Analysis of HRSG Cold-end Design Options for Best Performance and Reliability

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Executive Summary

Maximizing the overall fuel efficiency of modern combined cycle power plants requires fine tuning all of the main components in concert to reach a common optimum. Experience shows that there exists a large potential for improvement in the so-called 'cold-end' of the HRSG comprised of the condensate preheater section, the deaerator, and the first economizer sections of the steam generator. Since these sections of the HRSG may also be subject to severe problems due to accelerated corrosion or flow instabilities if operated under extreme conditions, caution must be applied when looking for the optimal configuration.

The paper studies and compares various commercially available configurations, designs and temperature control strategies of the HRSG cold-end. The study focuses on different configuration options to prevent potential problems with the sulphuric acid dew point and steaming in the low temperature economizer. It quantifies the impact of different fuel gas compositions, condensate inlet temperatures, duct firing loads and modes of operation for the deaerator on these key operational constraints. The study was performed using the EBSILON Professional heat and mass balance software to construct detailed thermodynamic models for both, design and off-design conditions in a typical 3-pressure level reheat HRSG. The study incorporates the extensive experience the authors have in power plant design, optimization, simulation and operations as well as available documentation, such as books, publications and patents by HRSG vendors.

Introduction

An HRSG plays an important role in combined cycles, by utilizing the exhaust energy from the gas turbine to raise steam, which is then supplied to the steam turbine. The HRSG's performance is therefore an important factor in the overall performance of the plant. It needs to contribute to the overall plant efficiency while keeping in mind the reliability, flexibility, availability and maintainability of the plant. This paper discusses the impact of the cold-end section of the HRSG on those factors. The cold-end of an HRSG of is sometimes referred to as the "LP section" or "back-end" as well. Various possible design configurations were considered for this paper and their relative performance is compared.

Base Model

The layout of the combined cycle power plant was chosen to be 2 x 2 x 1 (2 gas turbines and HRSGs, 1 steam turbine) with GE 9F.04 gas turbine type that was modelled with VTU's Gas Turbine Library software, developed for the EBSILON Professional power plant simulation program (1). Figure 1 below shows the overall simulation model for one of the variants analyzed in this study.

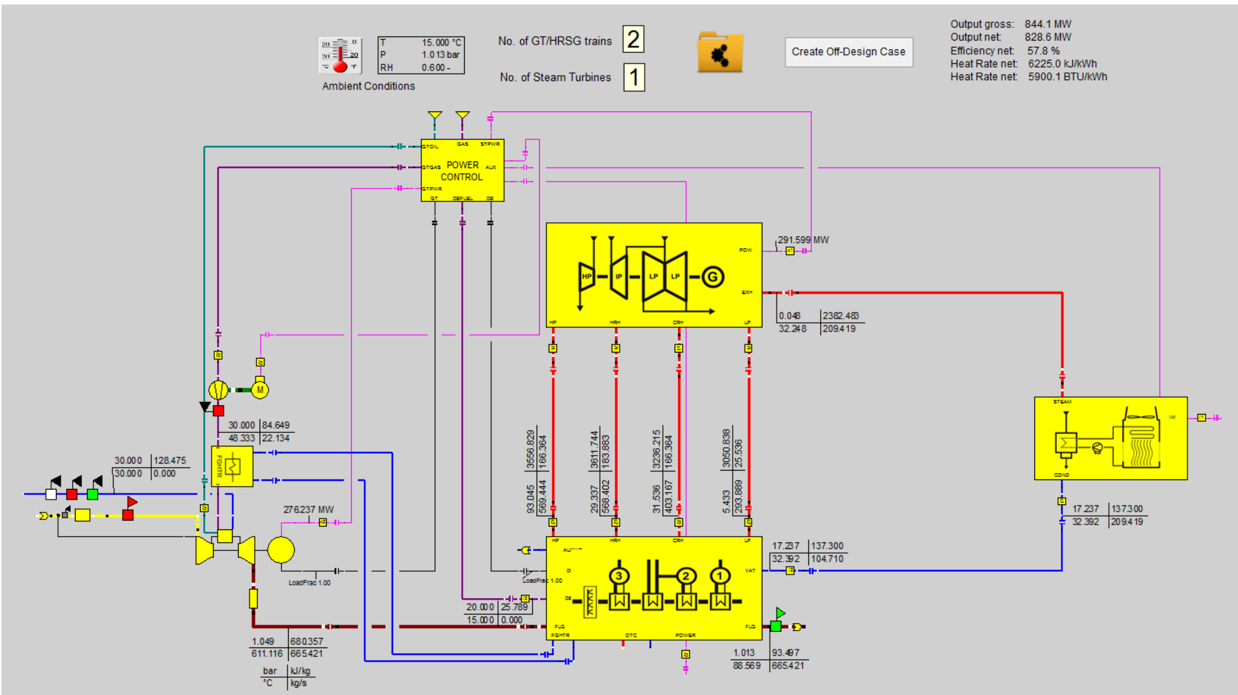

Figure 1: Overall plant simulation model

The HRSG configuration is horizontal with natural recirculation in the evaporators. The condenser uses a mechanical draft wet cooling tower to remove the heat of condensation. Since it was assumed that the two trains of the plant are always operated at the same load level if both are running, the model was simplified by using stream multipliers/dividers on the connecting streams between the GT/HRSG trains and the steam turbine and cooling system.

All EBSILON models used a base model that was derived from design information of a real plant, but the low pressure section (feed water pre-heaters, low pressure (LP) evaporators, LP economizers and a deaerator) was modified for the various cases that were studied. Table 1 below summarizes the key design parameters of the combined cycle power plant, and Figure 2 shows the details of the HRSG sub-model included in the overall plant model. The configuration of the HRSG from the leading HP superheater no. 4 to the LP superheater following the parallel IP/HP economizers was identical in all of the variants of the study. Only the low pressure section high-lighted with white background color in Figure 2 was exchanged with different cold-end configurations.

