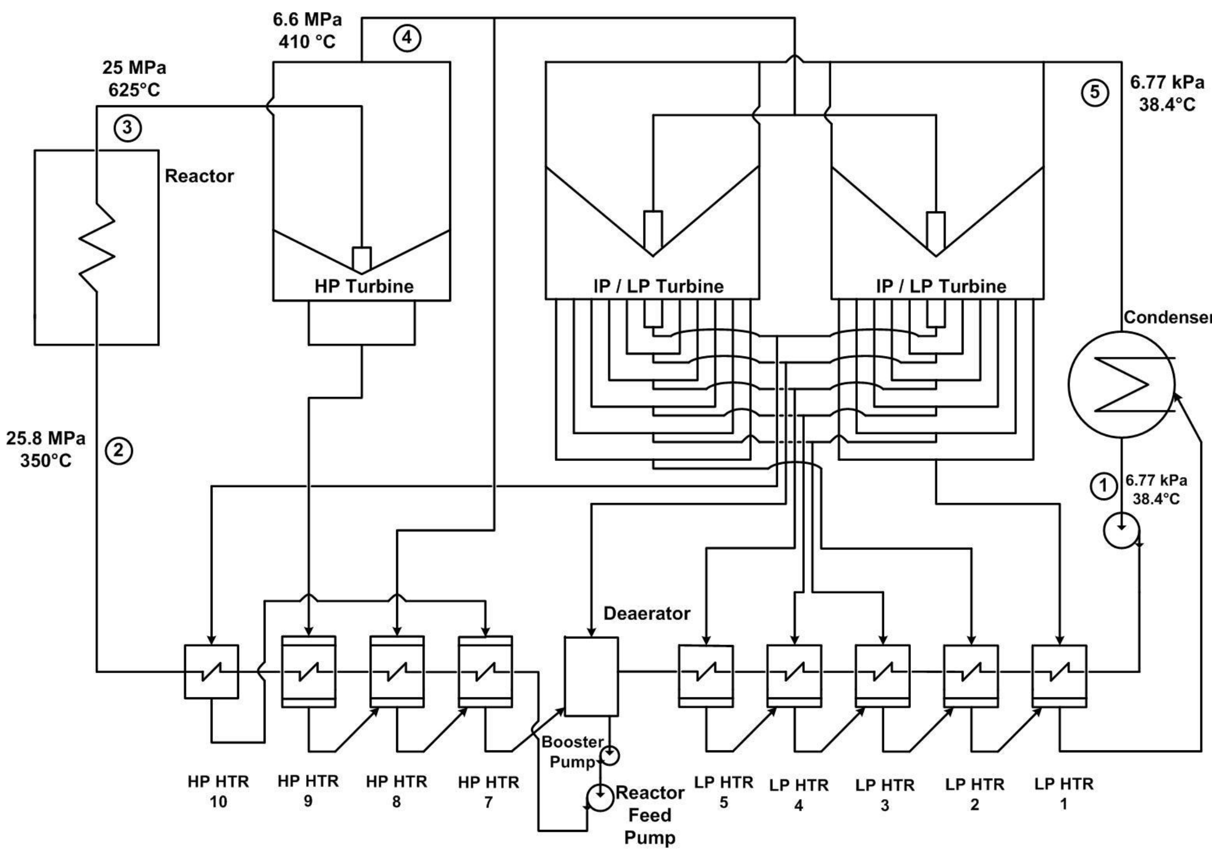


Figure 6. Schematic of direct no-reheat regenerative thermodynamic cycle for 1200 MW$_{el}$ PV or PT SCW NPP (Naidin et al., 2009).

