'FRONT END' DESIGN

W B Hall, FREng., F.I.Mech.E

Introduction

The term 'Front end' applied to a steam locomotive refers to the blast pipe, petticoat and chimney, contained in the smokebox. Its design is crucial if a balance is to be maintained between steam usage in the cylinders and steam production in the boiler: as the steam flows faster through the blastpipe, so the vacuum in the smokebox increases, causing a greater air flow through the fire and a higher steaming rate. The problem has a long history: it is said that George Stephenson, in an attempt to quieten the exhaust of one of his engines, turned it into the chimney and thereby greatly improved the steaming capacity of the boiler!

One of the earliest investigations into front end performance was that by Zeuner in 1863 [1] using a small scale model (nozzle diameter $\approx$ 10 mm). This and other work carried out in the period up to the 1920s is reviewed in reference [3]. Prominent amongst this work is the study by Goss of Purdue University [2] , starting in 1896 in collaboration with the American Railway Master Mechanics Association; full size locomotives were used, and the study had a considerable influence on locomotive design in the United States.

Perhaps the most comprehensive collection of data is contained in a report entitled "A Study of the Locomotive Front End" describing work carried out in the 1930s at the University of Illinois [3]. This presents experimental data for a range of designs of blastpipe and chimney using a 1/4 scale laboratory model, and it includes a useful review of earlier work. A particularly interesting conclusion (also confirmed by Goss) was that the pulsation in the flow through the blastpipe had a relatively insignificant effect on front end performance. It is perhaps too much to expect this conclusion to be universally valid, but it offers a welcome simplification for initial experiments, and indeed for theoretical models.

Experiment or Theory ?

Undoubtedly much of the early development of the front end was a matter of trial and error, but the above references indicate the importance that was attached to systematic experiment even in the early period of locomotive design. Unfortunately there was no satisfactory background of theory to guide the analysis and generalisation of the results, so that data for a particular design could not be related to those for other designs. Some of the above studies were accompanied by theoretical models which were based on a thermodynamic analysis of the process. In fact, the thermodynamic efficiency of the front end is so low that this approach does not provide a useful framework for a theoretical analysis.

A more fruitful approach is to base the analysis on fluid mechanics (i.e. to use a momentum balance rather than an energy balance), and with the advent of the digital computer it is now possible to link a fluid mechanics analysis of the front end with the boiler to which it is attached. Even so, the complexity of the problem is such that a purely theoretical approach needs to be carefully tested against experiment. Why bother with theory then, if experiment is still required? There is in fact a very good reason: a theoretical analysis indicates the way in which the quantities such as draught, chimney and blastpipe dimensions, steam flow etc. are related to one another, so that experimental data can be fitted to equations having the correct form. This means that one can cover a much wider range of the design parameters with a given set of data. The following analysis will show for example that the pressure in the blastpipe and the smokebox draught are quite simply related, and that the ratio of steam to gas flow in the front end should be related to the ratio of blastpipe nozzle diameter to chimney diameter.

The analysis is purely theoretical, although some of the elements (such as the 'turbulent free jet') have been thoroughly tested elsewhere, and the theory confirmed. The only other theoretical concept used is the so-called 'Momentum Theorem', which follows from the application of Newton's Laws of Motion to a fluid flow system. This states that the change in momentum of a fluid flowing through a system (such as a blastpipe, smokebox, chimney combination) must be balanced against the force acting on the fluid: we shall use this to estimate the draught produced by the interaction of the steam and gas jet with the chimney.