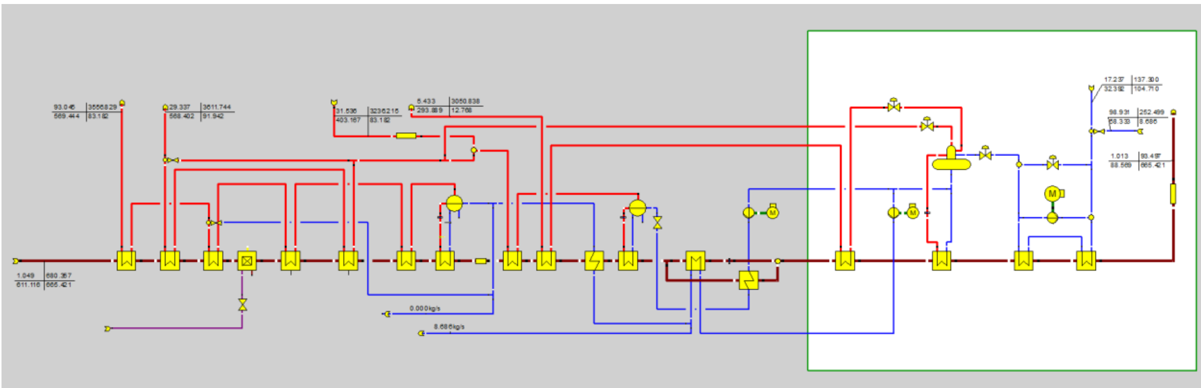

Figure 2: Heat balance model of the HRSG, cold-end section high-lighted with white background

HP Steam Pressure	93 bara	1350 psia
HP Steam Temperature	569.4 °C	1057 F
Hot Reheat Steam Pressure	29.3 bara	425.5 psia
Hot Reheat Steam Temperature	568.4 °C	1055F
IP Steam Pressure	30.5 bara	443 psia
IP Steam Temperature	307.2 °C	585 F
Duct Burner max. Temperature	715 °C	1319 F
Condenser Pressure	0.0483 bara	1.43 in. Hg
LP Steampressure	5.4 bara	78.8 psia
LP Steam Temperature	293.9 °C	561 F
HP Evaporator Pinch Point	12.5 °C	22.5 F
IP Evaporator Pinch Point	21 °C	37.8 F
LP Evaporator Pinch Point	12.5 °C	22.5 F
Preheater to Deaerator approach	10 °C	18 F
Design Ambient conditions	ISO (15 °C, 60% RH, 1.013 bara)	ISO (59 F, 60% RH, 14.696 psia)

Table 1: Key design parameters of the base model

For the fuel gas a typical Thai gas was chosen (74.67 vol% CH_4, LHV 32364 kJ/kg), and the fuel oil composition was set as follows: C = 86.33 H = 13.07 O = 0.004 S = 0.002.

Cold-End Configurations

Below is a description of the five different configurations of the HRSG back-end that were considered:

1. Low Temperature Heat Exchanger "LTE"

This LP section contains an LP evaporator with integral deaerator, and the preheater is a low temperature economizer (LTE) with bypass and recirculation. Its design was derived from figure 1 in the referred US patent (2).

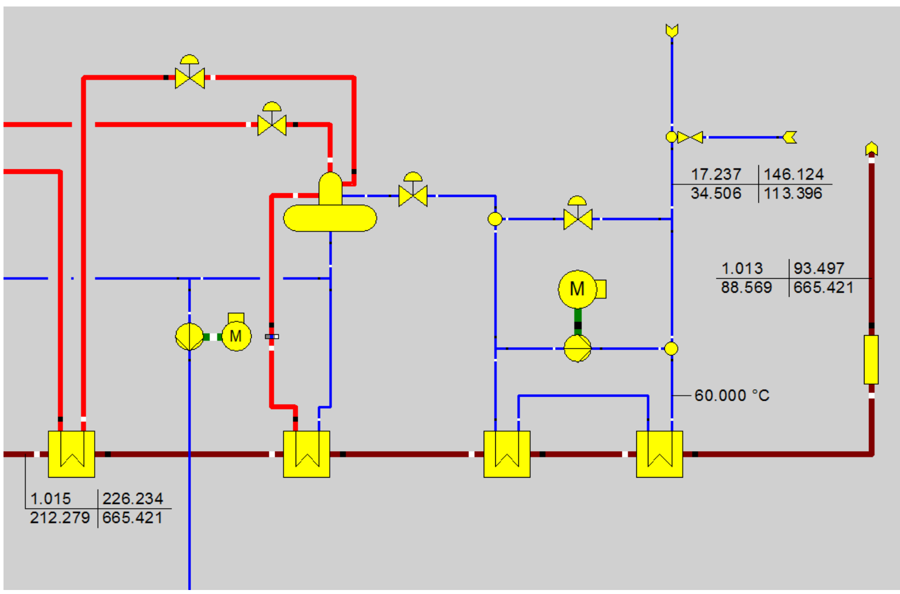

Figure 3: Cold-end configuration with low temperature heat exchanger ("LTE")

This configuration uses a low temperature economizer, which preheats water prior to entering an integral deaerator. Some of the warm water exiting the heat exchanger can be recirculated to preheat the incoming water to maintain the minimum required water inlet temperature. There is a maximum recirculation rate, limited mostly by the recirculation pump and heat exchangers. Condensate pump exit pressure must be relatively high compared to the other configurations to support the fluctuations in flow rate (and corresponding pressure drops) inside the preheater for the different operating conditions. This configuration is also equipped with a bypass around the preheater, in case the maximum recirculation flow rate is reached and the minimum temperature is still not achieved. This bypass will also be in operation in operating cases where the (integral) deaerator approach is too low to have proper control.

2. External Product/Feed Heat Exchanger ("PFHX")

This LP section contains an LP evaporator with integral deaerator, and the preheater is product/feed heat exchanger followed by a preheater. Its design was derived from figure 2 in the referred US patent (2).

