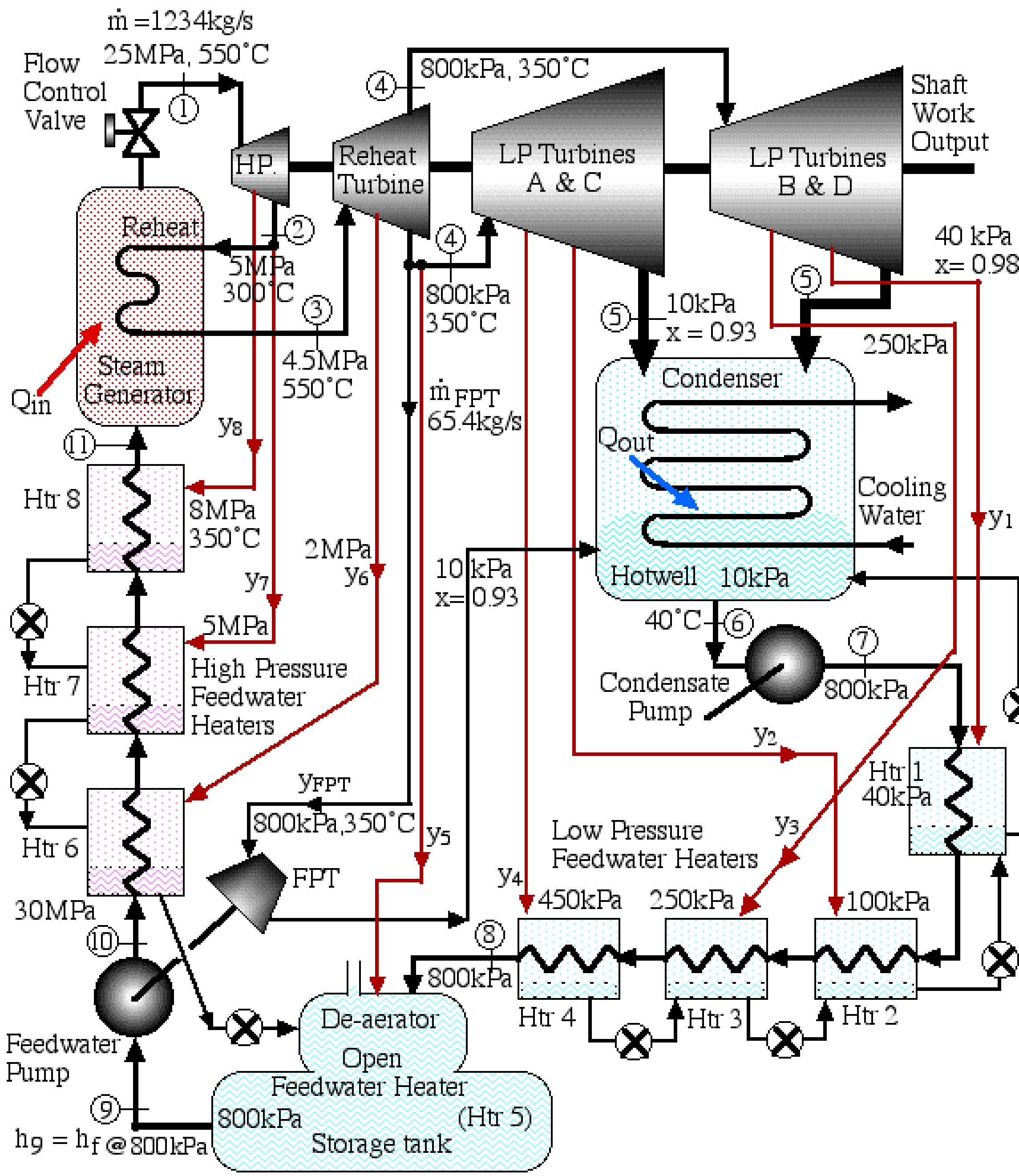
Steam Power Cycles Case Study - The General James M. Gavin Steam Power Plant

Background - The General James M. Gavin Plant's units 1 and 2 are identical, each with a generating capacity of 1300 MW. Unit 1 was completed in 1974 and Unit 2 was completed the following year. With a total generating capacity of 2,600 MW, Gavin Plant ranks as the largest generating station in the state of Ohio. It is located along the Ohio River at Cheshire, Ohio, and has an average daily coal consumption of 25,000 tons at full capacity. The coal arrives by barge and is stored in the plant's coal yard. Conveyer belts carry the coal from the yard into the plant where pulverizers grind the coal into a fine, talcum powder-like consistency. The powdered coal is injected into the steam generator where it is burned at high temperature providing the heat power $\dot{Q}_{in}$ which drives the power plant.

