BOILER DESIGN AND SELECTION IN THE CANE SUGAR INDUSTRY

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1.00 Properties and Uses of Steam

1.1 *Properties*

Steam is a colourless odourless gas produced by adding heat to water. Its physical properties have been determined experimentally and published in the form of steam tables. There are slight differences between the results obtained by various workers, but these do not materially affect the design of steam generating plant.

Callendar's 1939 steam tables (1) shown graphically in Fig. 1.1 illustrate the following imporant characteristics:

(*a*) The saturation temperature of steam rises continuously with pressure.

(*b*) The total heat of dry saturated steam reaches a maximum at about 465 p.s.i.a. The proportion of latent heat to total heat falling constantly until at the critical pressure (3,206.2 p.s.i.a.) no latent heat is added at all.

(*c*) The specific volume of steam decreases with pressure until it equals that of water at the critical pressure.

1.2 *Uses*

In a sugar factory steam is used primarily for:

(*a*) Generating power.

(*b*) Concentrating sugar juices.

Set out in table 1.1 are the pressure and temperature conditions normally encountered in a modern factory. power generating conditions are usually determined by the size and type of machine whilst process conditions are limited by a combination of the carameli-sing temperature and pressure vessel design factors, as well as overall factory thermal balance requirements.

2.00 Fuels

The economic viability of the cane sugar industry largely depends upon the use of bagasse as a fuel to generate power and process steam. In a well balanced raw sugar factory the quantity of bagasse available should be just sufficient to meet the total energy load. Unbalanced factories may experience a short fall or surplus of bagasse in which case either costly auxiliary fuels have to be used or costly bagasse disposal techniques employed. Where off crop outside power and irrigation loads are high, these can be met either by increasing factory thermal efficiency and stock piling baled bagasse for use during the off crop or by burning an auxiliary fuel. This choice is dependent upon local conditions.

Scheduled in table 2.1 are the chemical and physical properties of typical fuels used in the industry. Bagasse and hogged wood have similar characteristics, but differ radically from the other two. On a dry basis their chemical characteristics, as with most fibrous fuels, are almost identical, i.e. they have low ash and high volatile contents. Physically in the "as fired" condition they have high moisture contents, and low calorific values and bulk densities.

The average Natal bituminous coal is free burning, has a high ash fusion temperature (plus 1,400° C) and exhibits reasonable swelling characteristics. Sulphur content is low whilst its calorific value is fairly high (11,500 — 12,500 B.T.U/lb). Its bulk density is about eight times that of bagasse, while from 15–20 times

TABLE 1.1

Steam Conditions in a Cane Sugar Factory

Use	Equipment	Steam Conditions	Remarks
Electrical Power Generation	Turbo alternator of: back pressure, pass out/condensing or condensing design	250–900 p.s.i.g. 600–850° F.	Power is generated by expanding H.P. steam down to process conditions, i.e. 30–40 psia sat. Where condensing facilities are included, these cater for balancing electrical and steam loads and/or meeting off crop power demands.
Mill Drives	a) Electrical	—	Power obtained from main turbo-alternator station.
	b) Steam Turbine	300–450 p.s.i.g. 650–750° F.	Small horse powers preclude higher steam conditions. Due to high efficiencies, pressure reducing and desuperheating plant required to balance factory load.
	c) Reciprocating Steam Engines	100–250 p.s.i.g. sat–550° F.	High capital cost of plant and foundations and oil entrainment in steam have tended to make this type of prime mover obsolete.
Process	Evaporators, Juice heaters, pans, etc.	up to 40 p.s.i.a. sat.	Since the heat transfer co-efficient of saturated steam is about 10 times higher than superheated steam, superheated conditions should be avoided.