An 'Entrainment' model

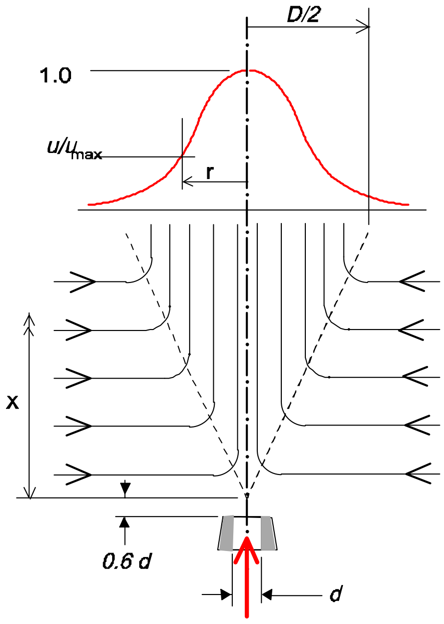


Fig. 1

The mechanism by which the flow through the blastpipe induces a flow of air through the boiler is best understood in terms of the 'free turbulent jet'. Many measurements have been made on jets of fluid emerging into a large volume of fluid, and in most situations of interest in engineering the flow in such jets is turbulent. The fluid surrounding the jet is entrained by turbulent mixing, and it transpires that a rather simple model of turbulence adequately describes the flow in the entrainment region - a region that is at uniform pressure. A description of the theoretical model and its supporting experimental data is given in the Appendix. (See also Schlichting[4], Townsend[5], Daily & Harlman[6].) For our purpose the following abstraction will suffice.

Fig.1 illustrates the flow into the jet, and the distribution of velocity across it. The flow pattern is such that the velocity u_{max} at the axis decreases linearly in the flow direction. The distribution of 'reduced' velocity u/u_{max} across the jet remains the same shape at all sections along the jet; consequently the jet is conical in form, and the angle of divergence can be determined if the velocity distribution is 'chopped off' at some radius where the velocity is negligible (the theoretical distribution extends out to infinity!). Since the pressure field is uniform, the momentum of the jet in its flow direction remains constant and equal to the momentum of the fluid issuing from the nozzle. This being the case one can specify the entrainment process entirely in terms of the diameter of the nozzle and the velocity of the fluid issuing from it.

(Flow very close to the nozzle does not conform to the above pattern because the velocity distribution at the nozzle is uniform and has a finite width; however, this fact may be allowed for by defining a fictitious origin for the jet at a point a distance 0.6d downstream of the nozzle, where d is the nozzle diameter. After a further distance of about 6.0d, the jet will be fully developed).

The volume of fluid forming the jet increases in proportion to the distance x from the jet origin, fluid being drawn in by the turbulent mixing process. This is consistent with the linear decrease in velocity because the cross sectional area of the jet increases as x^2; thus the product (which is equal to the volume flow) increases like x^2/x, or x. The rate of increase of the volume flowrate, V, in terms of the volume flowrate through the nozzle, V_0 , is given by the equation:

$$V/V_0 = 0.44\,(x/d) \qquad (1)$$

(This is not valid close to the nozzle outlet, but is applicable for values of x greater than about 5*d*)

Application to the Front End

The first point to make is that in a locomotive the jet is steam whilst the entrained fluid is composed of flue gases, and moreover the two are at different temperatures. The steam condition in the jet will depend upon the steam chest conditions and the efficiency of the engine. However, the thermodynamic efficiency of a non-condensing steam engine is so low that variations in its value will not have a great effect. Typically, the steam will be slightly superheated or slightly wet (i.e. at around 100°C) whilst the flue gases may be at a few hundred degrees. This temperature difference will tend to compensate for the difference in molecular weight of the two streams, and as a crude approximation they will be assumed to be of the same density.

The second point is that there is a more or less fixed relationship between the flowrates of the two streams which is determined by the combustion process. This will of course depend upon the fuel, the effectiveness of the combustion, and in particular the amount of excess air drawn into the firebox. A typical ratio of the mass flow of flue gas to the mass flow of steam (corresponding to 30% excess air) is around 1.8.

The jet velocity distribution shown in Fig.1 is in theory infinitely wide, but we are concerned with the central region which accounts for most of the momentum in the flow. In the Appendix it is shown that if the velocity distribution is truncated at a value of u/u_{max} = 0.10, the remaining core contains over 99% of the jet momentum. The corresponding included angle of the jet is about 21°, and Equation 1 should be modified as shown below (taking account also of the assumption of equal density of the steam and gas flows)

$$V_x/V_0 = m_x/m_0 = 0.30\,(x/d) \qquad \text{(1a)}$$

where Vx is the volume flow at a distance x from the jet 'origin', m_x is the corresponding mass flow, and V_0 and m_0 refer to the blastpipe flow.

It is generally accepted that the blastpipe nozzle should be positioned so that the expanding jet impinges on the wall of the chimney in the region of the 'choke', or minimum diameter. Using the angle of divergence found above, this implies that the nozzle should be positioned a distance of about 2.7 *D* below the choke, where *D* is the diameter at the choke. For the purpose of developing a theory we assume that this criterion is met; we shall later speculate on the possible behaviour of the system when the jet is either too big or too small to fit the choke.

