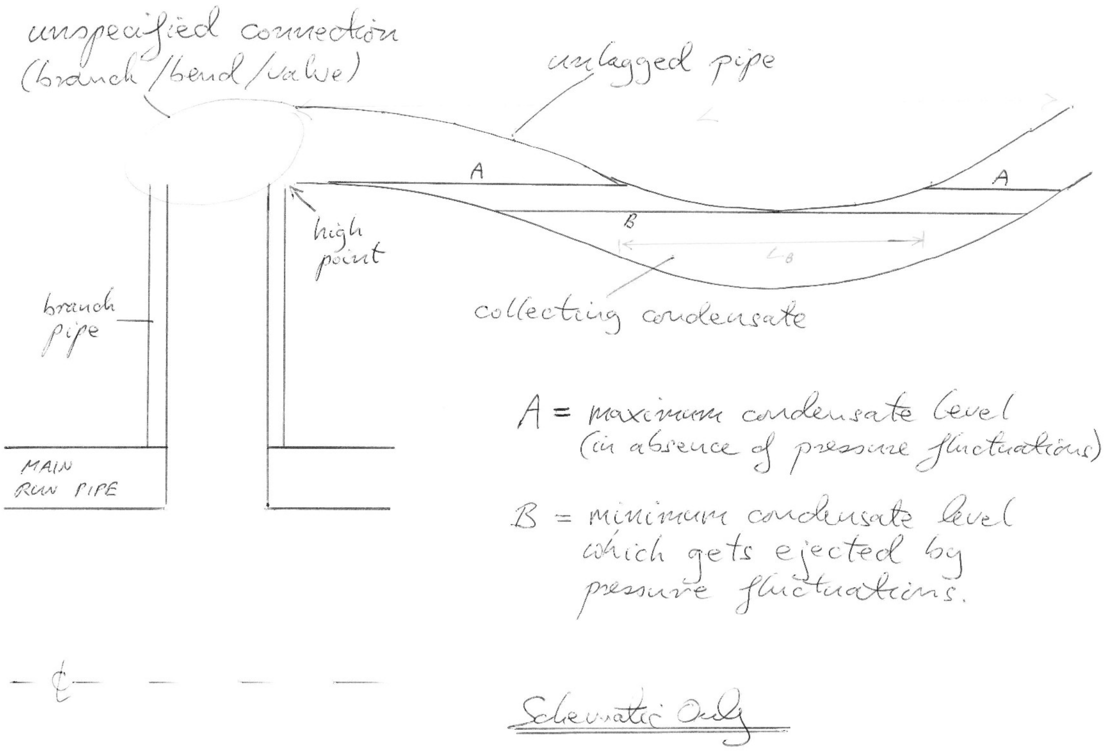
Crude Model for Small Bore Thermal Fatigue Cracking/Condensate Refluxing (Illustrated for a Hot Reheat System)

1. Model

The system consists of a horizontal main run pipe with a vertical branch-nozzle at the TDC of the main pipe. An unlagged pipe is connected to the branch-nozzle in some unspecified manner, e.g., via a branch, a bend or a valve. This will be called the "sample pipe". This is the pipe in which the condensate forms. It is assumed to be roughly horizontal, but has a low point so that condensate can collect (see sketch).



What causes the condensate to reflux (i.e., to fall back down the branch-nozzle)?

If the pressure in both the main pipe and the far end of the sample pipe were strictly steady, the condensate would rise to level A on the sketch. Refluxing would then consist merely of individual drops trickling over the high point. These would be unlikely to reach the level of the main pipe and this situation would seem to be relatively benign.

The mechanism by which the condensate falls back down the branch-nozzle is assumed to be random pressure fluctuations causing slight pressure differences between the main pipe and the far end of the sample pipe. Note that just a few inches of water head is likely to be sufficient, so the fluctuations required are extremely small (4 inches water head = 0.001MPa = 0.15psi). However, if the condensate level is below level B, these pressure fluctuations will merely move steam around the sample pipe, over the top of the condensate surface. Only when the condensate reaches level B will it form a 'plug' which can get ejected by the pressure fluctuations.

