## INTRODUCTION

## DESCRIPTION OF THE MODEL

### *THE FIREBOX*

It is assumed that the rate of combustion is limited only by the air supply, that the products of combustion leave at a (variable) specified fraction of the temperature of the fire bed, and that combustion is completed in the fire bed. Excess air can be admitted over and above that required for stoichiometric combustion. Given these assumptions it is possible to calculate the fire temperature and the rate at which heat is radiated from the fire to the firebox roof and walls. Free convection from the combustion gases will transfer further heat to the firebox walls and will reduce the temperature of the flue gases as they enter the fire tubes. The process is shown diagrammatically in Figure 1.

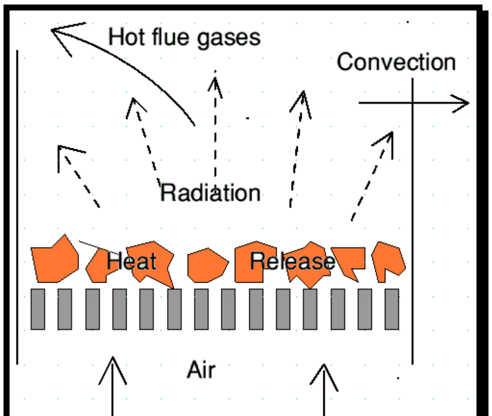

Figure 1

A heat balance on the fire gives:

$$m_a.h_a + cv \,.\, rate \;=\; m_f.h_f + 56.6\ 10^{-12}.\ T^4 \;\ldots\ldots\; [1]$$

where
$m_a$ = mass flow rate of air
$m_f$ = mass flow rate of flue gas
$h_a$ = enthalpy of air into grate
$h_f$ = enthalpy of flue gas
cv = calorific value of fuel
rate = rate of combustion
T = absolute temperature of fire

The calculation is simplified somewhat by reckoning the flue gas enthalpy from the air inlet temperature (293 K), thus eliminating the air enthalpy from the equation, and by assuming a specific heat of 1 kJ/kg K for the flue gas. The mass flow of the flue gas is taken as the stoichiometric mass plus a percentage of excess air. The calorific value, combustion rate and excess air are specified and equation [1] is solved for T using Newton-Raphson iteration.

In reality combustion will not be completed in the fire bed, and the fire thickness and the volume and conditions in the firebox may well be important in determining to what extent combustion continues before being quenched in the firetubes. It is unlikely that the flue gas will be in equilibrium with the fire temperature, particularly at high combustion rates. There is clearly scope for improving the model in this area, and relevant experimental data would be particularly valuable. ]

The Rayleigh Number for conditions in the firebox is then calculated, and the free convection heat transfer coefficient estimated.  A heat balance on the combustion gases then gives the flue gas temperature at entrance to the fire tubes.

## THE FIRETUBES

The assumption is made that the heat transfer from the tubes to the water is so effective by comparison with that from the flue gas to the tubes that the tube wall is effectively at the temperature of the boiling water (i.e. the saturation temperature Tsat at the pressure in the boiler. This assumption could be relaxed, but is thought to be sufficiently accurate at the present stage of development. The heat flux (i.e the heat flow rate per unit area of tube) is thus dependent on the heat transfer coefficient from flue gas to tube, and the temperature difference between them. Both of these quantities vary along the tube, so it is necessary to carry out an integration along its length. The heat transfer coefficient is evaluated at intervals equal to the diameter of the tube, and the gas temperature is calculated by integrating the heat flux and applying a heat balance.

An important preliminary step is the determination of the type of flow in the tubes, as determined by the Reynolds Number. It so happens that the flow is usually turbulent in full size, and usually non-turbulent (or laminar) in small scale locomotives. Heat transfer is radically different for the two types, so it is important to model this aspect correctly. The difficulty is that for some locomotives at some operating conditions the flow may be in a transition region between the two types - a region in which heat transfer is difficult to predict with accuracy. A linear interpolation is used to connect the ends of the two regimes.

With laminar flow at moderately high Reynolds Number the heat transfer coefficient is greatly dependent on distance along the tube; a formulation is used in which heat transfer coefficient depends on the local Reynolds No. and the ratio x/d, where x is the distance along the pipe and d is the pipe diameter. This formulation has been fitted to data of Kays [1] for heat transfer in the combined thermal and hydrodynamic entry region of a pipe.