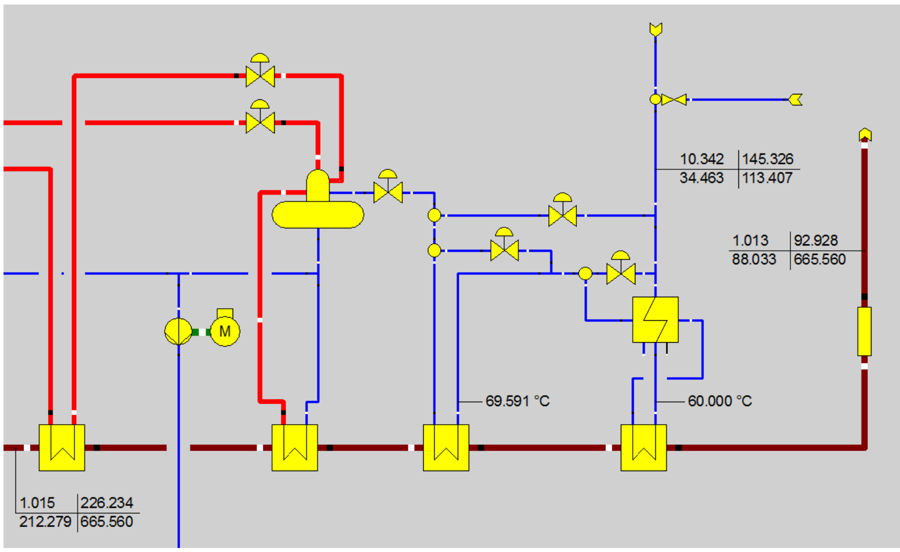

Figure 4: Cold-end configuration with external product/feed heat exchanger ("PFHX")

The cold condensate enters the external product/feed heat exchanger at a rather low temperature that results from mixing the main condensate flow at or slightly below the condensation temperature and the return flow of HP feedwater from the fuel preheater. As this heat exchanger is a water/water heat exchanger external to the HRSG, this fact causes no problems. In order to produce the required minimum inlet temperature into the preheater at the cold end of the HRSG, the exit stream of this preheater is returned to the product/feed heat exchanger. A bypass upstream of the product/feed heat exchanger allows for controlling the preheater inlet temperature. This bypass stream is mixed with the hot-side outlet of the product feed heat exchanger. If the resulting mixing temperature is above the lower limit for water temperature entering the HRSG, the resulting flow passes through another preheater section. In case the inlet temperature is too low, the second preheater may be bypassed completely. This design doesn't require an additional pump, as both bypass lines operate with positive pressure differences.

3. External Deaerator and Product/Feed Heat Exchanger "EXTDA + PFHX"

This design contains an LP evaporator with separate deaerator, and the preheating section consists of an external product/feed heat exchanger followed by the economizers at the three HRSG pressure levels.

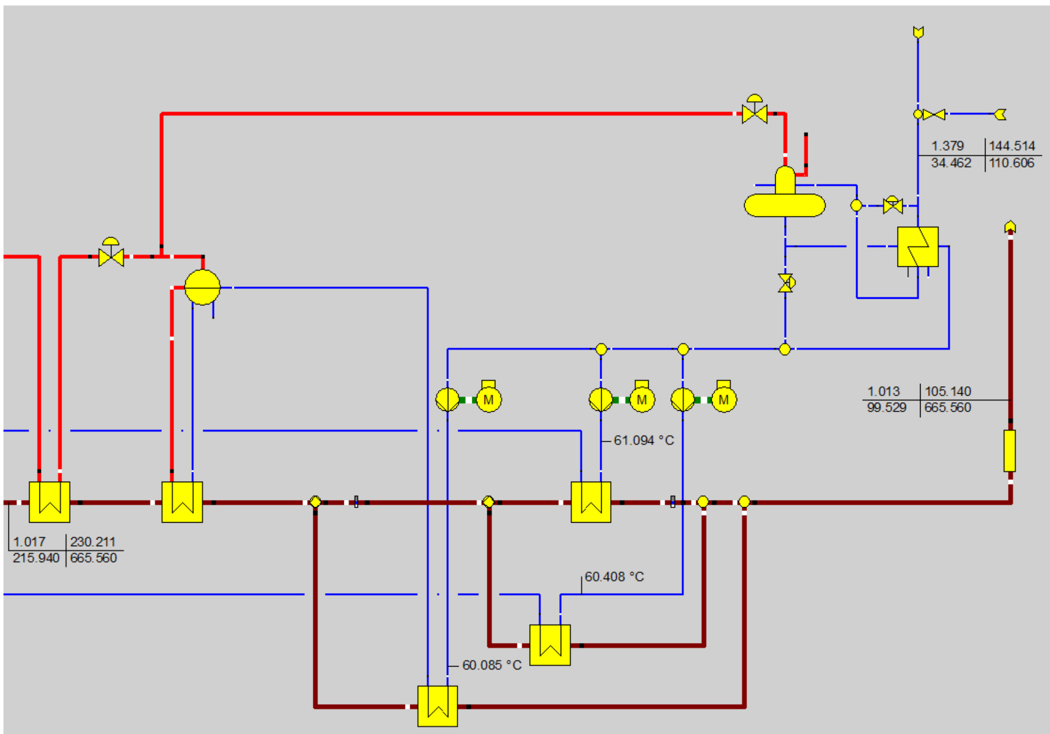

Figure 5: Cold-end configuration with external deaerator and product/feed heat exchanger ("EXTDA + PFHX")

In this configuration, the external product feed heat exchanger is linked to the deaerator operating just above atmospheric pressure. The controlling element for establishing the required water inlet temperature into the HRSG is the partial bypass of the hot side of the P/F heat exchanger which raises the economizer inlet temperature with deaerator outlet water at the expense of lowering the deaerator inlet temperature. In order to maintain a required level of sub-cooling of the inlet stream into the deaerator, the product/feed heat exchanger may also be bypassed on the cold side. If required, the deaerator pressure could be raised to increase the HRSG inlet temperature further (such as in case of operation with fuel oil). The LP, IP and HP circuits are separated once leaving the deaerator. This configuration also doesn't require an additional recirculation pump.

4. Product/Feed Heat Exchanger with two parallel LT Economizers " PFHX + 2LTE"

This LP section design consists of two preheaters located side by side in the flue gas stream, with feed water flowing through the external product/feed heat exchanger prior to entering the first preheater. Two further preheaters follow which may be used, if the respective inlet temperature limit is not violated. This design was derived from figure 5 in the referred US patent (2).