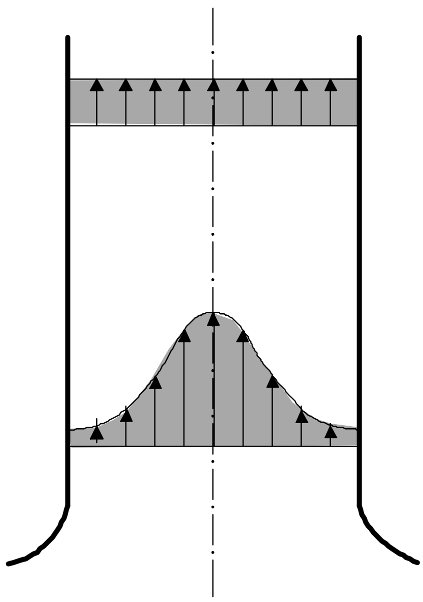


Fig. 2

The next step is to estimate the pressure rise that occurs in moving from the smokebox to the outlet of the chimney (i.e. to atmospheric pressure). For simplicity we shall first consider the case of a parallel chimney, and we shall assume that the frictional forces between the gas and steam mixture and the chimney wall are negligible. As shown in the Appendix, this system can be solved by a straightforward application of the Momentum Theorem, which balances the momentum flows into and out of the chimney against the pressure forces acting on the fluid. The shape of the velocity profile entering the chimney is that shown in Fig.1, and if the chimney is long enough, the profile at outlet will be flat. Fig.2 illustrates the process.

The volume flow of fluid is of course the same at inlet and outlet. If one calculates the momentum at the two sections, however, one finds that it decreases in moving from inlet to outlet. According to the Momentum Theorem, this implies that the pressure at the chimney outlet (i.e. atmospheric) must be greater than that at the inlet. This difference is of course the ‘draught’ produced by the front end. In practice there will be an additional force opposing the flow by virtue of the drag at the chimney wall. This can be ameliorated by making the chimney divergent so that the reduction in momentum, and thus the increase in pressure, in the flow direction are both greater.

The 'Ideal' Front End

As shown in the Appendix, if we take account of the above ratio of *x/D* for the arrangement in which the jet fits the chimney, the magnitude of the pressure difference (or 'smokebox draught') Δp is given by the expression:

$$\frac{\Delta p}{0.5\rho u_j^2} = 0.685\left(\frac{d}{D}\right)^2 \quad .. \qquad . \qquad . \qquad (2)$$

where ρ = density of the fluid (kg / cubic metre) OR (lb / cubic ft)
u_j = flow velocity through blastpipe nozzle (metre / sec) OR (ft / sec)
d = blastpipe nozzle diameter (metre) OR (ft.)
D = chimney choke diameter (metre) OR (ft.)
Δp = pressure difference across chimney (Pascal) OR (poundal / square ft.)

Let us pause here and consider the restrictions that are to be placed on the Equation (2). There is the explicit requirement that the jet from the blastpipe should 'fit' the chimney; this has been taken into account by fixing the ratio $x / D = 2.7$, and this is built into Equation (2). However, in using the turbulent 'free jet' data we have also implicitly assumed that the jet is able to entrain the amount of fluid that it would do if it discharged into a large volume of fluid. Given the vacuum generated by the front end, one may calculate the gas flow through the boiler, and if this is equal to the entrainment, Equation (2) should be valid. If not, it is likely that the entire flow pattern will change, in which case Equation (2) may not be valid. In view of the relationship between steam production m_s and flue gas production m_g (already mentioned above) we can quantify this additional requirement .

Since from Equation (1a), $(m_g + m_s)/ m_s = 0.3\, x/d$, and $x/D = 2.7$, the condition for balance between the flue gas entrained and the actual gas supply is:

$$d / D = 0.81\, m_s / (m_s + m_g) \qquad . \qquad . \qquad . \qquad (3)$$

Equation (2) is then changed to:

$$\frac{\Delta p}{0.5\rho\ u_j^2} = \frac{0.45}{\left(1 + {m_g}/{m_s}\right)^2} \qquad . \qquad . \qquad . \qquad . \qquad (4)$$

For a typical value of $m_g/m_s = 1.8$, the ratio *d/D* is 0.29, and the right hand term in Equation (4) equals 0.057. (Other values of m_g /m_s could be used to represent different boiler operating conditions - particularly in respect of the excess air used). Finally it is worth noting that the expression $0.5\rho\, u_j^2$ very closely equal to the pressure difference across the blastpipe nozzle, so the above ratio is simply the ratio of the smokebox draught to the backpressure in the blastpipe. If the ratio m_g/m_s remains constant then so does the draught/backpressure ratio.