Assuming that the required pressure fluctuations occur on a periodic timescale short compared with that required to form the condensate, it follows that each reflux event consists of a volume of condensate equal to that at level B.

2. Temperature of Condensate

Steam in the sample pipe will condense to water phase when its temperature drops to saturation at the relevant pressure. Here we assume that the main run pipe is a Torness hot reheat pipe, and hence that the main pipe steam conditions are typically,

Pressure 3.73 MPa; Temperature 516^{o}C

The saturation temperature at 3.73MPa is 246^{o}C, so this is the temperature at which the condensate forms.

For the purposes of the model, the condensate is assumed not to cool further once it attains water phase. This is rather arbitrary but avoids a difficult two-phase thermal transient problem.

3. Symbol Notation

t_m	thickness of main pipe
d_v	ID of branch nozzle
D_h	OD of roughly horizontal sample pipe
L	Length of sample pipe contributing to condensation
L_B	Mean length of condensate column when at level B (see sketch)
ρ_{air}	Density of air under room conditions, 1.2 kgm^{-3}
μ	Viscosity of air under room conditions, 1.8 x 10^{-5} kgm^{-1}s^{-1}
$\nu = \dfrac{\mu}{\rho_{air}}$	Kinematic viscosity of air under room conditions, 1.5 x 10^{-5} m^{2}s^{-1}
C_p^{air}	Specific heat of air under room conditions, 1.01 x 10^3 Jkg^{-1o}C^{-1}
k_{air}	Thermal conductivity of air under room conditions, 0.0256 Wm^{-1o}C^{-1}
$\Pr = \dfrac{\mu C_p^{air}}{k_{air}}$	Prandtl number of air under room conditions, 0.71
T_s	Absolute temperature of steam in main pipe, 516 + 273 = 789 K
T_c	Absolute temperature of condensate, 246 + 273 = 519 K
T_a	Absolute air temperature, 20 + 273 = 293 K
$Gr = \dfrac{g\left(\dfrac{T_c}{T_a} - 1\right)D_h^3}{\nu^2}$	Grashof number for sample pipe in air (see 4.1)
h	Heat transfer coefficient between sample pipe and air
$q = h(T_c - T_a)$	Heat flux from sample pipe to air

$Nu = \dfrac{hD_h}{k_{air}}$ Nusselt number for sample pipe in air

C_p^{steam} Specific heat of steam, Jkg^{-1o}C^{-1}

LH Latent heat (steam-to-water), Jkg^{-1}

ρ_w Density of water, 1000 kgm^{-3}

k_{steel} Thermal conductivity of main run pipe steel (CMV), 36 to 42 Wm^{-1o}C^{-1}

ρ_{steel} Density of steel, 7860 kgm^{-3}

C_p^{steel} Specific heat of steel, 545 to 670 Jkg^{-1o}C^{-1} at temperatures T_c to T_s

$\kappa = \dfrac{k_{steel}}{\rho_{steel} C_p^{steel}}$ Thermal diffusivity of steel, 6.8 x 10^{-6} to 9.8 x 10^{-6} m^{2}s^{-1}

τ Duration of each quench event, seconds

δ Radial depth of material from bore of hole in main affected by quenches, m

σ_{hoop} Hoop stress at bore of hole in main, wrt hole centre, MPa

E Young's modulus of CMV, average between T_c and T_s, 0.18 x 10^6 MPa

α Coefficient of thermal expansion of CMV, average between T_c and T_s, 13.6 x 10^{-6} ^{o}C^{-1}