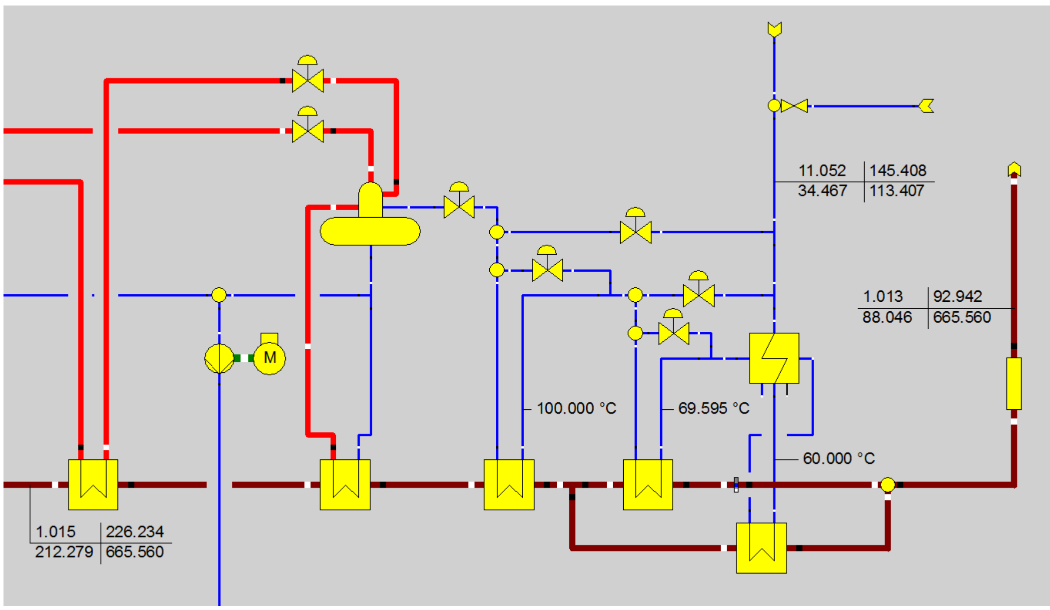

Figure 6: Cold-end configuration with product/feed heat exchanger and two parallel LT Economizers ("PFHX + 2LTE")

Under off-design conditions a bypass around the external heat exchanger is used in case the inlet temperature to the HRSG would drop too low (thereby preheating less water in the same heat exchanger, achieving a higher temperature). If that is insufficient or if that causes the inlet temperature to the second preheater stage to drop too low, additional bypass of the final stage of the preheater might be required. This bypass will also be in operation in operating cases where the integral deaerator approach is too low to have accurate control of the deaerator saturation conditions. No additional pump is required.

5. External deaerator with Recirculation" RECIRC EXTDA"

This LP section comprises of an external (vacuum) deaerator followed by a recirculation heater for deaerator water, followed by various economizers.

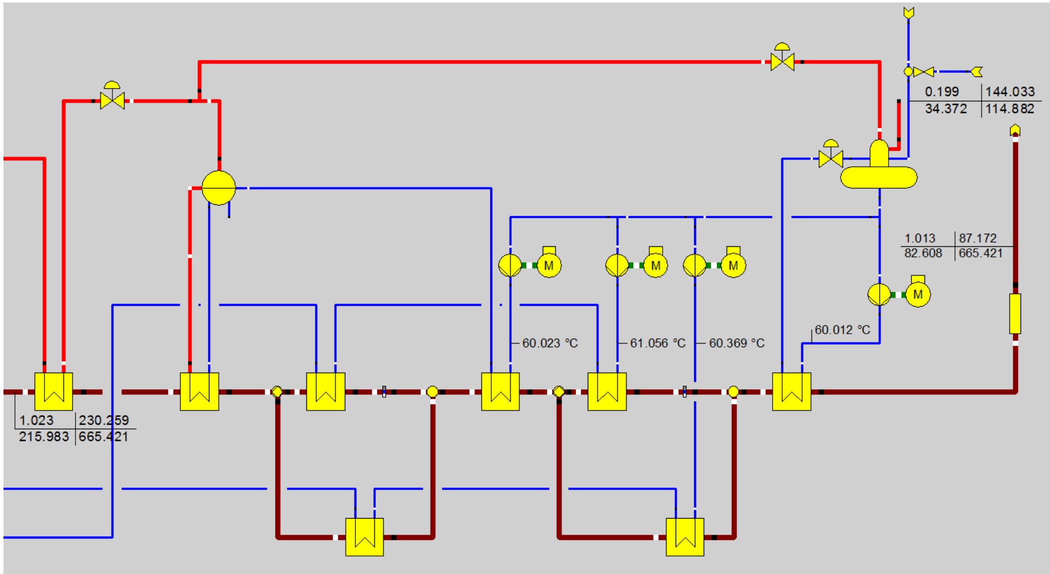

Figure 7: Cold-end configuration with external recirculating deaerator ("RECIRC EXTDA")

This configuration uses a vacuum pressure deaerator operated at the saturation pressure corresponding to meet minimum feed water heater inlet temperature. The LP, IP and HP circuits are separated once leaving the deaerator. One additional heat exchanger is added to reduce the stack temperature and to heat water to aid in the deaeration. The circulation flow in this heat exchanger is varied to minimize the steam requirement for deaeration. The recirculation flow rate could be varied, if desired, to achieve lower stack temperatures and better heat rates, but is not required for the purpose feedwater temperature control. However, for the purpose of this study it was assumed the flow rate can be varied.

Study Approach

To ensure this study properly compares the different configurations, care was taken to apply the same design conditions and limits, such as minimum HRSG water inlet temperature, pinch points, maximum duct firing temperature, etc. in all configurations.

The cold-end of the HRSG has to be designed to handle several potential problems. In particular, these involve sulphuric acid condensation, fatigue stress, erosion due to steaming, carbonic acid and other so called Flow Accelerated Corrosion (FAC) problems. Temperature control, deaeration, maintaining reducing environments and pH control all play a role in this.

In order to prevent condensation of sulphuric and other acid gases, the cold-end of the HRSG must be properly designed to ensure metal temperatures stay above the appropriate dew points.

Of the sulphuric gases which could cause corrosion of HRSGs, the one with the lowest dew point is typically SO_3 (H_2SO_4 in liquid form).

This empirical correlation can be used to estimate the sulphuric dew point:

T_{dew}, $H_2SO_4 = 1000/[2.276 - 0.0294 \ln(p\ H_2O) - 0.0858 \ln(p\ SO_3) + 0.0062 \ln(p\ H_2O\ p\ SO_3)]$;
with P in atm, T in K (3)

But this requires an assumption on SO_2 to SO_3 conversion, typically up to 5-6% (4, 5). The actual conversion rate will be dependent on the amount of excess air and the residence time.

Some additional H_2SO_4 can come from SO_2 to SO_3 conversion if SCR is present. And finally, some addition H_2SO_4 can form in the following reaction:

$NO_2 + SO_2 + H_2O = H_2SO_4 + NO$

Nitric acid usually has a lower dew point than H_2SO_4 and is therefore less restrictive.