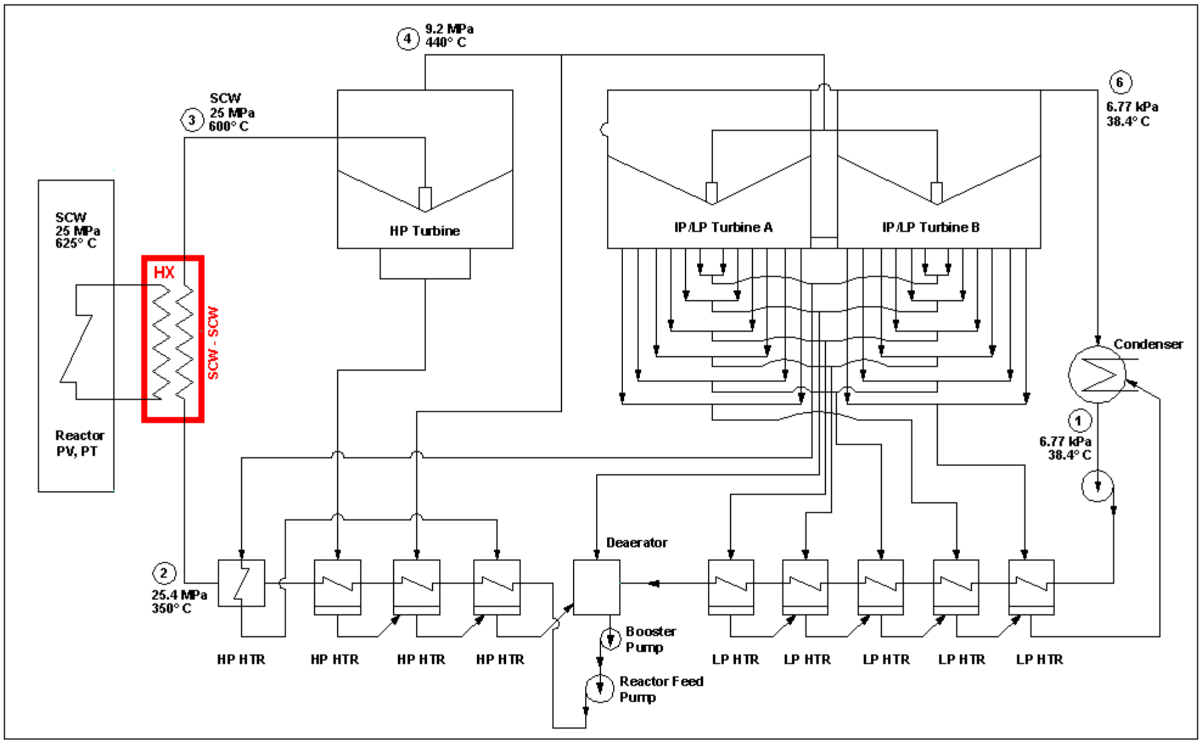


Figure 7. Schematic of in-direct no-reheat regenerative thermodynamic cycle for 1200 MW$_{el}$ PV or PT SCW NPP (Thind et al., 2010).