4. Rate of Condensate Formation

4.1 Free Convective Cooling of Horizontal Pipe in Air

For illustration we shall assume D_h = 33mm. The maximum Grashof number, assuming the wall temperature could be as high as T_s is thus,

$$Gr = 9.81 \times [(789/293) - 1] \times 0.033^3 / (1.5 \times 10^{-5})^2 = 2.7 \times 10^6 \quad (1)$$

However, the sample pipe wall will not be as hot as T_s. A better guestimate is that the wall will be close to T_c, which gives a Grashof number,

$$Gr = 9.81 \times [(519/293) - 1] \times 0.033^3 / (1.5 \times 10^{-5})^2 = 1.2 \times 10^6 \quad (2)$$

Laminar conditions prevail since $\Pr Gr < 10^9$. A correlation for the Nusselt number in terms of the Prandtl number and the Grashof number is given in Rohsenow and Hartnett, Section 6, Equ.(47), i.e.,

(Laminar conditions) $$Nu = 0.47 (\Pr Gr)^{0.25} = 14.3 \quad (3)$$

The heat transfer coefficient between the sample pipe OD and the air is given by,

$$h = Nu \frac{k_{air}}{D_h} = 14.3 \frac{0.0256}{0.033} = 11 \text{ Wm}^{-2o}\text{C}^{-1} \quad (4)$$

The length of sample pipe involved in the cooling is L, and hence the cooling surface area is $\pi D_h L$. The driving temperature difference between the pipe wall and the air could be as high as $T_s - T_a$ but is likely to be closer to $T_c - T_a$. Assuming the latter, the rate of heat loss from the sample pipe is,

$$\text{Rate of Heat Loss} = \pi D_h L \times h(T_c - T_a) \qquad (5)$$

4.2 Condensate-Reflux Periodic Time

Each pressure pulse is assumed to clear the sample pipe of condensate. The time between reflux events is therefore the time taken to form the depth B of condensate. This is a volume $\frac{\pi}{4} D_h^2 L_B$ of condensate (ignoring the difference between the pipe ID and OD). The amount of heat which must be extracted to condense this volume of water is,

$$\text{Heat Lost} = \frac{\pi}{4} D_h^2 L_B \rho_w \left[C_p^{steam} (T_s - T_c) + LH \right] \qquad (6)$$

The first term in […] is the heat loss required to cool the steam initially at temperature T_s to the temperature T_c, whilst remaining as steam phase. The second term, *LH*, is the latent heat which must be removed to condense the cooled steam. The term $C_p^{steam}(T_s - T_c)$ can be equated with the internal energy change between 516^{o}C and 246^{o}C and at ~40 Bar from steam tables, i.e., 2050 kJ/kg. Similarly the latent heat is found to be 1550 kJ/kg.

Dividing (6) by (5) provides the time between reflux events,

$$\text{Periodic Time} = \frac{L_B}{L} \cdot \frac{D_h \rho_w \left[C_p^{steam} (T_s - T_c) + LH \right]}{4h(T_c - T_a)} = 12{,}000 \frac{L_B}{L} \text{ seconds} \qquad (7)$$

The ratio $\frac{L_B}{L}$ depends upon the detailed geometry of the sample pipe. It is the fraction of the length of the sample pipe that becomes filled with condensate at level B. For example, if this were 10%, then the periodic time would be 1,200 seconds, or **<u>20 minutes</u>**.

However, even if the upper bound of $\frac{L_B}{L}$ = 1 is assumed, the periodic time between quench events cannot be longer than 12,000 seconds ~ **<u>3.3 hours</u>**.

This upper bound quenching interval scales with sample pipe diameter, D_h, but is otherwise independent of the details of the pipework. It depends only on the steam temperature, T_s, and the steam pressure, because the latter determines T_c.

5. Severity of Thermal Shock to Main Pipe

5.1 What Parameter Controls the Cooling?

It is extremely difficult to judge how much heat is pulled from the main pipe metal on each reflux event. It is possible that, for a long condensate column, most of the condensate merely enters the main steam flow before evaporating. On the other hand, for very small amounts of condensate reflux, the condensate may flash off to steam within the branch nozzle before it ever reaches the main pipe.