TABLE 2.1

Chemical and Physical Properties of Fuels used in Cane Sugar Industry

Property	Bagasse	Hogged Wood	Bituminous coal	Oil	Remarks
Chemical properties					
1) Proximate analysis "as fired"	from mill	as cut pine	Natal peas	medium/heavy	
Carbon %	11.5	16.7	60.0	—	Note the similarity between bagasse and wood, also their high volatile and moisture contents in relation to the solids content
Volatiles %	37.0	43.0	20.0	100.0	
Water %	50.0	40.0	4.0	—	
Ash %	1.5	0.3	16.0	trace	
	100.0	100.0	100.0	100.0	
2) Ultimate analysis "as fired"					
Carbon %	22.5	30.2	67.0	83.5	
Hydrogen %	3.0	3.7	4.2	11.7	Fibrous fuels while differing radically in physical respects have very similar chemical characteristics on dry basis
Sulphur %	trace	trace	1.3	3.3	
Nitrogen %	—	—	1.6	—	
Oxygen %	23.0	25.8	5.8	1.4	
Phosphorus %	—	—	0.1	—	
Moisture %	50.0	40.0	4.0	—	
Ash %	1.5	0.3	16.0	—	
	100.0	100.0	100.0	100.0	
3) Gross Calorific Value G.C.V. (BTU/lb.)	4,108	5,150	11,800	18,500	G.C.V. and N.C.V. for Bagasse calculated at 1.8% Sucrose from S.M.R.I. formulae: G.C.V.=8,190—18S—81W BTU/lb. N.C.V.=7,650—185—86.4W BTU/lb. where S = % Sucrose W = % Moisture N.C.V. of other fuels calculated by deducting [(weight total moisture + weight moisture due to oxidation of Hydrogen) × 1,055] BTU from G.C.V.
4) Net Calorific Value N.C.V. (BTU/lb.)	3,298	4,380	11,360	17,270	
5) Theoretical weight of air required for complete combustion per 10,000 BTU on G.C.V. (lbs)	6.37	7.08	7.60	7.38	
6) Max. theoretical CO_2 in flue gases measured by Orsat %	20.70	20.10	18.45	15.60	
7) Practical excess air requirements for complete combustion %	48	43	37	15	
8) Corresponding % CO_2 in flue gas as measured by Orsat %	14.0	14.0	13.5	13.6	
9) % moisture by weight in flue gases at above CO_2 %	15.95	12.20	3.18	6.32	
10) Practical boiler efficiencies (100,000 pph units) with heat recovery equipment designed to bring final gas temperatures down to 450° F at the above excess air figures at 80° F ambient air temperatures					Calculate losses and deduct from 100.00
Losses					
a) Unburnt carbon in ashes, grits and stack discharge %	3.00	2.00	3.16	—	It is extremely difficult to assess the unburnt losses when burning bagasse. The figure quoted however is considered to be reasonable.
b) Dry gas loss %	8.71	9.03	9.52	7.44	This is calculated as follows: (dry gas wt.) × (average S.H.) × (450–80)/G.C.V.
c) Wet gas loss %	22.80	17.30	4.31	6.84	This is calculated as follows: (wt. of water in flue gases) × (1,265–48)/G.C.V.
d) Moisture in air loss %	0.15	0.15	0.15	0.15	Varies from 0.1 to 0.2 depending upon humidity.

Property	Bagasse	Hogged Wood	Bituminous Coal	Oil	Remarks
Losses—continued.					
e) Radiation loss	0.73	0.73	0.73	0.73	Varies from 0.6 to 1.8 depending upon size of boiler. The figure quoted is for a 100,000 pph boiler.
f) Unmeasured losses . . %	0.50	0.50	0.50	0.50	Sensible heat in ashes, etc.
	35.89	29.71	18.37	15.66	
Efficiency on G.C.V. . . %	64.11	70.29	81.63	84.34	
Efficiency on N.C.V. . . %	80.00	82.55	84.80	90.40	$\text{Efficiency on N.C.V.} = \frac{\text{G.C.V.}}{\text{N.C.V.}} \times \text{Efficiency on G.C.V.}$
Physical Properties					
1) Appearance	Straw coloured fibrous material 50% minus ¼″	Chipped wood about 1″ square	Black gravel like substance 100% minus 1″ 30% minus ⅛″	Black/brown viscous liquid	
2) Bulk density lb/ft³					
Range	5–9	7–20	50–55	60–65	Figures can vary with degree of compaction Bulk density of bagasse at 10 lb/ft² is 7.0 lb/ft³
Average	7	12	54	62	
3) Heating value BTU/ft³ . . .	28,800	61,800	637,000	1,148,000	on G.C.V.
4) Ratio of volume of bagasse burnt to volume of fuel burnt to produce equivalent steam output	1	1.96	17.4	30.3	

the volume of bagasse must be burnt for an equivalent heat output.