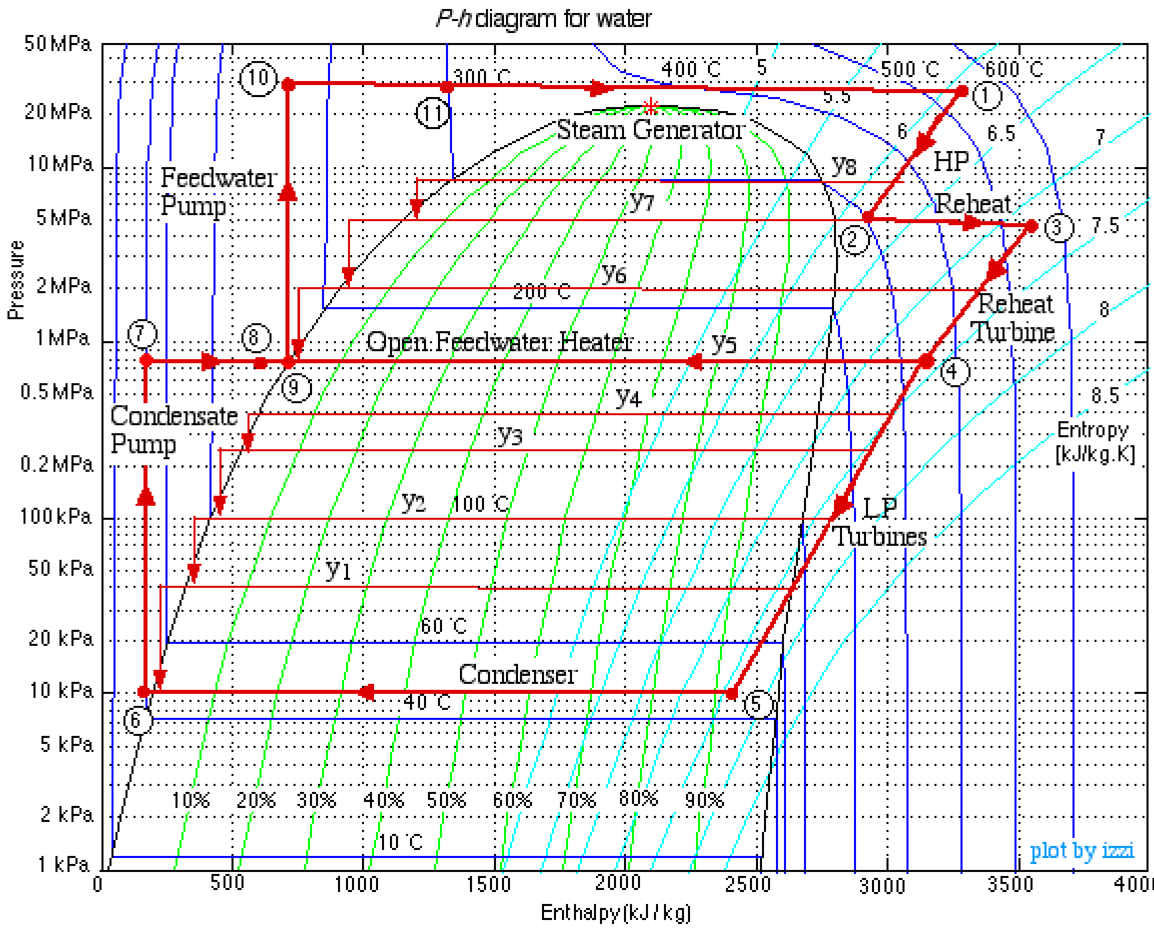
Schematic Diagram for Analysis - The formal schematic diagram of the Gavin Plant is extremely complex. There are six turbines on two separate parallel shafts, each driving a hydrogen cooled electrical generator producing 26,000 volts. Transformers outside the plant building step up this voltage to 765,000 volts so that it can be transmitted efficiently over a long distance. The high pressure (HP) turbine drives one shaft together with low pressure (LP) turbines A and B, and the intermediate pressure (Reheat) turbine drives the second shaft together with LP turbines C and D. The following represents a much simplified schematic diagram for purposes of doing an initial analysis of the system. Some of the state values shown were not available and represent estimates on the part of your instructor in order to enable a complete analysis.

Notice that the feedwater pump is driven by a separate 65,000HP turbine (FPT) which taps some of the steam from the outlet of the reheat turbine, returning the steam to the condenser hotwell. The feedwater pump pressurizes the water to 30 MPa, however the pressure at the HP turbine inlet drops to 25 MPa since the steam has had to pass through 350 miles of piping in the steam generator. The flow control valve together with the speed control of the feedwater pump enables control of output power matching it to the demand.

The system has four low pressure closed feedwater heaters, one open feedwater heater / de-aerator, and three high pressure closed feedwater heaters.