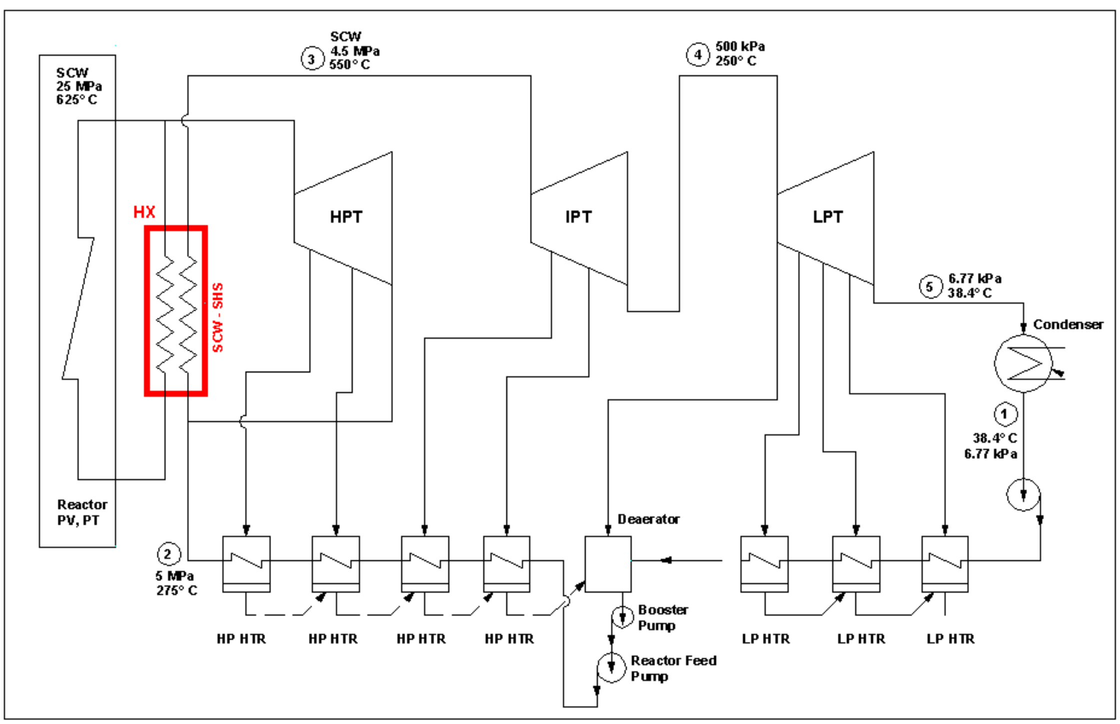


Figure 8. Schematic of dual no-reheat primary (SCW) loop and single-reheat secondary (superheated steam) loop regenerative thermodynamic cycle for 1200 MW$_{el}$ PV or PT SCW NPP (Thind et al., 2010).

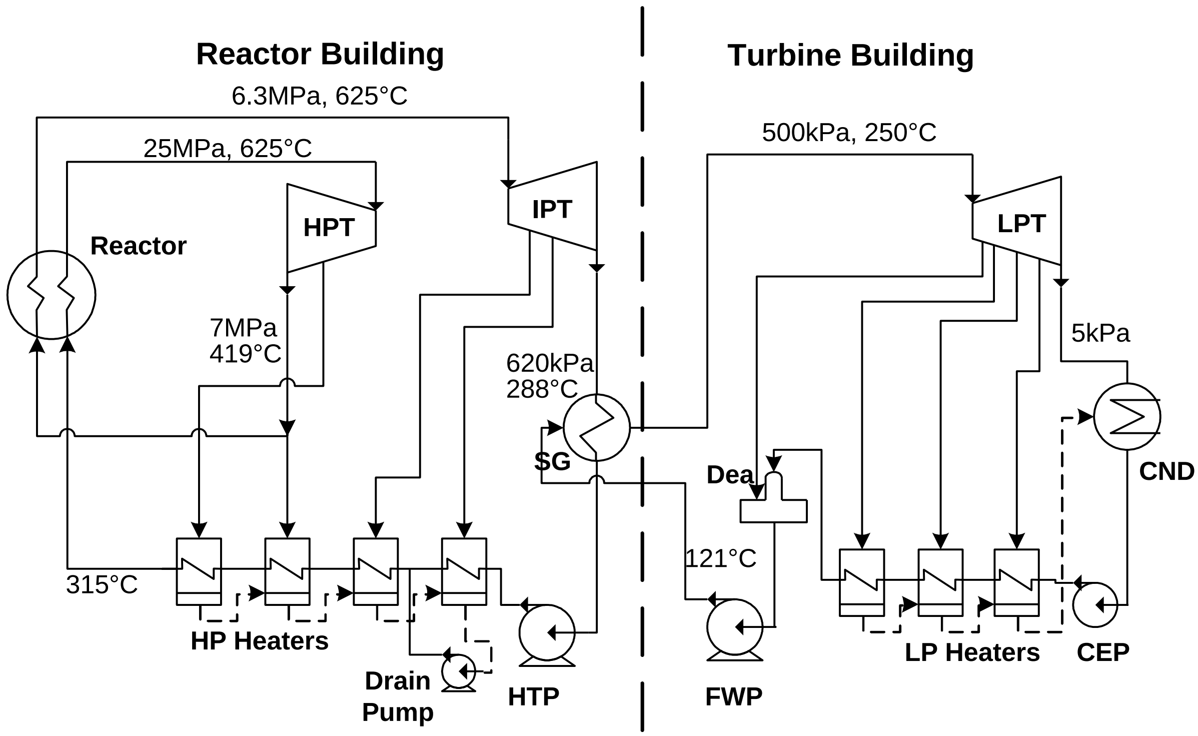


Figure 9. Schematic of dual single-reheat regenerative thermodynamic cycle for 1200 MW_{el} PT SCW NPP (Duffey et al., 2008c): High-pressure units located in Reactor Building for increased safety.

