4I10 Exam 2013-14 Crib

1. Power Cycles

(a) *System temperatures limited system pressure and requirement of no bulk boiling in the core. With the size of vessels required 150 bar is the normal maximum operating pressure and as saturation temperature is 342*°C *the Hot leg temperature is limited a margin below this value to ensure that the hottest channel has no bulk boiling.*

Simple calculation related to PWR temperatures, plus (small but negative) temperature difference across SG ~20°C*. Estimate of max possible efficiency based on* $T_2 =$ *Cold-leg Temperature of 280*°C *and* $T_1 =$ *Heat sink 30*°C

$$\eta = \frac{280-20-30}{273+280} = 42\%$$

If it was sought to increase efficiency levels with the same basic arguments:

1. *Much larger steam generators could reduce the temperature loss by a small amount, and*
2. *Critical pressure of water is 221 bar with a saturation temperature of 374*°C *limits the scope for increases in core temperatures to 20-30*°C

We can see that the materials limits of vessels plus the properties of steam-water limit the efficiency without considering super-critical cycles which could have very different cooling characteristics.

Real power cycles, must take into account the turbine efficiency which is low with wet steam and the other power losses in the system such as feed and MCPs. On the other hand feed-heating provides a useful way of increasing cycle efficiency making use of water bled between stages of the wet steam turbine.

The *answer should consider other factors e.g SG heat transfer area, feed heating (and perhaps) steam reheating, illustrated with T-s diagrams.*

(b) *Thermal η* = *Specific work out/(Heat added between SG and feed enthalpies)*

= *[(Enthalpy of saturated steam at 75bar – enthalpy steam at 0.04 bar)*75% - (Feed work + Station load)]/*

[Enthalpy saturated steam at 75bar

– Feed enthalpy at 140°C*, 75 bar]*

= *(2766-1750)*0.75-57)/(2766-600)*

= *34% (note generator efficiency not part of* <u>*thermal*</u> *efficiency)* *(ii) District heating thermal efficiency*

= *[(Enthalpy of saturated steam at 75bar -*

*- Enthalpy steam at 152℃, 5 bar)*84% - (Feed work+ Station loads)]/*

[Enthalpy saturated steam at 75bar – Feed enthalpy at 20℃, 75 bar]

= *(2766-2320)*0.84-57)/(2766-86-7)*

= *11.8%*

(iii) Revenue from conventional PWR system:

= *Generation MW per kg steam*per hour*unit price*

= *(2766-1750)*0.75*0.95-57)*60*60*80 = £192,067 per kg steam.*

Revenue from district heating:

*Power = (2766-2320)*0.84*0.95-57)*60*60*80 = £86,086 per kg steam*

Required revenue from heating = £105,981 per kg steam

*Heat from steam = (Steam enthalpy output from turbine – enthalpy of water returning from district heating system*per hour* district heating effectiveness =*

*= (2320-86)*3600*40% = 3,217 MWh per kg steam*

Hence heat price = £105,981/3,217 = £33/MWh

(iv) Increasing the temperature at which heat is taken from the system would require a higher sale price for the heat, all other things being equal and vice versa. However, these issues are interlinked. Higher pressure steam in district heating could make the system more effective and increasing the quantity of steam that is usable. However, it is probably wrong to equate energy at electricity, which has a high utility with district heat which can only be used for heating.

2. Core Temperatures

(a) Integrating with respect to radius

$$kr.\frac{dT}{dr} + q'''.\frac{r^2}{2} + C_1 = 0$$

At the centre of a fuel pellet: $dT/dr = 0 \quad at\, r = 0$

which leads to: $C_1 = 0$

Rearranging the above equation and integrating again:

$$\int_{Tmax}^{T} k.dT = \int_{r_V}^{r} q'''.\frac{r}{2}$$

or $$-\int_{Tmax}^{T} k.dT = \frac{q'''r^2}{4}$$

For: $r = r_{fo}$ $$\int_{T}^{T_{Max}} k.dT = \frac{q'''}{4}.r_{fo}^2$$

But the linear and the mean volume heating rates for a unit length are related by:

$$q' = \pi.r_{fo}^2.q'''$$

Hence $$\int_{T}^{T_{Max}} k.dT = \frac{q'}{4\pi}$$

or for constant k $$\boldsymbol{\Delta T = \frac{q'}{4\pi k}}$$

(b) The temperature drops through the gap, cladding and from the surface to the bulk fluid and are all proportional to the linear heating q'*. Assuming that for example the*

k(clad) ≠ function of T

and that the convective heat flux can be written in the form:

$$\varphi_{CONV} = Area * heat\ transfer\ coefficient * \Delta T$$

Therefore $$q' = \Delta T/(R_{gap} + R_{clad} + R_{surf}$$

Where $$R_{gap} = \frac{k}{2\pi}\ln\left(\frac{r_c}{r_f}\right); R_{surf} = \frac{h}{2\pi r_c}\ etc.$$

K is influenced by Reynolds Number and Prandtl Number of the flow and fluid respectively and by the thermal conductivity of the cladding and gas gap.

(c) (i)

SFEE $\dot{m}.C_p dT_c = q'(z).dr$

$$T_c = \frac{q'_{max}}{m\dot{C}_p}\int \cos(\beta z)dz + const$$

$$T_0 = T_c + q'(a + \frac{1}{4\pi k})$$

(ii) *Differentiating wrt z*

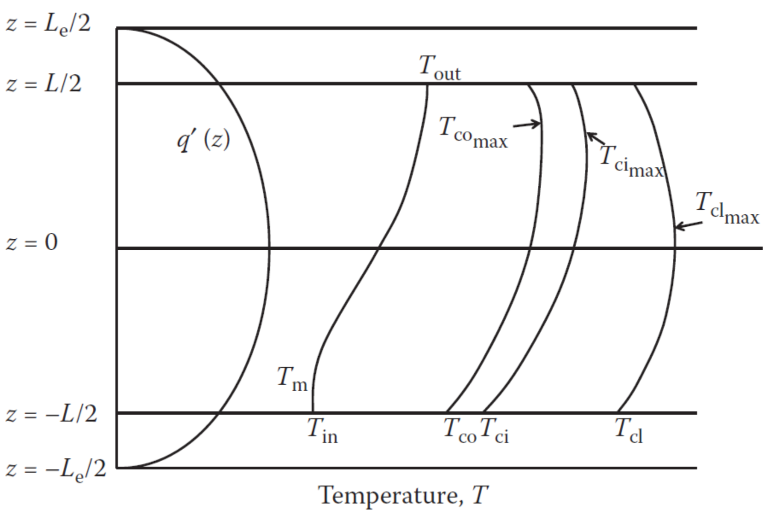
$$\frac{dT_0}{dz} = \frac{dT_c}{dz} + \frac{dq'}{dz}(a + \frac{1}{4\pi k}) \quad or \quad \frac{dT_0}{dz} = \frac{q'}{m\dot{C}_p} + \frac{dq'}{dz}(a + \frac{1}{4\pi k})$$

$$\frac{dT_0}{dz} = \frac{q'_{max}}{m\dot{C}_p}\cos(\beta z) - q'_{max}\beta.\sin(\beta z)(a + \frac{1}{4\pi k})$$

at T_0^{max} $\frac{dT_0}{dz} = 0$ $\frac{1}{\dot{m}C_p}\cos(\beta z) = \beta.\sin(\beta z)(a + \frac{1}{4\pi k})$

$$\tan(\beta z) = 1/[\beta\dot{m}C_p\left(a + \frac{1}{4\pi k}\right)]$$

(iii)



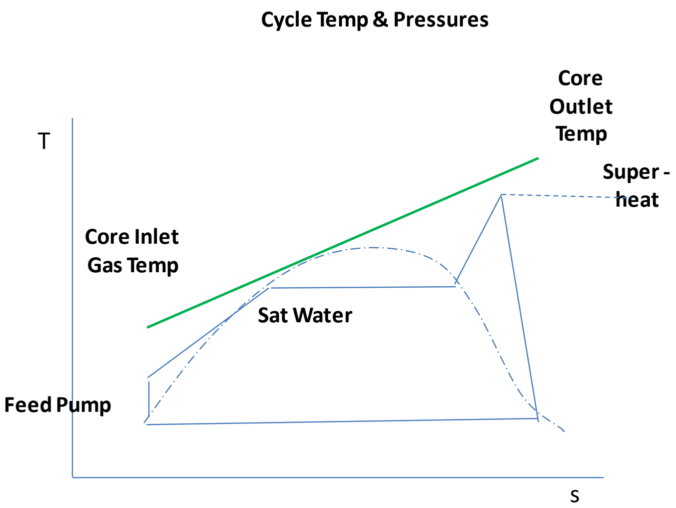
From: Todreas and Kazimi, Nuclear Systems – Volume I: Thermal Hydraulic Fundamentals, 2nd Ed.