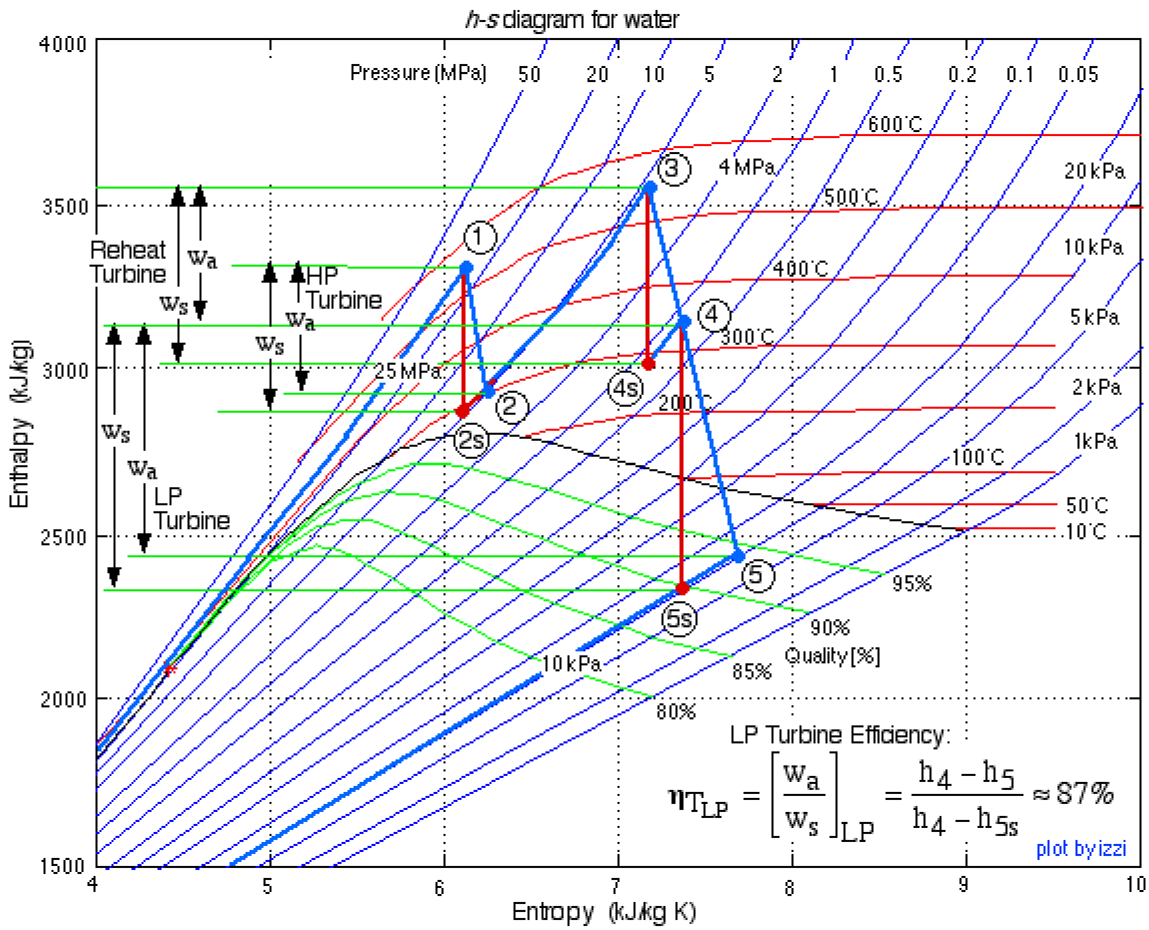


As always, prior to doing any analysis we always first sketch the complete cycle on a *P-h* diagram based on the data provided in the system diagram. This leads to the following diagram:



Notice from the *P-h* diagram how the three high pressure closed feedwater heaters progressively heat the steam from state (10) to state (11), thus the steam generator is only required to heat the steam from state (11) to state (1) leading to an increase in thermal efficiency. Similarly the four low pressure closed feedwater heaters progressively raise the temperature of the liquid from state (7) to state (8), thus reducing the fractional amount of steam required (y_5) in order to raise the temperature of the liquid from state (8) to state (9). It is true that as we draw off steam from the turbines for all the heaters, we reduce the output power accordingly, however the net effect of this process is to increase the overall thermal efficiency of the system.

One important consideration is the choice of the state (5) at the outlet of the low pressure turbines. The quality (x = 0.93) shown on the flow diagram is not a measurable quantity, and the identical pressure and temperature conditions exist throughout the quality region. The only guide that we have is the knowledge that steam turbine adiabatic efficiencies vary between 85% and 90%, thus in order to ensure that we are choosing reasonable state values we plot all three turbines on the companion *h-s diagram* indicating both the isentropic as well as the actual processes on the diagram as follows:



Thus from the diagram we determined that the choice of quality x = 0.93 brought us into the correct efficiency range. This is an extremely critical choice, since by choosing a quality that is too low can lead to erosion of the turbine blades and a reduction of performance. One example of the effects of this erosion can be seen on the blade tips of the final stage of the Gavin LP turbine. During 2000, all four LP turbines needed to be replaced because of the reduced performance resulting from this erosion. (Refer: Tour of the Gavin Power Plant - Feb. 2000)

We now do an enthalpy inventory of the known state points on the cycle using either the Steam Tables or more conveniently directly from the NIST Chemistry WebBook (avoiding the need for interpolation), leading to the following table:

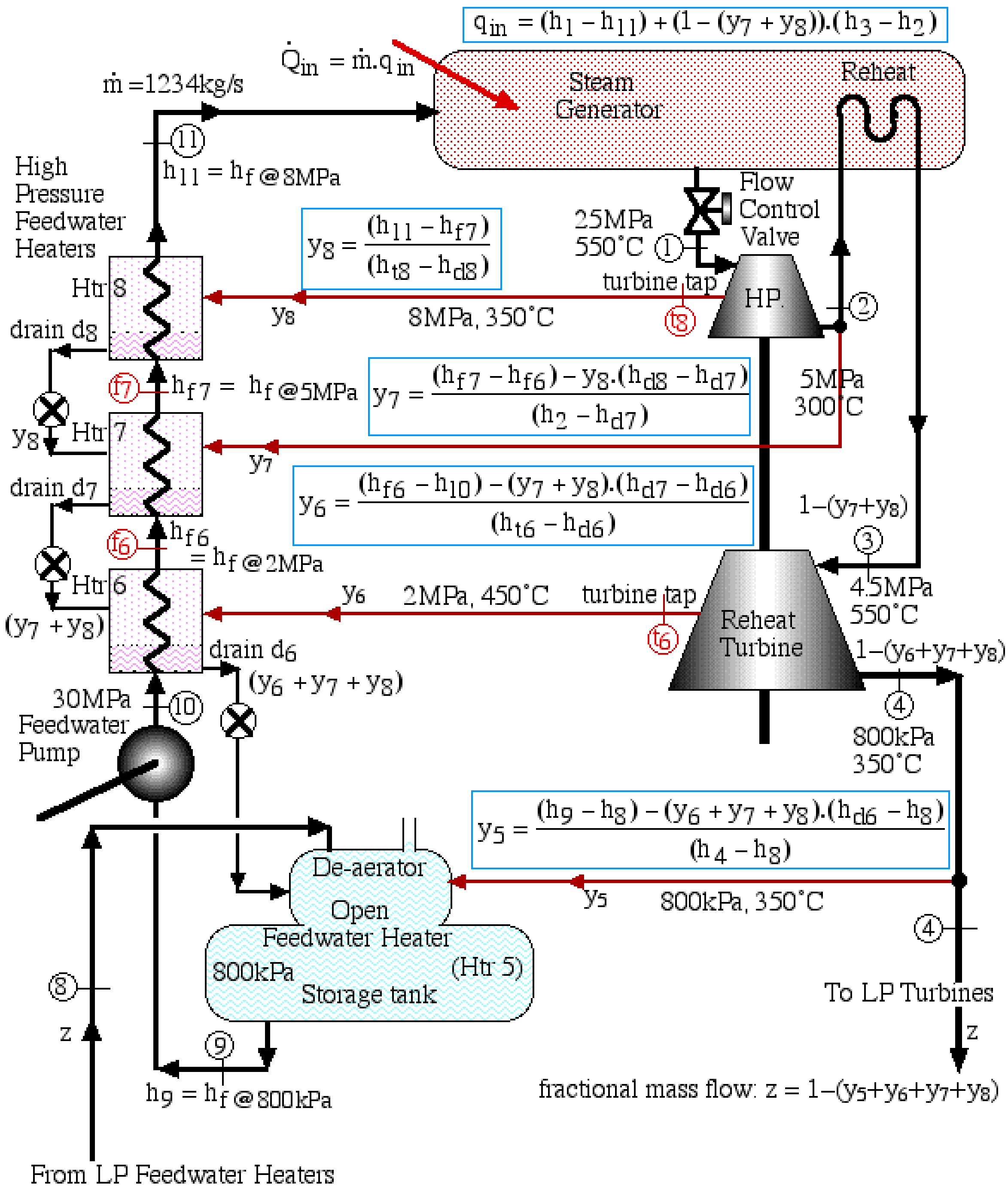
State	**Position**	**Enthalpy h [kJ/kg]**
1	HP turbine inlet	$h_1 = h_{25MPa,\ 550°C}$ = 3330 [kJ/kg]
2	HP turbine outlet	$h_2 = h_{5MPa,\ 300°C}$ = 2926 [kJ/kg]
3	Reheat turbine inlet	$h_3 = h_{4.5MPa,\ 550°C}$ = 3556 [kJ/kg]
4	LP turbine inlet	$h_4 = h_{800kPa,\ 350°C}$ = 3162 [kJ/kg]

5	LP turbine outlet (quality region)	$h_5 = h_{10kPa,\text{ quality }X=0.93} = h_f+X.(h_g-h_f)$ h_f = 192 [kJ/kg], h_g = 2584 [kJ/kg] => h_5 = 2417 [kJ/kg]
6	Hotwell outlet (subcooled liquid)	$h_6 = h_{f@40°C}$ = 168 [kJ/kg]
7	Condensate Pump outlet	$h_7 = h_6$ = 168 [kJ/kg]
9	Open Feedwater Heater (saturated liquid)	$T_9 = T_{sat@800kPa}$ = 170°C $h_9 = h_{f@800kPa}$ = 721 [kJ/kg]
10	Feedwater Pump outlet (compressed liquid)	$T_{10} = T_9$+5°C = 175°C $h_{10} = h_{30MPa,\,175°C}$ = 756 [kJ/kg] (Compressed liquid)

Note: State points (8) and (11) result respectively from the low- and high-pressure closed feedwater heaters and are evaluated below. Notice that the temperature T_{10} is 5°C higher than the temperature T_9. Normally we consider liquid water to be incompressible, thus pumping it to a higher pressure does not result in an increase of its temperature. However on a recent visit to the Gavin Power Plant we discovered that at 30MPa pressure and more than 100°C, water is no longer incompressible, and compression will always result in a temperature increase of up to 7°C. We cannot use the simple incompressible liquid formula to determine pump work, however need to evaluate the difference in enthalpy from the Compressed Liquid Water tables, leading to the enthalpy h_{10} shown in the table.