Turbulent heat transfer is also enhanced in the entry region of a pipe, but in this case the effect persists only a few diameters into the pipe. It is therefore satisfactory to apply a fixed enhancement factor which decays after about 9 diameters.[2]

The heat transfer formulations will be found in the procedure PROCtubes. It is sufficient here to note that they lead to values of the Stanton Number distribution along the firetube. The Stanton Number is a dimensionless group that, together with the Reynolds No. is especially convenient in convective heat transfer: they are defined as follows:

$$Re = qdr/m \qquad\qquad St = h/(qrc)$$

where
- q = mean flow velocity
- d = pipe diameter
- r = mean density of fluid
- m = viscosity of fluid
- h = heat transfer coefficient
- c = specific heat of fluid

However, the usefulness of the Stanton No. is apparent when interpreted as follows:

$$St = (dT/DT) . (d/4dx)$$

where $dT$ = change in mean fluid temperature in distance $dx$
$DT$ = temperature difference between pipe wall and fluid

Thus it will be seen (PROCtubes) that the decrement of flue gas temperature dT in a distance equal to the tube diameter is simply:

$$dT = 4 . St . DT = 4 . St . (T-Tsat) \qquad [2]$$

Equation [2] can be integrated numerically along the firetube, thus giving the gas outlet temperature. From this, the heat transferred to the boiling water (in addition to that contributed by the firebox) can be evaluated. The boiler efficiency (i.e. the proportion of the theoretical heat released from the fire that is effective in producing steam) then follows.

Fig.2 shows diagramatically the way in which the flue gas temperature varies along the firetubes. Obviously the heat transfer rate is highest towards the tube inlet because both the heat transfer coefficient and the temperature difference are greatest there.

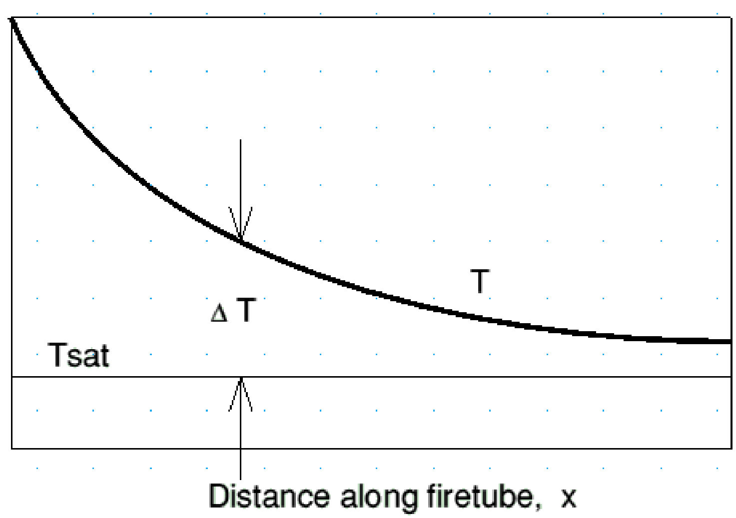


Figure 2

Similar considerations apply to the calculation of pressure drop along the tubes as to heat transfer; in fact the two processes are physically very similar [3]. Again, the formulation depends on whether the flow is laminar or turbulent. There exists a quantity, the friction factor f that is analogous to the Stanton No.; it also is a function of the Reynolds No. and the distance from the pipe inlet. The formulations used in PROCtubes have been fitted to the data of Langhaar [4] for laminar , and Blasius [5] for turbulent flow. An allowance has also been made for the pressure drop through the firebed using data for packed columns [6]

[ The 'REPEAT' loop at the start of PROCtubes steps down the firetube in increments equal to one diameter and calculates the decrease in flue gas temperature. At each point the difference in temperature between the flue gas (T), and the tube wall (Tsat) is calculated, and the Stanton No. is found by a call to PROCstanton. The latter decides whether the flow is laminar or turbulent and in turn calls the appropriate function (FNst_lam or FNst_tur ). A rather minor correction for changes in viscosity along the tube has also been included.