Both light and heavy oils are used in the industry. Light oils, because they require no pre-treatment, are instantly available and are therefore preferred as an emergency standby fuel. Cheaper heavy oils are used where an auxiliary fuel must be burnt constantly, i.e. in a mal-balanced factory or where a high off crop load exists.

3.00 Combustion

Combustion is an exothermic oxidising reaction. Its mechanism is not fully understood, but for practical purposes the products of combustion of a fuel such as bagasse can be determined from its ultimate analysis as shown in table 3.1. In a furnace the oxidising agent is air and the main products of combustion are nitrogen, carbon dioxide, water vapour and oxygen. Secondary products are the oxides of trace elements such as sulphur, phosphorous and vanadium.

To ensure complete and efficient combustion in a furnace an adequate supply of oxygen must be brought into contact with the fuel.

The simplest way of measuring combustion efficiency is to determine the percentage by volume of carbon dioxide, oxygen and carbon monoxide in the flue gases. A trace of carbon monoxide indicates incomplete combustion whilst the carbon dioxide or oxygen figures indicate the quantity of excess air present.

The heat released when oxidising carbon to carbon dioxide is 14,590 B.T.U/lb while only 10,210 B.T.U/lb are available if carbon is oxidised to only carbon monoxide.

In practice, unfortunately, to ensure complete combustion a small amount of excess air is needed. Depending on the type of fuel and furnace design, this varies from 10% to 50%, over and above that required for theoretical combustion. The products of combustion are thus diluted and the efficiency of heat recovery is reduced. The reduction in efficiency, however, due to incomplete combustion far outweighs the loss due to the small amount of excess air required to complete combustion. Fig. 3.1 shows the relationship between "excess air" and percentage carbon dioxide for a number of different fuels.

4.00 Combustion Equipment

Combustion equipment must be robust, easy to maintain, consume a minimum of power, and enable the fuel to be burnt as completely as possible in the furnace without using too much excess air. Different fuels require different types of combustion equipment to meet these conditions.

4.1 *Bagasse and Hogged Wood*

These two fuels, because of their high moisture and volatile contents are essentially gaseous in character. The most efficient way of firing them therefore is to introduce them into the furnace in a similar manner to a gas, i.e. with a proportion of the combustion air. Figs. 9.2A and 9.2C illustrate typical furnaces.

Due to their relative bulkiness and the fact that neither bagasse nor hogged timber can be stored successfully in a diverging bunker, a furnace having

TABLE 3.1

Combustion Reactions of Bagasse

Constituent	Parts by wt.	Weight of Oxygen required	Product	Weight of product	Orsat vol. of product in ft.^{3} at 32° F.
C	0.225	0.600	CO_2	0.825	6.74
H_2	0.030	0.240	H_2O	0.270	—
S	—	—	SO_2	—	—
N_2	—	—	N_2	2.010	25.80
O_2	0.230	—	—	—	—
H_2O	0.500	—	H_2O	0.500	—
Ash	0.015	—	Ash	0.015	32.54
	1.000	0.840			
Oxygen in fuel		0.230			
Oxygen required . . .		0.610			

$$\text{Weight of air required} = \frac{0.610}{0.232} = 2.62 \text{ lb.}$$

$$\text{Weight of Nitrogen} = 2.01 \text{ lb.}$$

$$\text{Theoretical dry gas } \%CO_2 = \frac{6.74}{32.54} \times 100 = 20.7\%$$

Notes on Table 3.1:

1) Weight of oxygen required for complete combustion is determined from the molecular weights of the substances involved

$$C \rightarrow CO_2 : C \times \frac{32}{12}$$

$$H_2 \rightarrow H_2O : H_2 \times \frac{8}{1}$$

$$S \rightarrow SO_2 : S \times \frac{1}{1}$$

2) From Avagadro's Hypotheses at a given pressure and temperature all gases have the same number of molecules per ft^3

i.e. $PV = KT$ where in the f.p.s. system $K = 10.7/\text{mol.wt.}$ At 32° F, 30" Mercury the volume of 1 lb. of:

CO_2 is 8.157 ft^3

SO_2 is 5.61 ft^3

N_2 is 12.81 ft^3

Air is 12.385 ft^3

large storage and thermal inertia characteristics can be installed. Fig. 9.2B illustrates a typical example of this type of unit. Should the bagasse supply fail, continuous steaming can be maintained for a period of some 10 to 20 minutes thus providing a reasonable time for bagasse to be reclaimed from store to maintain load, or in the event of a bagasse carrier failure to enable auxiliary power equipment to be brought on line.