Finally, do not forget that all values of enthalpy obtained should be checked for validity against the above *P-h* and *h-s* diagrams.

Analysis - We need to determine the mass fractions of all the feedwater heaters y_i as well as that drawn off for the feedwater pump turbine, in order to evaluate the heat input and the total power output of the system. We find it convenient to separate the system into a high pressure section including the HP and Reheat turbines, and a low pressure section including the two LP turbine sets. Using the techniques of enthalpy balance on the open and closed feedwater heaters developed in Chapter 8b, we obtain the mass fraction equations of the high pressure section as summarized in the following diagram.



In order to enable evaluation of the enthalpies at the various state points in the diagram we estimated the various intermediate temperature values at the turbine taps from the above *P-h*and *h-s* diagrams. The closed feedwater heaters are all of type counterflow heat exchangers, and we make the assumption that the outlet temperature equals the saturation temperature of the respective turbine tap, and that the drain temperature is 5°C above the inlet temperature value. The resulting enthalpy inventory of the intermediate state points follows:

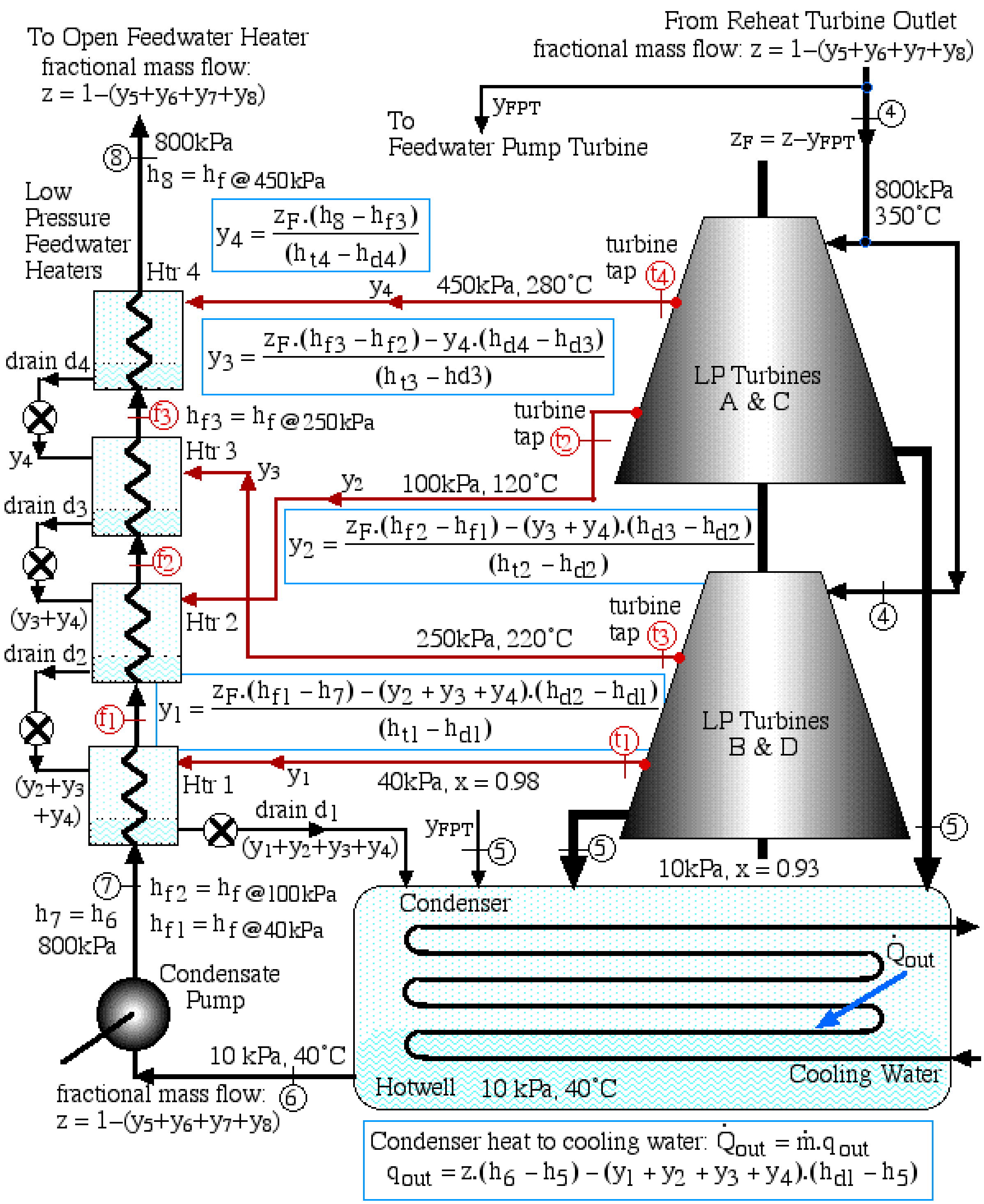
State	Position	Enthalpy h [kJ/kg]
t_8	HP Turbine tap	$h_{t8} = h_{8MPa,\ 350°C} = 2988$ [kJ/kg]
11	Closed Feedwater Heater #8 outlet	$T_{11} = T_{sat@8MPa} = 295°C$ $h_{11} = h_{30MPa,\ 295°C} = 1304$ [kJ/kg]
f_7	Closed Feedwater Heater #7 outlet	$T_{f7} = T_{sat@5MPa} = 264°C$ $h_{f7} = h_{30MPa,\ 264°C} = 1154$ [kJ/kg]
d_8	Closed Feedwater Heater #8 drain	$T_{d8} = T_{f7}+5°C = 269°C$ $h_{d8} = h_{8MPa,\ 269°C} = 1179$ [kJ/kg]
f_6	Closed Feedwater Heater #6 outlet	$T_{f6} = T_{sat@2MPa} = 212°C$ $h_{f6} = h_{30MPa,\ 212°C} = 918$ [kJ/kg]
d_7	Closed Feedwater Heater #7 drain	$T_{d7} = T_{f6}+5°C = 217°C$ $h_{d7} = h_{5MPa,\ 217°C} = 931$ [kJ/kg]
t_6	Reheat Turbine tap	$h_{t6} = h_{2MPa,\ 450°C} = 3358$ [kJ/kg]
d_6	Closed Feedwater Heater #6 drain	$T_{d6} = T_{10}+5°C = 180°C$ $h_{d6} = h_{2MPa,\ 180°C} = 764$ [kJ/kg]