Notice that equations (2) and (4) are in dimensionless form: i.e. the units on each side of the equation cancel out since, for example pressure has the same units as ρu_j^2 and d the same units as D . The equation can be used with any consistent set of units such as those shown above. (It may help to state that a pressure of 1 Pascal is equivalent to 0.102 mm water gauge, and 1 poundal / sq. ft. to 0.006 in. water gauge.) Since Equation (4) is such an important equation an example of its use is given below, using the two sets of units.

Mass flow of steam	=	13.61 kg/hr. (0.00378 kg/sec.)	30 lb/hr (0.00833 lb/sec)
Mass flow of gas	=	24.5 kg/hr.	54 lb/hr.
Nozzle diameter	=	7.37 mm (0.00737 m)	0.290 in. (0.0242 ft)
Chimney diameter	=	25.4 mm (0.0254 m)	1.0 in. (0.0833 ft.)
Steam density	=	0.6 kg/cu. m	0.0374 lb/cu.ft
Nozzle flow area	=	0.0000427 sq.m	0.00046 sq.ft

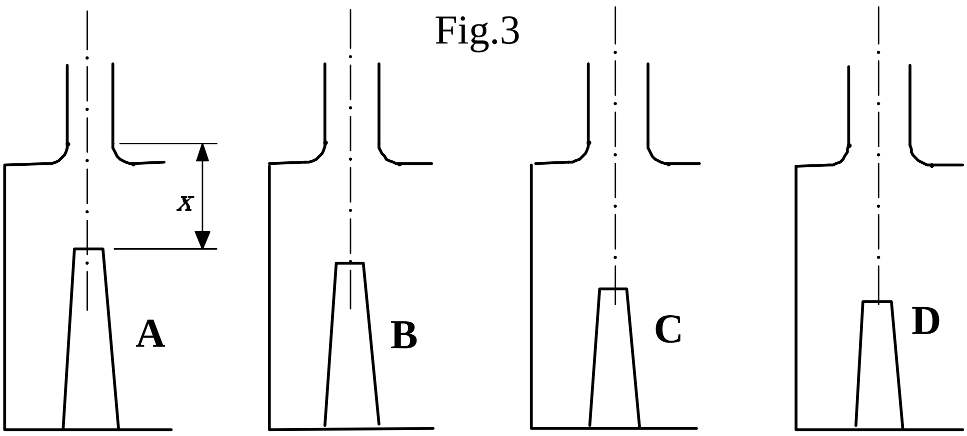
[And since velocity = mass flow ÷ (density × X-sectional area)]

Steam velocity	=	148 m/s	484 ft./s
Thus $0.5\rho u_j^2$	=	6571 Pascal (670 mm water gauge)	4380 poundals/sq.ft (26.38 in. water gauge)
Pressure difference (draught)	=	$0.685\times670\times(7.37/25.4)^2$	$0.685\times 26.38\times(0.29/1.0)^2$
	=	38.6 mm water	1.52 in water

We shall refer to front end designs that conform to Equation (4) - i.e. designs in which the jet fits the chimney and in which the stated value of m_g / m_s is maintained - as 'ideal' designs. They are ideal only in the sense that they conform to the theory in its present form. Experiment may show that the theory needs modification.

Equation (4) is illustrated in Fig.(3) which shows four different blastpipe and chimney designs, all with a steam mass flow of 30 lb/hr. and all maintaining the same ratio of steam to gas mass flow rates. If the details of the boiler were known one could estimate the draught required to produce the specified steaming rate and compare this with the draught that the front end is capable of producing.