Whilst most of the ash produced in this type of unit is disposed of while the boiler is on range through collectors in the boiler itself, the furnace must be shut down at weekly intervals to be manually cleaned. The shutdown can be timed to coincide with the normal weekly factory shutdown. The furnace is extremely simple, has no moving parts and possesses self feeding characteristics which simplifies auto-control.

The state of the fuel bed is quiescent in relation to suspension firing which reduces grit carryover and smut emission considerably. Grit collectors can be dispensed with whereas they are considered essential with suspension firing.

4.2 *Coal Firing*

There are a number of different ways of firing coal. The choice of equipment being dependent upon the fuel characteristics and the size of the unit.

Where unit capacities do not exceed 250,000 p.p.h. South African bituminous coals are best burnt on carrier bar stokers, of the type shown in Fig. 9.2D, having ratings of from 33–38 lb. of coal burnt per hour per ft.^{2} of active grate area. Above 250,000 p.p.h. an economic case can be made out for pulverising the fuel down to the consistency of a face powder and firing it in suspension using a portion of the combustion air as the conveying medium.

If coal and bagasse are to be fired simultaneously, they must be intimately mixed and the fuel bed must not be allowed to build up to more than a thickness of 6". This can best be achieved with the equipment shown in Fig. 9.2C.

Grit carry-over with spreader firing is high and reasonably high efficiency collectors should be installed to avoid a grit carry-over nuisance.

4.3 *Oil Firing*

Of the fuels used fairly extensively in the sugar industry, oil is probably the simplest to burn efficiently. The type of combustion equipment required depends upon its grading, i.e. whether it is a light or a heavy oil and the size of the combustion chamber, but in general consists of a nozzle, through which the oil can be introduced and atomised, surrounded by a chamber through which the combustion air is introduced. Atomising can be achieved by introducing the oil through tiny orifices at high pressure or by means of a mechanical rotor or by steam injection. Fig. 9.2E illustrates a typical oil fired boiler.

5.00 **Fundamental Heat Transfer Relationships**

Heat is added to water to produce steam in three stages, namely:

(*a*) Sensible heat.
(*b*) Latent heat.
(*c*) Superheat (if required).

Heat can be transferred from one medium to another by:

(*a*) Conduction.
(*b*) Convection.
(*c*) Radiation.

All three forms of heat transfer are encountered in a boiler. The fundamental mechanism of each being as ollows:

5.1 *Conduction*

Heat is transferred from one point in a substance to another at a lower temperature by conduction. Fourier's Law for conduction expressed mathematically is:

$$\frac{dQ}{d\theta} = -kA\frac{dt}{dx} \quad . \; . \; . \; 5.1$$

Where the proportionality factor k is known as the thermal conductivity. For steady state heat transfer through a solid of constant cross section the basic equation becomes:

$$q = h\triangle t \quad . \; . \; . \; 5.2$$

Where h is known as the heat transfer co-efficient. From this equation it is evident that the heat transferred is directly proportional to the difference in temperature between the hot and cold faces of the body.

If heat flows through two solids of constant cross section in series then the overall heat transfer co-efficient for the combination of materials acting together is determined by the equation:

$$\frac{1}{h} = \frac{1}{h_1} + \frac{1}{h_2} \quad . \; . \; . \; 5.3$$

If in this equation one heat transfer co-efficient is much larger than the other, the smaller is the predominating one and the larger co-efficient can be ignored. This point is very important in practical boiler design.

5.2 *Convection*

Heat transfer by convection is due to fluid motion. Cold fluid receives heat from a fluid at a higher temperature by mixing with it. Free or natural convection occurs when the fluid motion is not implemented by mechanical agitation, but merely by differences in fluid densities. When, however, a fluid is mechanically agitated, heat is transferred by a process known as forced convection.

Convection heat transfer in boilers is usually forced, natural convection playing only a very small part which is usually neglected. The rate of heat transfer by forced convection from a hot gas flowing at a constant mass rate to a cold tube of uniform diameter has been found to be influenced by the velocity v, density ρ, specific heat Cp, thermal conductivity k and viscosity μ, of the gas as well as the outside diameter D of the tube. The velocity, viscosity, density and tube diameter affect the thickness of the fluid film at the tube wall through which heat must first be conducted, as well as the extent of fluid mixing. The average temperatuie of the fluid is a function of its thermal conductivity and specific heat. By means of dimensional analysis the relationship given in the following equation

$$\frac{hD}{k} = A\left(\frac{DG}{\mu}\right)^a \left(\frac{Cp\mu}{h}\right)^b$$

can be obtained between the above variables.