The resultant fractional mass flow rates to the high pressure heat exchanger section follows:

Mass flow path	State conditions	Fractional mass flow
HP Turbine tap t_8 to Closed Feedwater Heater #8	8MPa, 350°C	$y_8 = 0.083$
HP Turbine outlet 2 to Closed Feedwater Heater #7	5MPa, 300°C	$y_7 = 0.108$
Reheat Turbine tap t_6 to Closed Feedwater Heater #6	2MPa, 450°C	$y_6 = 0.050$
Reheat Turbine outlet 4 to Open Feedwater	800kPa, 350°C	$y_5 = 0.025$

Heater #5		

Similar to the high pressure section above we obtain the mass fraction equations for the low pressure section as summarized in the following diagram:



The enthalpy inventory of the intermediate state points indicated on the above diagram follows:

State	Position	Enthalpy h [kJ/kg]
t_4	LP A&C Turbine tap	$h_{t4} = h_{450kPa,\ 280°C} = 3025$ [kJ/kg]
8	Closed Feedwater Heater #4 outlet	$T_8 = T_{sat@450kPa} = 148°C$ $h_8 = h_{800kPa,\ 148°C} = 624$ [kJ/kg]
t_3	LP B&D Turbine tap	$h_{t3} = h_{250kPa,\ 220°C} = 2909$ [kJ/kg]
f_3	Closed Feedwater Heater #3 outlet	$T_{f3} = T_{sat@250kPa} = 127°C$ $h_{f3} = h_{800kPa,\ 127°C} = 534$ [kJ/kg]
d_4	Closed Feedwater Heater #4 drain	$T_{d4} = T_{f3}+5°C = 132°C$ $h_{d4} = h_{450kPa,\ 132°C} = 555$ [kJ/kg]
t_2	LP A&C Turbine tap	$h_{t2} = h_{100kPa,\ 120°C} = 2717$ [kJ/kg]
f_2	Closed Feedwater Heater #2 outlet	$T_{f2} = T_{sat@100kPa} = 100°C$ $h_{f2} = h_{800kPa,\ 100°C} = 420$ [kJ/kg]
d_3	Closed Feedwater Heater #3 drain	$T_{d3} = T_{f2}+5°C = 105°C$ $h_{d7} = h_{250kPa,\ 105°C} = 440$ [kJ/kg]
t_1	LP B&D Turbine tap	$h_{t1} = h_{40kPa,\ quality\ X=0.98} = h_f+X.(h_{fg})$ $h_f = 318$ [kJ/kg], $h_{fg} = 2319$ [kJ/kg] => $h_{t1} = 2590$ [kJ/kg]
f_1	Closed Feedwater Heater #1 outlet	$T_{f1} = T_{sat@40kPa} = 76°C$ $h_{f1} = h_{800kPa,\ 76°C} = 319$ [kJ/kg]
d_2	Closed Feedwater Heater #2 drain	$T_{d2} = T_{f1}+5°C = 81°C$ $h_{d2} = h_{100kPa,\ 81°C} = 339$ [kJ/kg]
d_1	Closed Feedwater Heater #1 drain	$T_{d1} = T_6+5°C = 45°C$ $h_{d1} = h_{40kPa,\ 45°C} = h_{f@45°C} = 188$ [kJ/kg]