$4NO_2 + 2H_2O + O_2 = 4HNO_3$

T_{dew}, $HNO_3 = T_d\ NO_2 = 1000/(3.664 - 0.1446*\ln(H_2O\ /100*760) - 0.0827*\ln(NO_2/1000000*760) + 0.00756*\ln(H_2O\ /100*760)*\ln(NO_2/1000000*760)) – 273$;
with H_2O in vol%, NO_2 in ppmv (6)

Similarly the sulphurous dew point is also less restrictive:

T_{dew}, $H_2SO_3 = 1000/\{3.9526 - 0.1863*\ln(PH_2O) + 0.000867*\ln(PSO_2) - 0.00091*\ln(PH_2O*PSO_2)\}$
with P in inHg, T in K (7)

Once the dew point and the surface temperatures are known (or estimated), gas temperature does have some impact on corrosion too, namely on the acid deposit rates of the acid film. The impact might be counter-intuitive: the higher the stack temperature, the faster the deposit rate due to higher mass and energy transfer rates (8).

Finally, the water dew point is a function of the partial pressure of H_2O only.

For typical exhaust gas compositions in gas turbine fired power plants, the sulphuric dew point is the most restrictive, if Sulphur is present in the fuel gas or oil. In the absence of sulphuric gases the water dew point in the flue gas must be taken into consideration.

The design of the HRSG should allow for control of the surface temperature of the coldest heat exchanger tubes to ensure that they remain above the most restrictive dew point temperature. The coldest temperature is at the surface of the coldest tubes. Due to the larger interior heat transfer coefficient (liquid phase) compared to the exterior heat transfer coefficient (gas phase), the surface temperature is governed most by the water inlet temperature. Therefor the water inlet temperature must be maintained above the most restrictive dew point. This is achieved either by changing deaerator pressure (configuration 5), or by providing a preheater bypass/recirculation (in the other 4 configurations).

The main purpose of the deaerator is to remove Dissolved Oxygen (DO) and other unwanted gases, such as CO2, from the water stream. Leaving them in would cause or enhance corrosion which can occur even at relatively low temperatures. Additional purposes of the deaerator include preheating the condensate and providing the necessary Net Positive Suction Head (NPSH) to the Boiler Feed Pumps.

Unwanted gases can enter through leaks in condensate pumps or hotwell, which can for example be eliminated with welding. But, air entering along with gland seal steam must be removed by venting. Makeup water and process condensate return from cogeneration plants often contain dissolved gases as well.

There are also potential problems on the water side of the tubes. According to (9) Flow Accelerated Corrosion (FAC) is a progressive form of water-side metal wastage that strips away metal from the wetted surfaces of pressure parts. Pressure parts will thin if FAC is occurring, and failures will result if it is allowed to continue. Often two types of FAC are considered separately, single phase and 2-phase FAC.

The phenomenon is most likely to attack pressure parts under the following conditions:

- Reducing environment (zero oxygen, possible excess scavenger).
- Extremely low DO content (at or near zero).
- Low boiler-water pH (less than 9.2).

- High water-side velocities.
- Water temperatures between 110C and 249 C / 230F and 480F.
- Pressure parts made of carbon steel.

Practically this means that the cold-end section of the HRSG is most susceptible to this problem, and to some degree the IP section.

Iron can dissolve in water $Fe + 2H_2O \rightarrow Fe(OH)_2 + 2H+ +2e-$, and iron hydroxide may form:

$$2\ Fe + O_2 + 2\ H_2O \text{->} 2\ Fe(OH)_2 \quad (10)$$

This can then react further with O2:

$$4\ Fe(OH)_2 \quad + O2 +2\ H_2O \text{->} 4\ Fe(OH)_3$$

Reactivity increases with temperature. It can also react further to form magnetite (Fe_3O_4), which has the ability to form a protective layer:

$$3\ Fe(OH)_2 \text{->} Fe_3O_4 + 2\ H_2O + 2\ H^+ + 2e^-$$

This protective layer may not form when O_2 is present and the water is mildly alkaline (pH 7-9.2). Magnetite, under normal circumstances, allows only limited amounts of the iron to dissolve in water (up to 5ppb), but when FAC occurs, often due to a reduction reaction (such as hydrogen presence in the water) it can be much higher, quickly leading to thinning tubes. (9).

Two-phase FAC (steaming) cannot be influenced by Oxygen concentration, this is usually done by way of local pH control (~9.8). (11). While outside the scope of this paper, any material removed by way of FAC will contribute to Under-Deposit Corrosion, most likely in the HP evaporator.

CO_2 can lead to carbonic acid formation:

$$CO_2 + H_2O \rightarrow H_2CO_3$$

This leads to corrosion and increases acidity (lowers pH). Corrosion can also be reduced by maintaining a high enough pH, but that has an impact on the ability to form magnetite, as mentioned earlier. Chemically bound CO_2 cannot be removed by deaeration so pH is kept relatively high to avoid the formation of carbonic acid. This pH cannot be much above 9.8, since caustic corrosion would cause the protective magnetite layer to be dissolved (9). It should also not drop too much lower than 9.2 either to prevent acidic corrosion.

These and other undesired chemical reactions force the removal of CO_2 and O_2. The deaerator achieves this by spraying the cold or preheated condensate water into the top of the deaerator and steam is added to the deaerator to heat the water via direct contact up to the saturation point. Spraying to reduce surface tension and heating to reduce solubility of the gases both help to release unwanted gases from the deaerator. A second stage of removal can be achieved by using scrubbers, such as trays or by releasing steam at the bottom of the water vessel (12). The steam supply in an HRSG will mostly come from the LP evaporator, but additional pegging steam can be taken from IP or even HP letdowns in case the LP evaporator cannot produce enough steam. It is obvious then that the deaerator must be operated at a pressure below or at LP evaporator pressure. Some heating can be done in preheaters, so that less steam production is required. But since some steam is always required, the inlet temperature to the deaerator should maintain a minimum approach to prevent deaerator steam flow control issues. In case of preheaters, it is therefore often required to have a potential bypass around the preheater for part load conditions where the preheater exit temperature would approach the deaerator operating temperature too closely.