The dimensionless groups $\frac{hD}{k}$, $\frac{DG}{\mu}$, $\frac{Cp\mu}{h}$ are known as the Nusselt, Reynolds and Prandtl numbers respectively. The Prandtl number for most gases is very nearly constant and equal to 0.78.

The constants A, a and b are determined experimentally and for gases flowing normal to banks of tubes in line they are:

$A = 0.26$ $\quad a = 0.6$ $\quad b = 0.33$

Provided that $\frac{DG}{\mu}$ exceeds 2,100, equation 5.4 becomes:

$$Nu = 0.24\, Re^{0.6} \quad . \; . \; . \; 5.4a$$

Fishenden and Saunders (2) have plotted a number of curves correlating heat transfer co-efficient with tube diameter, mass gas flow and tube arrangement and these are quite often used in boiler design where more specific relationships do not exist. Where tubes are arranged in staggered formation or where gases flow olbiquely across or parallel to tube banks equation 5.4 holds with modified constants.

Fig. 5.2 shows the relationship between mass gas flow and heat transfer co-efficient for flue gases flowing transversely over a bank of tubes square pitched at two diameters. From this curve it is evident that heat transfer is greater for smaller diameter rubes, and higher mean film temperatures and mass gas gas velocities. Average gas side convection heat transfer co-efficients normally encountered in boiler design vary from 5 BTU/ft.^{2} hr. °F in the low temperature passes to 14 BTU/ft.^{2} hr. °F in the high temperature pases. The average heat transfer co-efficient from a hot tube to a steam/water mixture flowing through it is about 1,000 BTU/ft.^{2} hr. °F whilst the average heat transfer co-efficient from a hot tube to superheated steam flowing through the tube is about 120 BTU/ft.^{2} br. °F. From these figures it will be appreciated that the overall heat transfer co-efficient from gas to a steam/water mixture is virtually determined by the gas side co-efficient whilst the overall heat transfer co-efficient from gas to steam is determined by both gas side and steam side co-efficients. This is illustrated in Fig. 5.1. Note also the rise in Metal temperature in the second case.

5.3. *Radiation*

The relative importance of the three modes of heat transfer differs with the temperature level of the system. In conduction through solids the mechanism consists of an energy transfer through a body whose molecules except for small oscillations around a space within themselves remain continuously in a fixed position. In convection, heat is first absorbed by particles of fluid immediately adjacent to the source and then transferred to the interior of the fluid by mixing with it. Both mechanisms require the presence of a medium to convey heat from the source to the receiver. Radiant heat transfer does not require an intervening medium.

If the phenomena of conduction and convection on the one hand are contrasted with thermal radiation on the other, it is found that the former are affected by temperature differences and very little by temperature level whereas the latter increases rapidly with increase in temperature level. The rate of heat transfer by radiation varies in fact as the difference between the fourth powers of the absolute temperatures of source and sink. The relationship

$$q = \sigma A (T_1^4 - T_2^4) \quad . . . \quad 5.5$$

expresses mathematically this concept and is known as the Stefan-Boltzmann law. The constant of proportionality in this equation is known as the Stefan-Boltzmann constant. The relationship assumes that the emissivity of the source and sink are both unity, i.e. they are both referred to as black bodies. A black body is a body which will not reflect any radiant energy. In the combustion chamber of a boiler the emissivity of the flame and heating surfaces are not unity, nor do the heating surfaces present the equivalent of an infinite parallel plane to the radiating gases. The Stefan-Boltzmann relationship must therefore be corrected accordingly. The corrected relationship can be written as

$$q = \sigma A F_a \cdot F_{e1} (T_1^4 - T_2^4) \quad . . . \quad 5.6$$

where F_a is a dimensionless geometric factor and F_{e1} is a dimensionless emissivity factor.