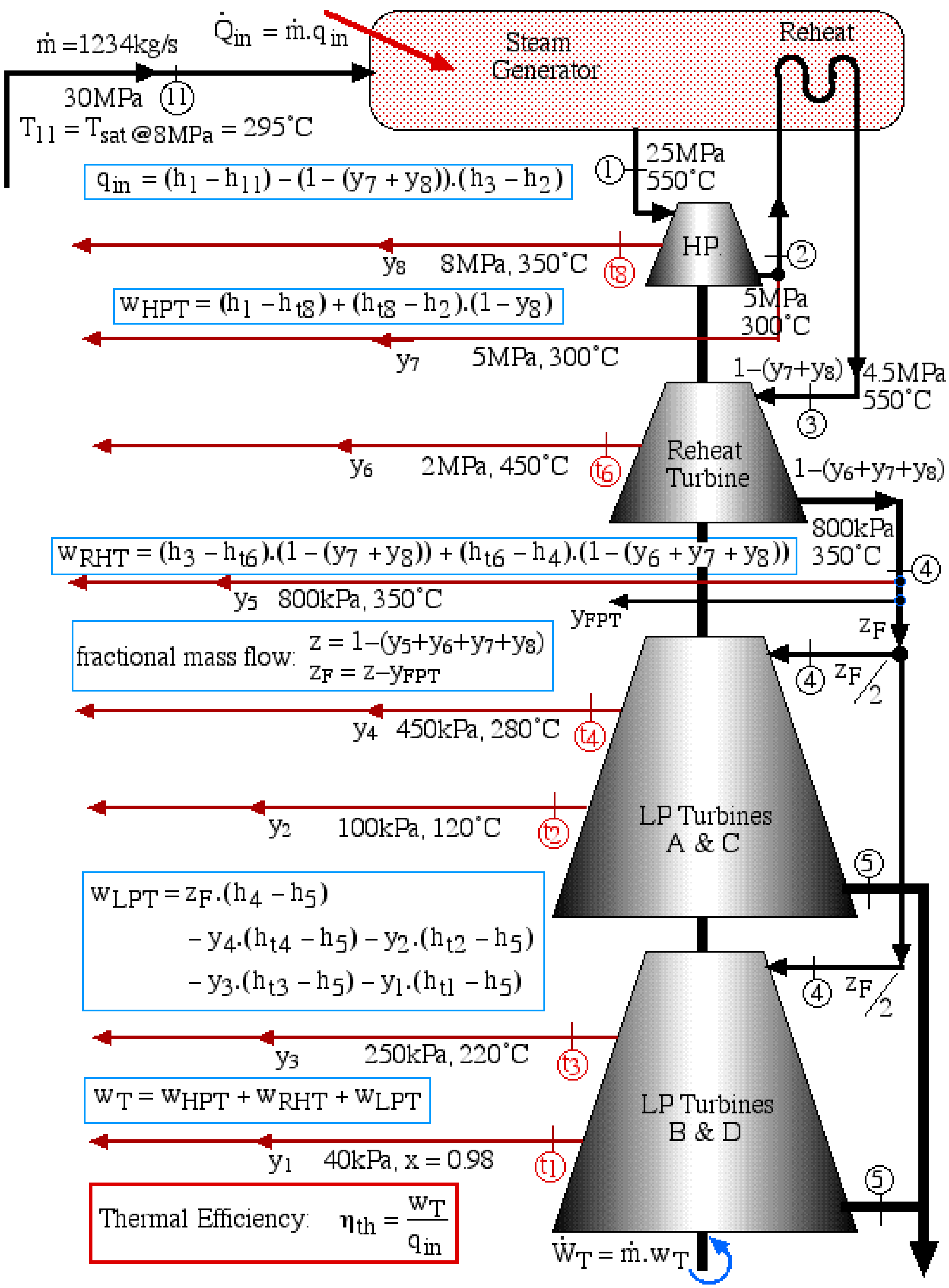
The resulting fractional mass flow rates to the low pressure heat exchanger section follows:

Mass flow path	State conditions	Fractional mass flow
Reheat Turbine outlet 4 to Feedwater Pump	800kPa, 350°C	yFPT = 0.053

Turbine (mass fraction 65,4[kg/s]/1234[kg/s])		
LP $Turbine_{A\&C}$ tap t_4 to Heater #4	450kPa, 280°C	$y_4 = 0.027$
LP $Turbine_{B\&D}$ tap t_3 to Heater #3	250kPa, 220°C	$y_3 = 0.033$
LP $Turbine_{A\&C}$ tap t_2 to Heater #2	100kPa, 120°C	$y_2 = 0.029$
LP $Turbine_{B\&D}$ tap t_1 to Heater #1	40kPa, quality X=0.98	$y_1 = 0.041$

From the above diagrams, an energy equation balance on the various components of the system leads to the following equations for the total turbine work output (wT kJ/kg), the total heat input to the steam generator (q_{in} kJ/kg) and the thermal efficiency η_{th}.