An upper bound to the cooling effect on the main pipe is to assume the whole of the condensate column is evaporated by heat pulled from the main pipe. It is simple to show that this upper bound cannot possibly be achieved, as follows.

The main pipe would need to supply at least the heat required to change the phase of the condensate back to steam. Hence, the heat pulled from the main pipe per reflux quench would need to be at least the latent heat part of (6), i.e.,

$$\text{Heat Required from Main} = \frac{\pi}{4} D_h^2 L_B \rho_w LH = 1.3 \times 10^6 L_B \ \text{ Joules} \tag{8}$$

where L_B is in metres. Assuming the bore of the hole falls from T_s to T_c , and that the full temperature T_s is regained at a radial distance δ from the bore, the heat lost assuming a linear gradient is,

$$\pi d_v t_m \delta \rho_{steel} C_p^{steel} \left(\frac{T_s - T_c}{2} \right) = 2.2 \times 10^6 \delta \tag{9}$$

In evaluating (9) we have assumed $d_v = t_m = D_h = 33mm$ for illustration. Equating (8) and (9) requires that,

$$\delta \approx 0.6 L_B \tag{10}$$

Assuming that L_B is a substantial fraction of a meter, at least, it is not credible that the cooling penetration depth could be so deep as implied by (10). The quench event can only last a few seconds at most. The cooling depth is given roughly by $\sqrt{3\kappa\tau}$, where τ is the duration of the event. Since the thermal diffusivity is not more than ~10^{-5} m^2s^{-1}, even a relatively long duration event of ~10 seconds can only affect material to a depth of $\sqrt{3 \times 10^{-5} \times 10} = 17$ mm. A better guess might be an event duration of ~1 second, with a penetration depth of 5 mm.

It follows that the cooling effect on the main pipe is controlled by the duration of the quench event. The bulk of the energy required to evaporate the condensate will come directly from the steam, not the steel.

5.2 Estimate of Transient Temperature Distribution

Reversing the above observation, we shall assume an event duration, τ , and hence derive a penetration depth of $\delta \approx \sqrt{3\kappa\tau}$. The transient temperature distribution will be assumed to be T_c at the bore of the hole in the main pipe, rising to the steady operating value of T_s over a radial distance of δ .

The advantage of this is that the thermal stress is independent of the unknown condensate column length, L_B . On the other hand, the duration of the quench events, τ , is also unknown. However, it is physically clear that this cannot be more than a few seconds, and may only be a fraction of a second. Hence, for τ between 0.3 and 3 seconds we have $\delta = 2.5$mm to 10mm. Because of the assumption that the surface temperature is pulled down to T_c , the shorter the assumed event duration, the steeper is the resulting temperature gradient.

However, remarkably, this does not matter anyway…

5.3 Peak Thermal Transient Stress

Rather remarkably, the thermal stress does not depend upon the temperature gradient in this case. The analytic solution for an axisymmetric temperature change around a hole in a plate[1] gives the hoop stress at the hole to be simply $\sigma_{hoop} = -\frac{E\alpha\Delta T_0}{\xi}$, where ΔT_0 is the temperature change at the bore of the hole, and $\xi = 1$ for plane stress or $\xi = 1-\nu$ in plane strain. This result applies whatever the radial temperature distribution, provided only that the plate is large and the temperature at great distances is unchanged. These conditions are appropriate for our problem, provided we approximate the main pipe in the vicinity of the branch as a flat plate. Near the ID of the main pipe the conditions are plane stress, so the peak *elastic* thermal stress is expected to be well approximated by,

$$\sigma_{hoop} = -E\alpha(T_c - T_s) = +670\,\text{MPa}$$

which is obviously well past yield. It is assumed that there is no reverse cycling, so that this is also the elastic stress range. At the bore of the hole, and on the ID, the other two principal stresses are zero. So the stressing is locally uniaxial.