In the direct cycle, SC "steam" from an SCWR is fed directly to an SC turbine. This concept eliminates the need for complex and expensive equipment such as steam generators (heat exchangers). From a thermodynamic perspective, this allows for high steam pressures and temperatures, and results in the highest cycle thermal efficiency for the given parameters. The direct single-reheat cycle with current SC "steam" parameters will have the gross thermal efficiency of about 52% and no-reheat cycle – about 51% (Naidin et al., 2009). However, the direct single-reheat cycle is easier to implement in PT SCWR and might be impossible to implement in PV SCWRs. The direct no-reheat cycle can be implemented in both types of SCWRs.

The single-reheat cycle is widely used in thermal power industry, but we have not found any information on thermal power plants operating on the no-reheat cycle. The major technical challenge for the no-reheat cycle is relatively high moisture content at the outlet of the LP turbine (about 19%). However, the moisture can be reduced by implementing contoured channels in the inner casing for draining the water and moisture removal stages.

The indirect and dual cycles utilize heat exchangers (steam generators) to transfer heat from the reactor coolant to a turbine. The indirect cycle has the safety benefit of containing the potential radioactive particles inside the primary coolant. Also, this cycle arrangement prevents deposition of various substances from the reactor coolant on turbine blades. However, the heat-transfer process through heat exchangers reduces the maximum temperature in the secondary-loop coolant at least by 25 – 75°C, thus lowering the efficiency of the cycle. Also, heat exchangers (steam generators) can be quite large units with about 200 thousand square meters of heat transfer surfaces (Duffey et al., 2008c; Thind et al., 2010).

From the SCWR design point of view, the following can be concluded:

- The single-reheat cycle has the advantage of higher thermal efficiency (compared to that of the no-reheat cycle) and reduced development costs due to a wide variety of single-reheat SC turbines manufactured by companies worldwide. The major disadvantage is the increased design complexity associated with the introduction of steam-reheat (SHR) channels to the reactor core (Fig. 10). And
- The no-reheat cycle offers a simplified SCW NPP layout, contributing to lower capital costs. However, the efficiency of this cycle is lower and no-reheat SC turbine has been possibly developed.

As such, configurations based on the single-reheat and no-reheat cycles were chosen for the analysis in this lecture.

2.3. Thermodynamic Analysis of Direct Cycles for SCWRs

Table 5 lists selected parameters of proposed SCW NPP direct regenerative cycles, and Table 6 - selected parameters of proposed SCWR fuel channels.

Table 5. Selected parameters of proposed SCW NPP direct cycles (Naidin et al., 2009).

Parameters	Unit	Description / Value	Description / Value
Cycle type	–	Direct Single-Reheat (Fig. 45)	Direct No-Reheat (Fig. 67)
Reactor type	–	Pressure Tube	
Reactor spectrum	–	Thermal	
Fuel	–	UO_2 (ThO_2)	
Cladding material	–	Inconel or Stainless steel	
Reactor coolant	–	H_2O	
Moderator	–	D_2O	
Power Thermal	MW_{th}	2300	2340
Power Electrical	MW_{el}	1200	1200
Thermal Efficiency	%	52	51
Pressure of SCW at inlet	MPa	25.8	25.8
Pressure of SCW at outlet (estimated)	MPa	25	25
T_{in} coolant (SCW)	°C	350	350
T_{out} coolant (SCW)	°C	625	625
Pressure of SHS at inlet	MPa	6.1	–
Pressure of SHS at outlet (estimated)	MPa	5.7	–
T_{in} coolant (SHS)	°C	400	–
T_{out} coolant (SHS)	°C	625	–
Power thermal SCW channels	MW_{th}	1870	2340
Power thermal SRH channels	MW_{th}	430	–
Power thermal / SCW channel	MW_{th}	8.5	8.5
Power thermal / SRH channel	MW_{th}	5.5	–

Parameters	Unit	Description / Value	Description / Value
# of fuel channels (total)	–	300	270
# of SCW channels	–	220	270
# of SRH channels	–	80	–
Total flow rate of SCW	kg/s	960	1190
Total flow rate of SHS	kg/s	780	–
Flow rate / SCW channel	kg/s	4.37	4.37
Flow rate / SRH channel	kg/s	10	–

Table 6. Selected parameters of proposed SCWR fuel channels (Naidin et al., 2009).