Source: http://www.ohio.edu/mechanical/thermo/Applied/Chapt.7_11/SteamPlant/GavinCaseStudy.html



Performance Results - Finally we have all the data and equations required to determine the performance with the following results:

- The work done by the HP, Reheat, and LP turbine set

$$w_{HPT} = (h_1 - h_{t8}) + (h_{t8} - h_2).(1 - y_8) \Rightarrow \boxed{w_{HPT} = 399\ [kJ/kg]}$$

$$w_{RHT} = (h_3 - h_{t6}).(1 - (y_7 + y_8)) + (h_{t6} - h_4).(1 - (y_6 + y_7 + y_8))$$

$$\Rightarrow \boxed{w_{RHT} = 309\ [kJ/kg]}$$

$$w_{LPT} = z_F.(h_4 - h_5)$$

$$- y_4.(h_{t4} - h_5) - y_3.(h_{t3} - h_5) - y_2.(h_{t2} - h_5) - y_1.(h_{t1} - h_5)$$

$$\Rightarrow \boxed{w_{LPT} = 471\ [kJ/kg]}$$

$$w_T = w_{HPT} + w_{RHT} + w_{LPT} \Rightarrow \boxed{w_T = 1179\ [kJ/kg]}$$

- The total heat input to the steam generator including the reheat section:

$$q_{in} = (h_1 - h_{11}) - (1 - (y_7 + y_8)).(h_3 - h_2) \Rightarrow q_{in} = 2536\ [kJ/kg]$$

Boiler Efficiency: $\eta_{boiler} = 88\% \Rightarrow \boxed{q_{in,actual} = 2882\ [kJ/kg]}$

- The thermal efficiency of the system. Up until now we have not considered the boiler efficiency. This is dependent on many factors, including the grade of coal used, the heat transfer and heat loss mechanisms in the boiler, and so on. A typical design value of boiler efficiency for a large power plant is 88%.

Thermal Efficiency: $\eta_{th} = \dfrac{w_T}{q_{in}} = \dfrac{1179\ [kJ/kg]}{2536\ [kJ/kg]} \Rightarrow \eta_{th} = 46\%$

Boiler Efficiency: $\eta_{boiler} = 88\% \Rightarrow \boxed{\eta_{th,actual} = 41\%}$

- The Feedwater pump and turbine performance

Feedwater Pump: $w_{FP} = (h_9 - h_{10}) \Rightarrow w_{FP} = -35\ [kJ/kg]$

Feedwater Pump Turbine: $w_{FPT} = y_{FPT}.(h_4 - h_5)$

$$\Rightarrow w_{FPT} = 39.5\ [kJ/kg]$$

Feedwater Pump Efficiency: $\eta_{FP} = \left|\dfrac{w_{FP}}{w_{FPT}}\right| = 89\%$

Feedwater Pump Power: $\dot{W}_{FPT} = \dot{m}.w_{FPT} = 48.7\ MW$ $(\approx 65{,}000\ hp)$

- The power output of the turbines, and heat power to the steam generator:

Mass flow rate of steam : $\dot{m} = 1234$ [kg/s]

Total turbine output power : $\dot{W}_T = \dot{m}.w_T \quad \Rightarrow \quad \boxed{\dot{W}_T = 1455\ \text{MW}}$

Steam generator heat power : $\dot{Q}_{in} = \dot{m}.q_{in,actual} \quad \Rightarrow \quad \boxed{\dot{Q}_{in} = 3556\ \text{MW}}$

Note: It is always a good idea to validate ones calculations by evaluating the thermal efficiency using only the heat supplied to the steam generator and that rejected by the condenser.

$$q_{out} = [z - (y_1 + y_2 + y_3 + y_4)].h_5 + (y_1 + y_2 + y_3 + y_4).h_{dl} - z.h_6$$

$$\Rightarrow \quad q_{out} = 1361\ [\text{kJ/kg}]$$

Thermal Efficiency: $$\eta_{th} = \frac{w_T}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} = 1 - \frac{1361\ [\text{kJ/kg}]}{2536\ [\text{kJ/kg}]}$$

$$\Rightarrow \quad \eta_{th} = 46\%$$

This is the same efficiency value as obtained by the direct method, thus validating the method.

Discussion - We were extremely satisfied that a system as complex as the Gavin Power Plant is amenable to this simplified analysis. Notice that no matter how complex the system is, we can easily plot the entire system on a *P-h* diagram in order to obtain an immediate intuitive understanding and evaluation of the system performance. The diagram also serves as a usefull validity check by comparing each value of enthalpy evaluated to the values on the enthalpy axis of the *P-h* diagram.

The analytical power output (1455 MW) is higher than the actual power output of 1300 MW mainly because of the significant electrical power required to run the power plant and the heat and pressure drop losses inherent in a large complex system. In order to justify the complexity of the seven closed feedwater heaters we analysed two simpler systems for comparison. In all cases we used the same steam mass flow rate of 1234 kg/s and the same feedwater pump turbine system as above. Note that the open feedwater heater also acts as a de-aerator and storage tank, and is thus a necessary component of the system.

- No closed feedwater heaters in the system. This allows all of the steam to be directed to the turbines resulting in a much higher power output of 1652 MW, however with a reduction in thermal efficiency from 46% to 41%.
- Using only the three high pressure closed feedwater heaters and not the four low pressure closed feedwater heaters. This requires a significant increase in the steam tapped from the outlet of the reheat turbine to be directed to the open

feedwater heater resulting in a lower power output of 1397 MW with a thermal efficiency of 45%.

Thus use of the seven closed feedwater heaters is justified, resulting in the maximum thermal efficiency together with a satisfactory power output,

Source: http://www.ohio.edu/mechanical/thermo/Applied/Chapt.7_11/SteamPlant/GavinCaseStudy.html