The luminous products of combustion radiate heat to the furnace walls which can be made of refractory material, water cooled steel surfaces or a combination of these. The actual amount of heat transferred can be calculated using Mullikens (3) theory, the combustion chamber heat balance equation being

net heat input

= heat transferred by radiation

+ heat carried over into convection passes

i.e. N.C.V.

+ heat in air/lb. of fuel above ambient temperature

$$= 0.1723 \, A \, F_a \, F_{e1} \left[\left(\frac{T_{fg}}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] + C_p W_g (T_{fe} - T_a) \quad . . . \quad 5.7$$

T_{fg} from the Stefan-Boltzmann relationship is the temperature of the source in deg. F. absolute.

A furnace approximates to these conditions only if the mass of hot gas acts like a black body. To what extent this is true is difficult to determine but small or microscopic black particles in the flame and the effect of reradiating refractory walls will cause the effective emissivity to approach unity.

To simplify the calculations T_{fg} is taken equal to the furnace exit gas temperature rather than a possible higher mean radiating temperature. The effective emissivity based on the furnace exit gas temperature will be higher than the effective emissivity based on the higher mean radiating furnace temperature. Mathematically speaking, this is simply stating that the radiation from a body at a given temperature and at a given emissivity may be equal to that of the same body at a lower temperature and higher emissivity. It is believed that the emissivity based on the furnace exit gas temperature may not be far from unity. Experimental data are not conclusive in this respect, but calculations using this assumption should yield an answer within $3\frac{1}{2}\%$ F of the true exit gas temperature.

The combustion chamber heat balance equation involving a term raised to the power four can be solved by trial and error if it is written in the form

Net heat input =

$$\frac{0.1723 \, A \, F_a \, F_e}{W_f} \left[\left(\frac{T_{fe}}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] + C p W_g (T_{fe} - T_a) \quad . . . \quad 5.8$$

where T_{fe} is the furnace gas exit temperature and F_e the corrected emissivity factor. Typical values of F_a and F_e are given in Fig. 5.3 and table 5.1 respectively.

TABLE 5.1

Emissivity Factors for Various Fuels

Fuel	F_e = Col. 1 × Col. 2	Flame emissivity Col. 1	Chamber factor Col. 2
Bagasse	0.72	0.85	0.85
Wood	0.72	0.85	0.85
Bituminous Coal . .	0.81	0.90	0.90
Oil	0.85	0.90	0.95

On leaving the furnace the hot gases are non-luminous and at a temperature of 1,700–2,200 deg. F. Heteropolar gases however such as water vapour and carbon dioxide possess radiant emission bands in the infra-red range of sufficient magnitude to merit consideration in the convection passes of the boiler. Heat transfer by non-luminous gas radiation varies with the partial pressure of the gas and also with a mean beam length which is defined as the radius of an equivalent hemispherical gas mass. For gas shapes of industrial importance it is found that any shape is approximately representable by an "equivalent" hemisphere of proper radius. The evaluation of the equivalent hemisphere involves tedious graphical or analytical methods and a number of text books deal with this aspect of the problem. Fig. 5.4 illustrates typical curves for 2″ o.d. tubes pitched at two diameters.

While it is not strictly correct to talk in terms of a radiant heat transfer co-efficient in the same sense as conductive or convective heat transfer co-efficients, it is nevertheless useful to use such a concept in order to obtain a quantitive comparison of the three types of heat transfer. Typical figures are therefore given in table 5.2. From this table it will be noted that the luminous radiant heat transfer co-efficient is much higher than the convective co-efficient, and therefore it follows that it is more economical to transfer heat radiantly than convectively. Other considerations, however, which are discussed later limit the amount of heat which can be transferred in this manner.

TABLE 5.2

Approximate Overall Heat Transfer Co-efficients encountered in Boiler Design

Location	Type of heat transfer	overall heat transfer coeff (BTU/ft²hr°F)
Combustion Chamber	Luminous radiation	20—30
Superheater	Convection	10—14
	Non-luminous radiation	2—3
Convective boiler passes	Convection	4—14
	Non-luminous radiation	0.5—2
Economiser—Bare tube	Convection	6—10
	Non-luminous radiation	0.4—0.5
Cast Iron	Convection	3.5—4.5
Air Heater—Tubular	Convection	2—6

6.00 Fluid Friction

As early as 1874 Osborne Reynolds pointed out that there was a relationship between convective heat transfer and fluid friction. The analogy between the two arises from the fact that the transfer of heat and the transfer of fluid momentum can be related. In simple terms the relationship is:

$$\frac{\text{Heat actually given up}}{\text{Total heat available to be given up}} = \frac{\text{Momentum lost by friction}}{\text{Total momentum available}}$$

The Reynolds analogy was extended by Prantdl to include the laminar layer which exists near a pipe wall as well as the turbulent layer. The Prandtl modification is sometimes called the Prantdl analogy. Modern theory now presumes that the distribution of velocities no longer ends abruptly at the laminar layer but that there is instead a buffer layer within the laminar layer in which transition occurs. Other extensions, therefore, of the analogies have been derived.