Parameters	Unit	Description / Value		
T_{max} cladding (design value)	°C	850		
T_{max} fuel centerline (industry accepted limit)	°C	1850		
Heated fuel-channel length	m	5.772		
# of bundles / fuel channel	–	12		
# of fuel rods per bundle	–	43		
Bundle type (Leung, 2008) (Fig. 11)	–	CANFLEX	Variant-18	Variant-20
# of heated fuel rods	–	43	42	42
# of unheated* fuel rods	–	–	1	1
Diameter of heated fuel rods (# of rods)	mm	11.5 (35) & 13.5 (8)	11.5	11.5
Diameter of unheated fuel rod	mm	–	18	20
D_{hy} of fuel channel	mm	7.52	7.98	7.83
D_h of fuel channel	mm	9.04	9.98	9.83
Heated area of fuel channel	m^2	9.26	8.76	8.76
Flow area of fuel channel	mm^2	3625	3788	3729
Pressure tube inner diameter	mm	103.45		
Average parameters of fuel channels in single-reheat (Fig. 4) and no-reheat (Fig. 6) options				
Heat flux in SCW channel (both cycles)	kW/m^2	918	970	970
Heat flux in SRH channel (single-reheat cycle)	kW/m^2	594	628	628
Mass flux in SCW channel (both cycles)	kg/m^2s	1206	1154	1172
Mass flux in SRH channel (single-reheat cycle)	kg/m^2s	2759	2640	2682

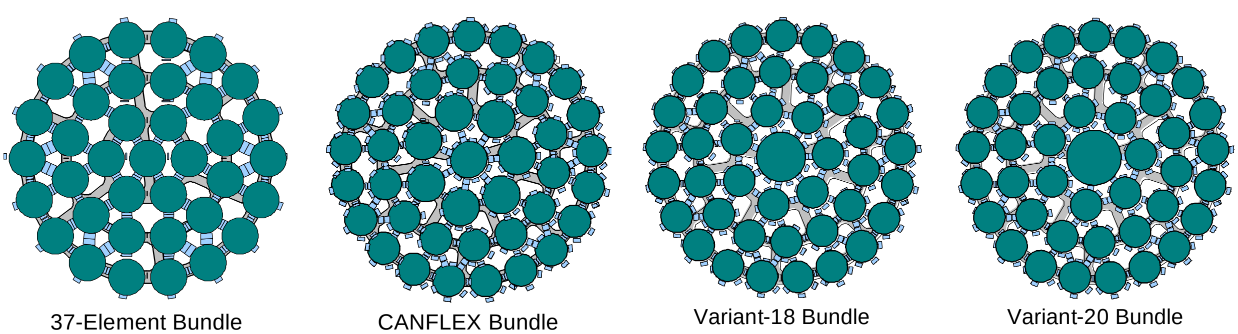


Figure 11. CANDU-reactor fuel bundles: 37 elements - current design and 43-elements (the rest) - new designs (Leung, 2008).

PART 3. CO-GENERATION OF HYDROGEN

Due to the well-known contribution of greenhouse gas emissions generated by fossil fuels to global warming, worldwide research is being conducted to identify a clean energy carrier. As such, hydrogen was identified as one of the promising options. However, most of the hydrogen supply currently available is obtained from fossil fuels through reforming processes, which release greenhouse gases.