In the case of the pressure drop it is unnecessary to integrate along the tube, and average values of density and friction factor can be used. In the case of both the Stanton No. and the friction factor, polynomials were fitted to the data of references 1, 2 and 4 to facilitate the calculation.

As a matter of interest the values of Reynolds No., Stanton No. and gas temperature T along the tube are stored in an array so that they can be recalled and displayed if reqiured. Otherwise the output from the procedure is the outlet gas temperature (from which can be calculated the heat input to the boiler) and the pressure drop. ]

## BLASTPIPE AND CHIMNEY

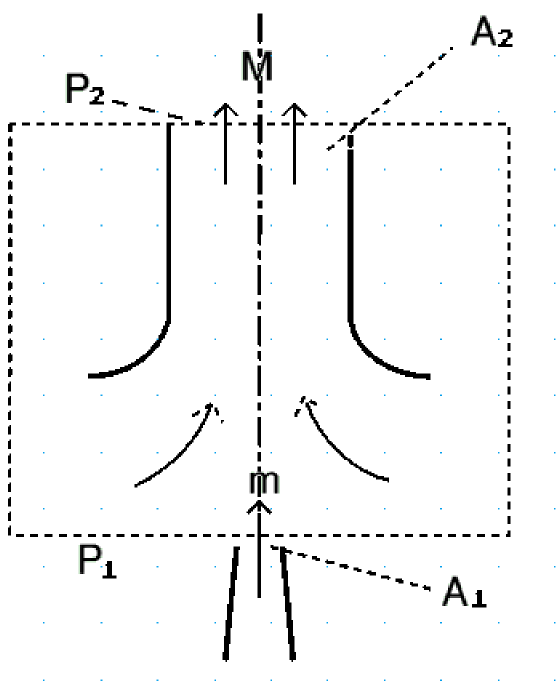


Figure 3

For a given pressure rise through the chimney (equal to the drop through the boiler) and a given mass flow M of flue gas, we need to be able to calculate the necessary mass flow of steam through the blast pipe. The blast pipe / chimney combination form a simple ejector, and its performance can be estimated using the Momentum Theorem. (Briefly this equates the pressure force acting on a control volume -á the large dotted rectangle in Fig. 3 - to the momentum flux through the control volume). Imagine the control volume to be entirely within the smokebox. Then

The Momentum Theorem can then be written in the form of a quadratic in m ;

[ The procedure PROCblast solves the above quadratic for m, the steam flow required to produce the necessary draught ( called Mstreq in the programme). Since the amount of steam produced is Mst, the excess (Mst - Mstreq) must flow through the safety valves. This safety valve flow is, of course, an indication of how well balanced are the heat transfer and draughting arrangements

The model used to solve the blastpipe and chimney ejector is somewhat idealised, and would benefit greatly from experiment. Important points to investigate would be the effect of pulsating flow, the use of a diffuser (i.e. an increasing area in the flow direction) in the chimney, and the effect of the distance between the blast nozzle and the skirt of the chimney. For the momentun theorem to be reasonably applicable the spread of the jet from the blast nozzle would have to be reasonably within the diameter of the chimney. Whilst it might be necessary to adjust the model in the light of experimental results, it is probably of the correct form.]

*(A new model of the performance of the 'front end' has been set up, based on data for the entrainment of fluid by a turbulent jet. This is a more satisfactory model, but gives results that are fairly close to the simple model outline above. It is described in a separate note entitled 'Front End Design', 1995. I am hoping to get some experimental data using an air driven test rig.)*

W.B.Hall, April 1991.

REFERENCES

1. W.M.Kays, "Convective heat and mass transfer" p143, Table 8-10. Mc.Graw Hill, 1966

2. Ibid. p195, Fig. 9-14

3 Osborne Reynolds, "On the Extent and Action of the Heating Surface for Steam Boilers" . Proc. Manchester Literary and Philosophical Society, 1874.

4. W.M.Kays, "Convective heat and mass transfer. p62. Mc.Graw Hill.1966

5 Ibid. p73, Eqn.6-44

6. Rohsenow & Choi "Heat, Mass and Momentum Transfer" p80, Prentice Hall, 1961. See also Mc.Adams : 'Heat Transmission" p125, Mc.Graw Hill , 1942.

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