Fig.3



Arrangement	A	B	C	D
Nozzle diameter (in)	0.290	0.319	0.348	0.377
Chimney diameter (in)	1.000	1.100	1.200	1.300
Dimension “x” (in)	2.700	2.970	3.240	3.510
Draught (in. water)	1.520	1.040	0.730	0.530
Backpressure (psi)	0.950	0.650	0.460	0.330

ALL CASES: Steam flow 30 lb/hr Flue gas flow 54 lb/hr

The 'Non-Ideal' Front End

It is obvious that even if the 'ideal' arrangement can be met for one operating condition, it is unlikely to hold for different rates of steam usage and different firing conditions. This is of course true for any theory or design rule; at least our theory has the advantage of being of the correct form! For a start let us consider how the 'ideal' front end behaves as the steam flow changes. Other things remaining the same, the draught produced will increase in proportion to the square of the steam mass flow, and the mass flow of entrained gas will increase in proportion to the steam mass flow. If the front end is to properly serve the boiler to which it is attached, the flow resistance of the boiler must also increase in proportion to the square of the gas mass flow. This may well happen in large boilers in which the flow in the boiler tubes is turbulent, but not in small boilers operating at laminar flow conditions where the flow resistance increases in proportion to the flow rate (or, at any rate less rapidly than the flow rate squared). In the latter case, the tendency would be for the draught to be more than adequate as the steaming rate increases, and less than adequate as it decreases. Of course, it may well be the case that in a particular locomotive the front end is designed to produce a more than adequate draught over the greater part of the steaming range, in which case one can always open the firehole door!

If the mismatch between draught and boiler pressure drop is such that less flue gas is available at the smokebox than the jet can entrain there must be a change in the behaviour of the flow. The most likely change is for there to be a reverse flow close to the chimney wall, the air flowing in being subsequently entrained in the jet and discharged through the chimney. It may be possible to model this using one of the 'Computational Fluid Mechanics' programmes that are now available. Further work, both experimental and theoretical is required.

Conclusions

A theory has been advanced for an ideal arrangement of the front end. In this arrangement the blastpipe is so placed that the jet of steam and flue gas impinges on the chimney wall at the 'choke', or narrowest section. It is also assumed that the supply of flue gas matches the entrainment of gas into the jet; this is achieved when the pressure drop through boiler and fire is equal to the draught produced by the front end. In these circumstances the draught produced can be shown to be proportional to the pressure difference across the blastpipe nozzle.

Further work, both experimental and theoretical, is required on the effect of taper chimneys and on the effect of 'non-ideal' arrangements when the jet from the blastpipe does not fit the chimney, or when the flow of gas into the jet is restricted. Even if ideal conditions exist at one rate of steaming it is unlikely that they will prevail over the whole range. It should be stressed that reliable experimentation must always take precedence over theory; however, experimental results can be generalised and better understood if they can be placed in a theoretical framework.

It is not easy to isolate front end performance from the behaviour of the rest of the locomotive - particularly the boiler. A computer programme that links the front end with the fluid flow and heat transfer characteristics of the boiler has been written [7], and can be made available to anyone who is interested. This enables a particular design to be 'operated ' at various firing rates, and the match between boiler pressure drop and front end draught can then be examined.

References

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Appendix

The turbulent free jet can be defined in terms of the following relationships:

1. The distribution of velocity across the jet - i.e. $u / u_{max} = f(r / x)$ (See Fig.1)
2. The centreline velocity u_{max} as a function of u_j , d, and x (See Fig.1)
3. The momentum of the jet as a function of x (This must remain constant because there is no external force acting on the jet.)

Velocity distribution across jet

A simple mixing length turbulence model used to describe the motion of the jet yields a constant turbulent or 'eddy' viscosity throughout. (The scale of the turbulence remains proportional to the width of the jet, and the turbulent velocity fluctuations inversely proportional to the width: thus eddy viscosity, the product of the two, remains constant). This means that the mathematical solutions developed for non-turbulent viscous flow in jets are applicable to turbulent jets. When fitted to experimental data for turbulent jets this yields the distribution:

$$\frac{u}{u_{\max}} = \frac{1}{\left[1 + 62.5\left(r/x\right)^2\right]^2} \qquad . \quad . \quad . \quad . \qquad \text{(A1)}$$

The volume flow is $$V = \int_0^\infty 2\pi ur\ dr = 0.016\pi\, u_{\max} x^2 \qquad . \quad . \quad . \qquad \text{(A2)}$$