Hydrogen is needed in large quantities by many industrial sectors, i.e., Canadian oil sands, petroleum products, agriculture, and transportation. Thus, a technology suitable for large-scale sustainable production of hydrogen needs to be developed and implemented. Thermo-chemical water splitting is one of the most promising technologies for hydrogen generation without the negative consequences of pollutants. By using intermediate compounds, a series of chemical and physical processes decompose water into its two constituents. Thermo-chemical hydrogen production is also much more efficient than other methods, such as electrolysis, because the heat is used directly to produce hydrogen, rather than being converted first to electrical energy.
Although over about 200 thermo-chemical cycles have been identified (Naterer et al., 2010, 2009), proof-of-principle demonstrations have been completed only for a few of them. However, most of these processes use process heat above 800°C, thus requiring very high temperatures that are not currently available in nuclear or thermal power plants. The copper-chlorine (Cu-Cl) cycle is the only demonstrated cycle functioning at a lower temperature of approximately 500°C, which makes it suitable for linkage with a SCW NPP cycle. The relatively lower operating temperature can also lead to a reduction in material and maintenance costs.

Table 7: Chemical reaction steps and basic parameters of the copper-chlorine cycle.

<table>
<tr><th>Step</th><th>Reaction</th><th>Temperature Range (°C)</th><th colspan="2">Feed/Output</th></tr>
<tr><td>1</td><td>$2Cu(s) + 2HCl(g) \rightarrow CuCl(l) + H_2(g)$</td><td>430 – 475</td><td>Feed:
Output:</td><td>Electrolytic Cu + dry HCl + Q
H_2 + CuCl(l) salt</td></tr>
<tr><td>2</td><td>$2CuCl(s) \rightarrow 2CuCl(aq) \rightarrow CuCl_2(aq) + Cu(s)$</td><td>Ambient (electrolysis)</td><td>Feed:
Output:</td><td>Powder/granular CuCl and HCl + V
Cu and slurry containing HCl and $CuCl_2$</td></tr>
<tr><td>3</td><td>$CuCl_2(aq) \rightarrow CuCl_2(s)$</td><td><100</td><td>Feed:
Output:</td><td>Slurry containing HCl and $CuCl_2$ + Q
Powder/granular $CuCl_2$ + H_2O/HCl vapors</td></tr>
<tr><td>4</td><td>$2CuCl_2(s) + H_2O(g) \rightarrow CuO{*}CuCl_2(s) + 2HCl(g)$</td><td>400</td><td>Feed:
Output:</td><td>Powder/granular $CuCl_2$ + $H_2O(g)$ + Q
Powder/granular $CuO{*}CuCl_2$ + 2HCl (g)</td></tr>
<tr><td>5</td><td>$CuO{*}CuCl_2(s) \rightarrow 2CuCl(l) + 1/2O_2(g)$</td><td>500</td><td>Feed:
Output:</td><td>Powder/granular $CuO{*}CuCl_2(s)$ + Q
Molten CuCl salt + oxygen</td></tr>
<tr><td colspan="5">Q - thermal energy and V - electrical energy</td></tr>
</table>

Currently, UOIT (University of Ontario Institute of Technology) in collaboration with AECL and other partners are developing the Cu-Cl cycle with a maximum temperature in the cycle of up to 500°C (for details, see Table 7 and Naterer et al. (2010), (2009)). Therefore, using the high-temperature heat from a SCWR to heat water and endothermic reactors in the hydrogen-production loop is a viable option. Heat exchangers of a recuperator-type have to be used for this purpose.

As mentioned in previous sections, due to the high temperature coolant at the reactor outlet, the SCW NPP cycles are suitable for hydrogen co-generation. Figure 12 shows the heat extraction points for the no-reheat configuration. As such, the first extraction point would be that located directly on the main SC "steam" lines coming from the reactor core. The temperature and pressure of the liquid are 625°C and 25 MPa, respectively. The SC "steam" is used to heat the water and endothermic reactors in the hydrogen loop through a heat exchanger, as shown in Figure 12. The cooler fluid at the outlet of the heat exchanger is returned at a suitable point along the feedwater heating system.

Alternatively, another source of high-quality heat is identified to be the superheated steam coming from the exhaust of the HP Turbine. The temperature and pressure of the fluid are 460°C and 9.2 MPa, respectively. Similar to the above discussion, this steam can be used to heat the water in the hydrogen loop, using a heat exchanger.
The third possible option is to divert steam from the HP Turbine extraction point. According to Figure 12, this fluid is heating the feedwater passing through HP HTR9. However, a portion of this high-quality steam (approximately 12.5 MPa and 500°C) can be used to heat up the process water in the hydrogen loop.