3. Steam Generator Design

(a) *Indirect cycles have a heat exchange between the core and the power cycle. This is common for safety reasons, wishing to separate with a physical barrier the core which is or can be contaminated with radioactivity and the power cycle. It is an example of defence in depth. Also, such separation keeps the power equipment 'radioactively clean' and hence much simpler to maintain.*

The negative aspect is thermodynamic, which for a fixed maximum core temperature the heat supplied to the power cycle will always be at a lower temperature, with attendant cycle thermal efficiency loss.

(b) *(i) Flow and heat transfer conditions:*



1. *Gas-side - single phase convective heat transfer – convective heat transfer correlation*

2. *Economiser from feed to saturation– sub-cooled convective heat transfer correlation*

3. *Evaporator from saturated water to saturated steam – boiling sequence: film, bubbly, slug and then mist flow until saturated mixed steam – boiling heat transfer correlation.*

4. *Super-heater from saturated steam to maximum super-heater temperature - convective heat transfer correlation*

5. *Evaporator from saturated water to saturated steam – boiling sequence: film, bubbly, slug and then mist flow until saturated mixed steam – boiling heat transfer correlation.*

6. *Super-heater from saturated steam to maximum super-heater temperature - convective heat transfer correlation*

(ii) Tube lengths estimate:

$$\frac{1}{U} = \frac{1}{h_P} + \frac{1}{h_S} + \frac{t}{k}$$

*Core flow = Core Power/[Specific heat CO_2*Core temperature difference]*

*= 600*10^3/[1.1*(640-340] = 1.182 tne/s*

Gas flow per SG = 151.5 kg/s

Gas Reynolds Number based on Max velocity = 60,480

Prandtl Number = 0.75

Colbourn correlation Nu = 222

*Gas side heat mean transfer coefficient = 2010*55/28/10^3 kW/m^2 K*

= 436 W/m^2 K

Secondary heat transfer coefficients for economiser and evaporator are given and the tube conductivity coefficient is (linear approach) k/t = 12.3 kW/m^2 K for ferritic sections and 6.28 kW/m^2 K for austenitic sections.

We need to calculate the super-heater secondary heat transfer coefficient.

On the basis that the same flow pass through the economiser and super-heater and both are convective single phase heat transfer of the form of Dittus-Boetler, we can say:

$$\frac{Nu_{sh}}{Nu_{ec}} = \frac{Re_{sh}^{0.8}.Pr_{sh}^{0.33}}{Re_{ec}^{0.8}.Pr_{ec}^{0.33}} \quad or$$

$$\frac{h_{sh}}{h_{ec}} = \frac{k_{sh}}{k_{ec}} \cdot \left(\frac{\mu_{ec}}{\mu_{sh}}\right)^{0.8} \cdot \frac{Pr_{sh}^{0.33}}{Pr_{ec}^{0.33}}$$

Hence $\frac{h_{sh}}{h_{ec}} = 0.46$

	Mean T_w C	Mean T_g C	HTC pr W/m²/K	Wall W/m²/K	HTC sec W/m²/K	1/U	U	UdT	dH kW	A m²	L m
Econ.	257	395	436	12300	20000	0.0024	412	56910	18201	319.8	4155
Evap.	353	495	436	12300	50000	0.0024	418	59293	15304	258.1	3353
Superh.	462	591	436	6280	9200	0.0026	390	50361	16495	327.5	4255

(iii)

Adding re-heater will improve thermodynamic efficiency of the cycle for a number of reasons:

1. *Increase the average temperature of heat addition to the cycle without increasing the maximum temperature.*

2. *Allow higher steam generator pressure without increasing the steam quality at the turbine exhaust.*