Momentum of jet

$$M = \int_0^\infty 2\pi\,\rho\, r\, u^2\ dr = \int_0^\infty \frac{\pi\,\rho\, u_{\max}^2 x^2\ d\left(r/x\right)^2}{\left[1 + 62.5\left(r/x\right)^2\right]^4}$$

$$= \ 0.00533\,\pi\rho u_{\max}^2 x^2 \qquad . \quad . \quad . \quad . \qquad \text{(A3)}$$

Centreline velocity of jet

Assume $u_{max} = k\ _{uj}\ d/x$

(i.e a centreline velocity that varies like 1/*x*, - consistent with experiment)

Momentum $M = 0.00533\,\pi\rho x^2 k^2 u_j^2 d^2 / x^2$

and since there is no change in momentum in the *x*-direction this must be equal to the momentum issueing from the blast nozzle, which is:

$$M = (\pi/4)\, d^2\, \rho\, u_j^2$$

Therefore $k = 1/\sqrt{(0.00533 \times 4)} = 6.86$

Using this to eliminate u_{max} from eqn. (A2), and noting that the volume flow through the nozzle is equal to $Vo = (\pi/4)d^2u_j$, the ratio of the volume flow *V* at *x*, to the volume flow from the nozzle is:

$$\frac{V}{V_0} = 0.44\frac{x}{d} \quad . \quad . \quad . \quad . \quad . \quad . \qquad \text{(A4)}$$

(Note that Eqn.(A4) defines the volume flow in the theoretically infinitely wide jet)

Practical jet width

Clearly the bulk of the jet momentum is contained in the central region of the jet, and we therefore define a width which contains virtually all the momentum. For example, if we integrate out to $u/u_{max} = 0.1$, we find that the included angle of the jet is 21°, and the volume flow is given by:

$$\frac{V}{V_0} = 0.30\,\frac{x}{d} \quad . \quad . \quad . \quad . \quad . \quad . \qquad \text{(A5)}$$

By repeating the momentum integral out to $u/u_{max} = 0.1$ we can then show that the momentum of the practical width of jet is over 99% of the infinitely wide jet. From now on we shall define the jet in terms of Eqn.(A5) and the momentum at the nozzle :

$$M = (\pi/4)\, d^2\, \rho\, u_j^2 \quad . \quad . \quad . \quad . \quad . \qquad \text{(A6)}$$

Fit jet to chimney

Using the above definition of jet boundary we find that if the jet is to impinge on the chimney wall at the choke (diameter *D*) it must be positioned so that:

$$\frac{x}{D} = 2.70 \quad . \quad . \quad . \quad . \quad . \quad . \qquad \text{(A7)}$$

Pressure rise in parallel chimney

The idea here is to calculate the jet momentum at the chimney outlet on the assumption that the velocity is uniform at this point, and to subtract this from the known jet momentum at the choke. If we neglect drag at the chimney wall the Momentum Theorem can be applied to calculate the pressure force on the fluid in the chimney (i.e. the draught " cross-sectional area of chimney)
Outlet velocity from chimney is given by:

$$\bar{u} = \frac{V}{\pi/4\, D^2} \quad = \frac{0.30\, x V_0}{\pi/4\, D^2 d} \quad = \frac{0.30 x\, \pi/4\, d^2 u_j}{\pi/4\, D^2 d}$$

which together with Eqn. (A7) gives:

$$\bar{u} = 0.811 \frac{d}{D} u_j$$

The outlet momentum is therefore:

$$M_0 = \pi/4\, \rho D^2\, \bar{u}^2 \quad = \pi/4\, \rho\; d^2 u_j^2\; (0.811)^2$$

Applying the Momentum Theorem we find that the pressure rise in the chimney (i.e the 'draught') *Δp* is:

$$\Delta\, p = \frac{M - M_0}{\pi/4\, D^2} = \; 0.685\, \frac{\rho u_j^2}{2} \left(\frac{d}{D} \right)^2 \qquad . \qquad . \qquad \text{(A8)}$$

Sometimes it is more convenient to express the draught in terms of the mass flow rate through the blastpipe nozzle:

$$\Delta\, p = 0.55\, \frac{m_j^2}{\rho d^2 D^2} \quad . \qquad . \qquad . \qquad . \qquad \text{(A9)}$$

Note that the above expressions assume implicitly that there is sufficient gas flow through the boiler to match the amount entrained by the jet. (See the section entitled 'The Non-ideal Front End')

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