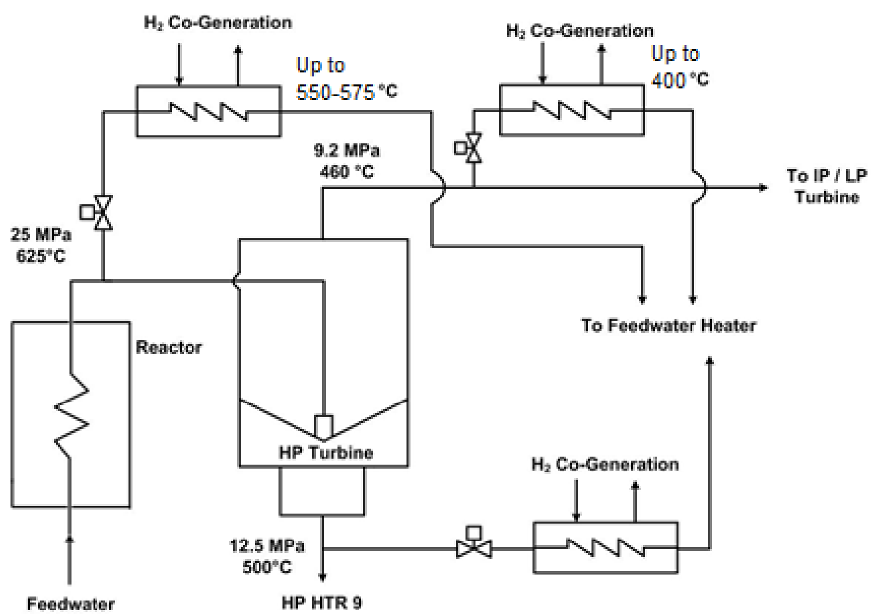


Figure 12: Heat-extraction points for H_2 co-generation associated with no-reheat SCW NPP.

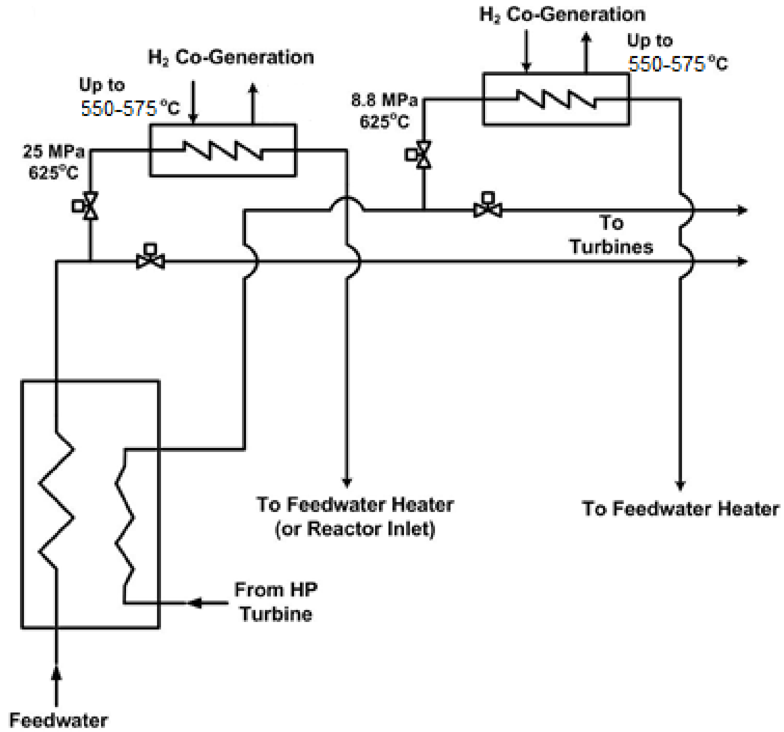


Figure 13: Heat-extraction points for H_2 co-generation associated with single-reheat SCW NPP.