In addition to controlling the steam flow, the deaerator pressure can be controlled, too. If the preheaters don't provide a lot of energy to the deaerator, lowering the pressure in the deaerator reduces the steam requirement. Of course there is a minimum pressure that must be maintained in the LP evaporator, dictated by the LP admission pressure into the LP steam turbine inlet.

Furthermore, the maximum deaerator pressure is also limited once the material of the boiler feed pump is chosen. The temperature in the deaerator often may thus not be allowed to exceed 150C.

The benefit of having an integral deaerator versus a separate deaerator lies in investment cost savings. But a disadvantage is that the deaerator pressure and LP evaporator pressure cannot be independently controlled and optimized. Furthermore, independent pH control for the HP and IP and LP sections is more difficult, if the LP drum has an integral deaerator.

Another advantage of having a recirculation system is apparent during start up. Most feedwater heaters will contain water during shutdown and will initially heat up, but without flowing, since steam production has not started yet. By the time steam production starts, cold water would be added to the already warm water inside the feed water heaters. Such cold water being added to warm tubes would lead to shock and the resulting thermal fatigue, which in turn leads to cracks. Furthermore, it can damage the protective magnetite layer (9). Recirculating warm water prior to steam production and mixing it with the cold condensate prior to entering the preheater during startup operation is one way to prevent such problems. Furthermore, a recirculation system is useful to maintain minimum flow in the preheater under low load conditions when steaming in the preheaters may occur or when tube-to-tube temperature differences might get too high (13). Another way to prevent steaming is to ensure that he condensate pump provides sufficient pressures that are much higher than the deaerator or LP evaporator pressures.

It is obvious from the previous discussions that there are several optimization points within the cold-end section of the HRSG, and also several restrictions.

Optimization points include location of deaerator, preheater size, deaerator pressure, or LP evaporator pressure. Some of the optimizations need to be made at the design point, while others require adjustments derived from evaluation a range of operating conditions / points.

Restrictions such as: minimum feed water heater inlet temperature and minimum deaerator approach temperature need to be considered. For this study, we assumed the following restrictions related to the cold-end design:

- minimum feed water heater inlet temperature = 60C / 140F
- minimum deaerator approach temperature = 10C / 18F
- SO_2 to SO_3 conversion rate = 5%

To compare the performance of the various HRSGs, one cannot simply compare the HRSG performance by itself. It is after all possible to improve the HRSG performance without improving plant performance. One example of this would be that a lower LP evaporator pressure might reduce stack temperatures and appear to improve performance, but lower LP evaporator pressures might not benefit the steam turbine as much as a higher LP evaporator pressure would.

For each configuration studied, several runs were performed to investigate their impact on the HRSG cold-end parameters

a. Base case ("ISO"), with ISO ambient conditions, no duct firing, all HRSGs in operation
b. Off-design ("Duct Fire"), 35C ambient. temperature, 90% rel. humidity and maximum duct firing
c. Off-design ("Fuel Oil") as the higher sulphur case (fuel oil fired GT, w/o duct burner) at ISO
d. Off-design ("1 GT off") as 1GT/HRSG outage mode under ISO conditions with maximum duct firing

Results

While the achievable net electrical output is approximately the same in all configurations (see Table 3 below), the heat rate for those configurations with external product/feed heat exchanger and integral deaerator (i.e. configurations 2 and 4) show significantly higher heat rates in the oil-fired case.

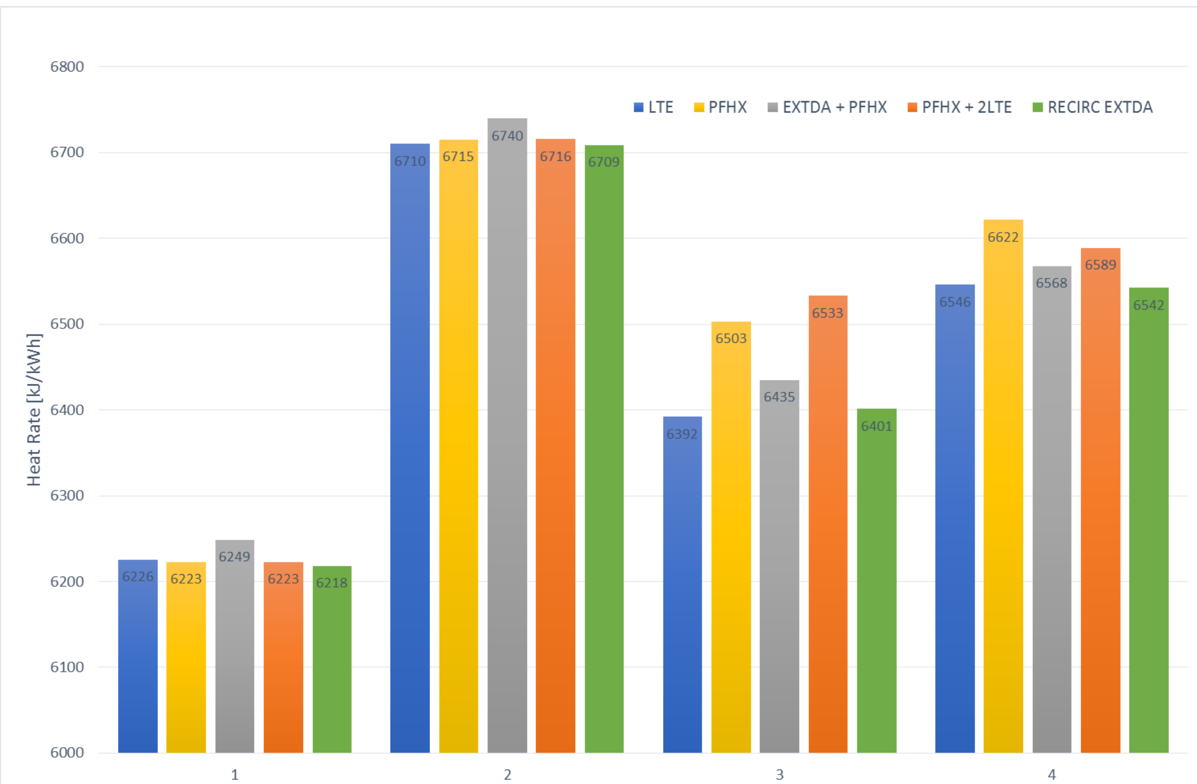

Figure 8: Comparison of overall plant net heat rate for the cold-end configurations for four operating cases: base load at ISO (1), 35C ambient temperature with maximum duct firing (2), base load operation with fuel oil (3), and outage mode operation with 1 GT/HRSG train off (4).