6. Crude Scoping Creep-Fatigue Initiation Assessment

I'm not pretending this is a complete R5V2/3 assessment, but some salient scoping estimates are…

6.1 Shakedown?

R66 Rev.008 gives $\sigma_{0.2}^{LB} = 191$ MPa; $\sigma_{0.2}^{BE} = 250$ MPa. Since K_S for ferritics is <1, it follows that the structure is outside strict shakedown even if best estimate data were used. In fact, this would be the case even for K_S = 1. Hence, realistically we expect the creep-fatigue mechanism to be active, with plastic cycling at the bore of the hole.

I can't be bothered with global shakedown right now.

6.2 Neuberisation

$(\Delta\sigma\Delta\varepsilon)_{el} = 670^2 / 0.18\times10^6 = 2.44\,\text{MPa}$.

R66 Table 8.2 gives a wide range of cyclic stress-strain fits (all at 550°C, which is assumed sufficiently close). The extremes of the Ramberg-Osgood parameters are,

(a) A = 5949 MPa; β = 0.365

(b) A = 556 MPa; β = 0.128

The latter is quite severely aged, so may be too onerous. On the other hand R5V2/3 Appendix A1, Section A1.5.3.1 suggests that quite modest amounts of ageing may lead to ~50% increase in effective strain range. So fit (b) may actually be the more accurate.

Applying Neuber gives,

(a) $\Delta\sigma_{ep} = 549\,\text{MPa}$; $\Delta\varepsilon_{ep} = 0.44\%$

(b) $\Delta\sigma_{ep} = 293\,\text{MPa}$; $\Delta\varepsilon_{ep} = 0.83\%$

---

[1] See http://rickbradford.co.uk/HoleInPlateThermalStress.pdf

6.3 Fatigue Damage

R66 Rev.008 Table 9.2 gives the coefficients for the Langer equation fit to fatigue endurance. The <u>best estimate</u> endurance is,

$$N_f^{BE} = \left(\frac{\Delta\varepsilon - 0.212}{115.07} \right)^{-1/0.711}$$

Hence, (a) $\Delta\varepsilon_{ep} = 0.44\%$ gives $N_f^{BE} = 6334$ cycles, and (b) $\Delta\varepsilon_{ep} = 0.83\%$ gives $N_f^{BE} = 1558$ cycles. I haven't bothered to evaluate the lower bound endurances, as would normally be done, since the best estimates are clearly going to predict crack initiation.

6.4 Dwell Stress

$(K_S S_y)_{246^o C} = 218$ MPa; $(K_S S_y)_{516^o C} = 172$ MPa; So the centre of the hysteresis cycle is shifted to (218 - 172) / 2 = 23 MPa. The dwell stress is thus,

(a) $\Delta\sigma_{ep} = 549$ MPa; $\sigma_{Dwell} = -549/2 + 23 = -252$ MPa

(b) $\Delta\sigma_{ep} = 293$ MPa; $\sigma_{Dwell} = -293/2 + 23 = -124$ MPa

The dwell occurs at compressive stress so the creep damage will be smaller than the tensile dwell case. It has been ignored here.

6.5 Crack Initiation Times

Even if we assume the longest periodic time between quenches, i.e., 3.3 hours, and ignore creep damage, the <u>best estimate</u> fatigue endurance is either 6334 x 3.3 = 21,000 hours (case a), or 1558 x 3.3 = 5,200 hours (case b). So, a realistic estimate is that crack initiation is to be expected in roughly <u>**1 to 3 years**</u>.

The remarkable thing is that this prediction should apply, roughly, to any hot reheat branch subject to refluxing, since the result is not sensitive to the details of the particular instance – depending primarily just on the system temperature and pressure.

(However the initiation time is predicted to be proportional to the diameter of the sampling pipe, due to (7) – so smaller bore pipes are worse).