3. *Allow higher steam flow rate through the low pressure turbine stages.*

Factors to be considered when sizing the re-heater:

1. *Additional pipework and penetrations to the reactor building necessary to transport the steam for re-heat and back to the low pressure turbine.*

2. *Given the total heat exchanger volume is limited, the re-heat stage may need to be introduced at the expense of other stages making the heat transfer less efficient (i.e. requiring larger delta-T to compensate for less heat transfer area).*

3. *The steam properties would be somewhat different at 40 bar impacting the heat transfer coefficient on the steam side of the heat exchanger and therefore the length of the tubes.*

4. LOCA Safety

(a) (i)

Tenets of safety:

1. ***Control** of reactor **power and reactivity***
2. *Ensuring the core is always **cooled***
3. *Ensuring that radioactivity is **contained***

(ii) Defence in depth: This is both conceptual and physical:

1. *Systems which preclude an accident – **inherent safety***
2. *Systems which control an incident – **automatic control***
3. *Systems which stop the operator from unsafe situation – **protection systems***

LOCA protection example:

1. *Keeping the core cladding cooled in LOCA by injecting water and removing heat ensures that the radioactivity is retained in the fuel and the clad. Pressuriser is the cooling first line of defence;*
2. *Core cooling systems initiated automatically by low pressure trip;*
3. *If high pressure control is lost because the pressuriser empties or/and a high pressure injection stops being effective, back-up systems fill the gap;*
4. *If high pressure systems not effective low pressure water injection systems operate – with redundancy to ensure high reliability;*
5. *In longer term water re-circulated to vessel from containment well to ensure cooling continues;*
6. *Third level of defence provided by leak-proof containment structure which ensures that radioactive contamination is not allowed to escape. Containment designed to accept the maximum pressure predicted both from decay heating and from any hydrogen burning. Long term containment cooling system provided.*

(iii) Short-term LOCA cooling:

Once the reactor is shut-down the power falls quickly below 2% of previous power – LOCA cooling in three conceptual phases:

1. *'**Blowdown'** phase in which pressure in the circuit falls quickly (~20s) and the coolant is ejected from the circuit exposing the core. At this stage the automatic LOCA systems (Emergency Core Cooling Systems – ECCS) start to become effective;*

2. *'**Refilling**' the primary circuit;*

3. ***'Re-flooding'** the core and re-wetting the clad. The highest clad temperatures are reached during this last phase.*

(iv) CANDU LOCA protection differences:

Many similarities in the approach to LOCA protection: pressuriser, high pressure injection, low pressure ECCS and large leak-tight containment. Initially CANDU has a small positive void coefficient and shut-down systems need to operate quickly to stop the nuclear reaction.

1. *Much lower power density of CANDU means that the accident proceeds more slowly and there is more time for coolant injection;*

2. *Pressure tubes are surrounded by large tank of cool heavy water which can acts a heat sink during such and accident;*

3. *Fuel is in horizontal pressure tubes rather than a closed end vessel. Therefore either in a single pressure tube, or across the range of tubes there is the possibility for stratification such that some voidage uncovers part of the channel fuel or the core.*

(i) Containment cooling reaches equilibrium when decay heat equals the heat removal rate 5.5MW.

Decay heat expression from NE data book:

Decay Heat (Beta & Gamma): $P(t) = 0.0622P_0.\left(t^{-0.2} - (t+t_0)^{-0.2}\right)$

For: t_0= 3 year (in seconds) and t= 20 days (in seconds

*P(t) = 0.00194 P_0 = 0.00194*1000/0.35 MW = **5.53MW***

(ii) Volume of stored water to absorb energy during 20 days

Final enthalpy based on 50% saturated at 2.5 bar 127°C (535 kJ/kg) and 50% saturated steam at 2,5bar (2,716 kJ/kg). Initial enthalpy: 84 kJ/kg.

Evaluate the energy generated, by integrating decay heat equation.

$$G(t) = 0.0622.P_0 * 1/0.8\,(t^{0.8} - (t+t_0)^{0.8} + t_0^{0.8})$$

G(20days in seconds) = 13,924 GJ

Mass of stored water = 13,924,000/(2,716/2+535/2-84) tne

*= **9,033 tne***