		Configuration 1				Configuration2				Configuration 3				Configuration 4				Configuration 5			
		LTE				PFHX				EXTDA + PFHX				PFHX + 2LTE				RECIRC EXTDA			
Parameter	**UOM**	ISO	Duct Fire	Fuel Oil	1GT off	ISO	Duct Fire	Fuel Oil	1GT off	ISO	Duct Fire	Fuel Oil	1GT off	ISO	Duct Fire	Fuel Oil	1GT off	ISO	Duct Fire	Fuel Oil	1GT off
Plant Net Heat Rate	kJ/kWh	6226	6710	6392	6546	6223	6715	6503	6622	6249	6740	6435	6568	6223	6716	6533	6589	6218	6709	6401	6542
Plant Net Power	MW	828.5	804.8	769.1	458.8	828.9	804.3	756.0	453.4	825.4	800.6	764.0	457.0	828.8	804.2	752.5	455.9	829.5	803.2	768.0	458.3
Auxiliary Power	MW	15.6	17.8	12.6	12.9	15.2	17.4	11.4	12.5	14.8	16.9	11.2	12.3	15.2	17.4	11.5	12.5	15.3	17.2	11.8	12.7
HP Steam Production	kg/sec	166.4	231.6	155.8	124.1	166.4	231.7	156.5	124.2	166.3	230.4	156.0	123.6	166.4	231.7	156.8	124.3	167.1	230.8	156.6	123.9
HP Steam Pressure	bara	93.0	127.4	86.6	71.0	93.0	127.5	86.7	70.9	93.0	126.8	86.7	70.7	93.0	127.5	86.7	71.1	93.0	126.4	86.6	70.6
HP Steam Temperature	°C	569.4	569.4	555.9	569.4	569.4	569.4	555.6	569.4	569.4	569.4	555.8	569.4	569.4	569.4	555.7	569.4	569.4	569.4	555.9	569.4
HRH Steam Flow	kg/sec	183.9	243.3	174.1	130.5	183.9	243.4	172.0	128.3	184.5	241.8	175.0	129.9	183.9	243.4	168.5	130.6	184.2	241.2	174.7	129.9
HRH Steam Pressure	bara	29.3	38.6	27.5	20.8	29.3	38.6	27.1	20.4	29.3	38.3	27.5	20.6	29.3	38.6	26.5	20.8	29.3	38.2	27.5	20.7
HRH Steam Temperature	°C	568.4	567.8	554.2	567.8	568.4	567.8	554.5	567.8	568.4	567.8	554.1	567.8	568.4	567.8	554.2	567.8	568.4	567.8	554.1	567.8
IP Steam Production	kg/sec	8.8	5.8	9.1	6.4	8.8	5.8	9.4	6.5	9.1	5.7	9.5	6.4	8.8	5.8	9.4	6.4	8.6	5.2	9.0	6.0
IP Steam Pressure	bara	30.5	40.0	28.7	23.9	30.5	40.0	28.2	23.4	30.5	39.7	28.7	23.7	30.5	40.0	27.6	23.8	30.5	39.6	28.7	23.7
IP Steam Temperature	°C	307.2	338.7	300.9	309.8	307.2	338.7	300.3	309.4	307.2	339.6	300.9	310.2	307.2	338.7	299.9	309.8	307.2	339.8	300.7	310.1
LP Steam Production	kg/sec	25.6	18.0	21.1	6.0	25.5	15.7	0.0	0.0	19.3	13.0	9.4	4.0	25.5	15.6	0.0	0.0	28.2	20.9	18.3	7.2
LP Steam Pressure	bara	5.4	6.7	5.0	3.5	5.4	6.7	4.4	3.3	5.4	6.8	4.9	3.6	5.4	6.7	4.3	3.4	5.4	6.7	4.9	3.5
LP Steam Temperature	°C	293.9	327.9	296.5	310.3	293.9	332.5	146.9	136.8	293.9	328.8	309.0	311.7	293.9	332.7	146.1	137.4	293.9	325.0	303.9	308.6
Condenser Pressure	mbar	48.3	144.7	44.7	32.7	48.3	143.6	39.5	31.1	48.3	144.9	43.4	32.9	48.3	143.5	38.7	31.5	48.3	143.3	43.6	32.4
Preheater Recirc Flow	kg/sec	28.8	6.8	165.0	73.7	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	79.1	25.5	160.0	160.0
Preheater Bypass Flow	kg/sec	0.0	0.0	0.0	0.0	0.0	0.0	83.5	0.0	0.0	0.0	99.0	0.0	0.0	0.0	0.0	29.4	0.0	0.0	0.0	0.0
Deaerator Pressure	bara	6.25	7.10	5.64	3.82	6.25	6.97	4.37	3.30	1.03	1.03	1.14	1.04	6.25	6.96	4.29	3.36	0.20	0.20	1.01	0.20
PH Inlet Temperature	°C	60.0	60.0	99.9	60.0	60.0	69.4	100.0	61.1	60.0	74.0	100.0	59.7	60.0	69.0	bypass	60.0	60.0	60.0	100.0	60.0
PH2 Inlet Temperature	°C	-	-	-	-	69.6	77.5	bypass	bypass	-	-	-	-	69.6	77.0	bypass	71.9	-	-	-	-
PH3 Inlet Temperature	°C	-	-	-	-	-	-	-	-	-	-	-	-	100.0	96.0	bypass	bypass	-	-	-	-
Stack temperature	°C	88.6	84.6	105.8	71.6	88.0	88.6	152.7	105.5	99.5	98.2	126.8	81.8	88.0	88.8	160.5	95.1	82.6	81.4	109.8	68.0
Sulphuric Dew Point	°C	0.0	0.0	89.5	0.0	0.0	0.0	89.5	0.0	0.0	0.0	89.5	0.0	0.0	0.0	89.5	0.0	0.0	0.0	89.5	0.0
Nox Dew Point	°C	11.0	13.5	8.3	11.7	11.0	13.5	8.3	11.7	11.0	13.5	8.3	11.7	11.0	13.5	8.3	11.7	11.0	13.5	8.3	11.7
Sulphurous Dew Point	°C	-0.7	0.2	0.1	0.2	-0.7	0.2	0.1	0.2	-0.7	0.2	0.1	0.2	-0.7	0.2	0.1	0.2	-0.7	0.2	0.1	0.2
Water Dew Point	°C	42.8	51.7	33.9	45.2	42.8	51.7	33.9	45.2	42.8	51.7	33.9	45.2	42.8	51.7	33.9	45.3	42.8	51.7	33.9	45.2
Total Surface Area	m²	230,196				224,964				248,987				223,651				272,166			