CONCLUSIONS

The following conclusions can be made:

1. The vast majority of the modern SC turbines are single-reheat-cycle turbines. Just a few double-reheat-cycle SC turbines have been manufactured and put into operation. However, despite their efficiency benefit double-reheat-turbines have not been considered economical.
2. Major inlet parameters of the current and upcoming single-reheat-cycle SC turbines are: the main or primary SC "steam" – pressure of 25 – 30 MPa and temperature of 600 – 625°C; and the reheat or secondary subcritical-pressure steam – P = 3 – 7 MPa and T = 600 – 625°C.
3. In order to maximize the thermal-cycle efficiency of the SCW NPPs it would be beneficial to include nuclear steam reheat. Advantages of a single-reheat cycle in application to SCW NPPs are:
   a. High thermal efficiency (about 50%), which is the current level for SC thermal power plants and close to the maximum thermal efficiency achieved in the power industry at combined-cycle power plants (up to 55%).
   b. High reliability through proven state-of-the-art turbine technology; and
   c. Reduced development costs accounting on wide variety of SC turbines manufactured by companies worldwide.
4. The major disadvantage of a single-reheat cycle implementation in SCW NPPs is the requirement for significant changes to the reactor-core design due to addition of the nuclear steam-reheat channels at subcritical pressures.

5. Based on the abovementioned analysis, the direct or in-direct single-reheat cycles with heat regeneration and the corresponding arrangement appear to be the most advantageous as a basis for a SCW NPP. However, other cycles, for example, dual cycles might be considered.

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NOMENCLATURE

$\bar{c}_p$	average specific heat, J/kg·K, $\left(\frac{H_w - H_b}{T_w - T_b}\right)$
D	inside diameter, m
G	mass flux, kg/m^{2}s
H	enthalpy, J/kg
h	heat transfer coefficient, W/m^{2}K
k	thermal conductivity, W/m·K
m	mass-flow rate, kg/s
P	pressure, Pa
Q	heat-transfer rate, W
T	temperature, ºC

Greek letters

μ	dynamic viscosity, Pa·s
ρ	density, kg/m^3

Dimensionless numbers

Nu	Nusselt number $\left(\frac{h \cdot D}{k}\right)$
$\overline{\mathbf{Pr}}$	average Prandtl number $\left(\frac{\mu \cdot \bar{c}_p}{k}\right)$
Re	Reynolds number $\left(\frac{G \cdot D}{\mu}\right)$

Subscripts

b	bulk
ch	channel
el	electrical
h	heated
hy	hydraulic
in	inlet

out	outlet
pc	pseudocritical
max	maximum
th	thermal
w	wall

Abbreviations

AECL	Atomic Energy Canada Limited
AHFP	Axial Heat Flux Profile
BWR	Boiling Water reactor
CANDU	CANada Deuterium Uranium (reactor)
CANFLEX	CANDU FLEXible (fuelling)
CEP	Condensate Extraction Pump
CL	CenterLine
CND	Condenser
Dea	Deaerator
FWP	Feed Water Pump
HF	Heat Flux
HHV	Higher-Heating Value
HP	High Pressure
HPT	High Pressure Turbine
HTC	Heat Transfer Coefficient
HTR	HeaTeR
HWR	Heavy Water Reactor
HX	Heat eXchanger
ID	Inside Diameter
IP	Intermediate Pressure (turbine)
IPT	Intermediate Pressure Turbine
KP-SKD	Channel Reactor of Supercritical Pressure (in Russian abbreviations)
LHV	Lower-Heating Value
LP	Low Pressure
LPT	Low Pressure Turbine
LWR	Light Water Reactor
Mix Ch	Mixing Chamber (deaerator)
NA	Not Available
NPP	Nuclear Power Plant
OD	Outside Dimater
PT	Pressure Tube (reactor)
PV	Pressure Vessel (reactor)
PWR	Pressurized Water Reactor
RFP	Reactor Feed Pump
RMS	Root Mean Square
SC	SuperCritical
SCW	SuperCritical Water
SCWR	SuperCritical Water Reactor
SG	Steam Generator
SH	SHeath
SHS	SuperHeated Steam
SRH	Steam ReHeat
UOIT	University of Ontario Institute of Technology

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