Table 3: Summary of heat balance results for plant configurations 1 to 5

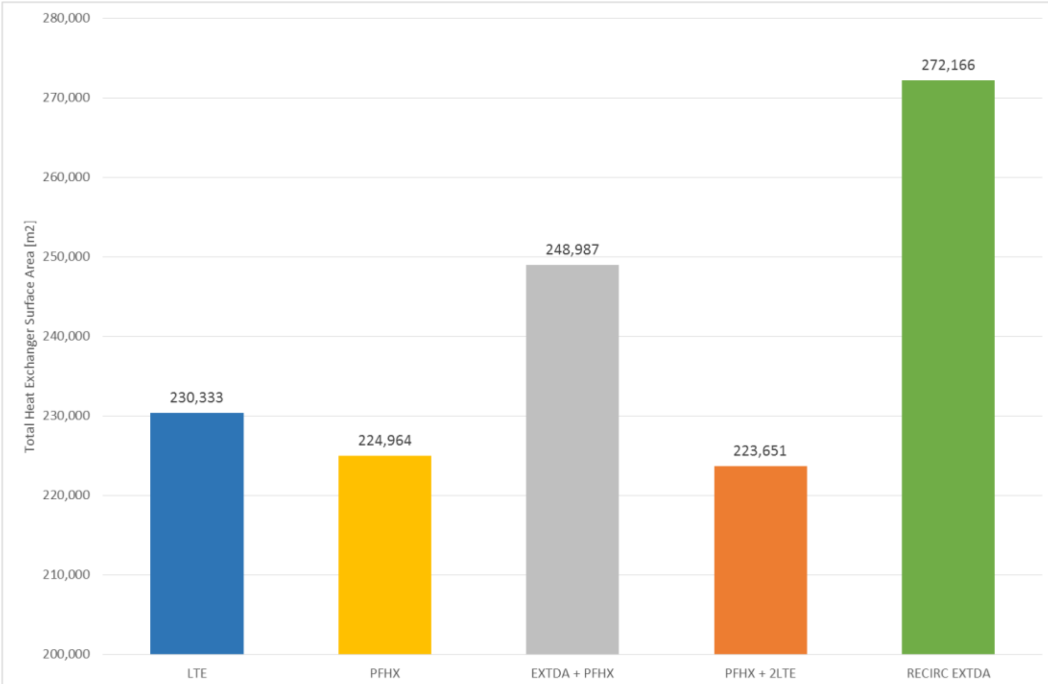

Figure 9: Comparison of total HX surface area for the five configurations

Conclusions

The pros and cons of the various configurations are summarized in Table 4 below.

Criterion/Feature	**LTE**	**PFHX**	**EXTDA + PFHX**	**PFHX + 2LTE**	**RECIRC EXTDA**
Heat Rate (average)	best	worst	average	average	best
Heat Rate for at design conditions	average	average	worst	average	best
Power (average)	highest	lowest	average	second lowest	second highest
Power at design conditions	average	average	lowest	average	highest
Stack Temperature (average)	second lowest	highest	average	Second Highest	lowest
Deaerator type	integrated	integrated	external, ~ 1 atm	integrated	external, vacuum
Deaerator upstream of gas/H_2O HX	no	no	yes	no	yes
HP and IP drum fed from LP drum	yes	yes	no	yes	no
Feed Water Inlet Temperature Control	recirculation, followed by bypass if needed	external HX bypass, followed by preheater bypass if needed	external HX bypass + deaerator pressure	external HX bypass, followed by preheater bypass if needed	deaerator pressure
Heat Exchange Area	average	smallest	second largest	smallest	largest
Number of Drums (HP,IP, LP,DA)	3: HP, IP, LP/DA	3: HP, IP, LP/DA	4: HP, IP, LP, DA	3: HP, IP, LP/DA	4: HP, IP, LP, DA
Pumps	HP, IP, Recirc	HP, IP	HP, IP, LP	HP, IP	HP, IP, LP, Recirc

Table 4: Qualitative assessment of the cold-end configurations under evaluation

Performance differences are obviously best expressed by looking at power and heat rate differences. But, other factors should be considered too. For example, the location of the deaerator impacts where the unwanted dissolved gases are removed and the effect of corrosion will be impacted by the location of the deaerator. If an HP and IP steam water/system is fed by the LP drum then this increases the likelihood of corrosion deposit carry-over from the LP system to the HP/IP systems, which could increase their vulnerability to Under-Deposit Corrosion. Furthermore, having the LP, IP and HP systems separated would make it easier to independently regulate the chemical composition of the water in the different sections to combat the various corrosion impacts, each of which of have different temperature dependencies.

Feedwater temperature control for the water stream entering the HRSG will determine in part the ability to decrease stack temperature (recirculation configurations out-perform bypass configurations in this aspect).

Parameters such as heat exchanger area, number of drums and number of pumps obviously have an impact on the cost. And parasitic loads will generally be higher when recirculation pumps are used.

Performance (power and heat rate) is nearly the same for all configurations, at least for the most basic operating points. But for the less common operating points, such as fuel oil operation and a GT outage mode there are substantial differences. There is no configuration that is consistently the best option for power heat rate, capital cost and ease of operation.

Due to the differences in performance between recirculation options vs. external heat exchanger options, particularly for less common operating points, and due to the differences in investment cost, it is likely that the best configuration is dependent on the amount of running hours the plant is expected to spend in the less common operation points, to see if the additional cost is worthy investment.

Compared to the recirculation configurations (configuration 1 and 5), the external heat exchanger configurations (such as 2 and 4) reduce the overall heat exchanger surface area, and they eliminate a recirculation pump and its parasitic power requirement. This reduces capital and O&M costs and improves reliability. But as shown, the heat rate can be worse depending on the operating conditions.

In order to identify the optimal configuration for a given set of operating points, a flexible modeling software will be required which can handle all configurations and all off-design scenarios the plant is likely to encounter.

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