In practical boiler terms the Reynolds analogy implies that in order to obtain a higher convective heat transfer co-efficient, more work must be done in overcoming the fluid friction between hot gases and the heating surfaces. Since this involves using more fan power the economic advantages of higher heat transfer rates and hence smaller boilers must be balanced against the cost of additional fan power consumption.

7.00 Circulation

The water side heat transfer co-efficient as in the case of the convective gas side heat transfer co-efficient is a function of the velocity of the fluid flowing. If the water side co-efficient becomes small in relation to the gas side co-efficient, the tube metal temperature will approach the gas temperature, thus overheating the metal surfaces and causing blistering and possibly even rupture. In boilers with good circulation metal temperatures are only about 10°–20° F higher than the saturated water temperatures.

Natural circulation in a boiler is caused by the difference in density between water in the feeding leg of a circuit and the water/steam mixture in the riser leg of the circuit. In Fig. 7.1 a simple circuit is shown where two pressure vessels are connected by a bank of heated downcomers and a bank of heated risers. The number and diameter of the tubes in the riser bank is fixed by the heating surface necessary to transfer the required amount of heat to the water, while the number of tubes in the downcomer circuit is fixed by the cross sectional area of tubing required to maintain an adequate water supply to the risers. The term "adequate water supply" is defined empirically by a circulating factor which is the ratio of the number of pounds of water/steam mixture at any point in a circuit to the number of pounds of steam at that point. The circulating factor is introduced to fix the quantity of water to be supplied to a circuit and is a function of the heating intensity and an empirical overheating parameter. The ratio of steam to water obviously affects the density of the mixture and hence the circulating factor must affect the static head which produces circulation, the static head being the difference in weight of fluid per unit area in the downcomer and riser tubes. It should be remembered that density is affected by the difference in velocity between steam and water flowing in a tube. This difference has the apparent effect of increasing the circulation factor and hence decreasing the available static head.

The static head in the circuit is used to overcome the head drop across the riser tubes and downcomer tubes due to friction, entrance and exit losses, acceleration losses and bend losses all of which are functions of the quantity and density of the water/steam mixtures flowing.

Since the static head is a function of the difference in density between the fluid flowing in the riser and downcomer tubes and the vertical height of these tubes it follows that tubes where possible should always be placed vertically in order to obtain maximum circulation. Fig. 7.2 illustrates a typical boiler circuit.

As pressure conditions rise the difference in density between water and steam approaches zero (see Fig. 1.1) and the available static head decreases. High pressure steam generators therefore have to be built as high head units or alternatively some means of forced or assisted circulation must be employed. A typical example of a forced circulation steam generator is the Lamont unit. Steam generators operating at super critical pressures are generally of the "once through" type.

8.00 Boiler Design

8.1 *Boiler Heat Balance*

The previous sections have dealt with the properties and uses of steam, types of fuel available, their methods of combustion and the relationships between the rate of heat transfer, friction loss and circulation.

The performance of a boiler can be measured without any reference to its physical characteristics. Only the initial conditions of the feedwater and fuels and the final conditions of the steam, exhaust gases, ash and grits and casing temperatures need be considered. Fig. 8.1 shows diagrammatically the factors which go to make up a boiler heat balance; i.e.

Heat in = Heat transferred to steam
+ Heat lost in dry exit gases
+ Heat lost in evaporating moisture produced when hydrogen in fuel is oxidised to water.
+ Heat lost in evaporating free and inherent moisture in fuel
+ Heat lost in evaporating moisture in combustion air.
+ Heat lost due to unburnt carbon in ashes.
+ Heat lost due to unburnt carbon being carried over with grits to grit hoppers and stack.
+ Heat lost by radiation and convection from the boiler casing.
+ unaccounted losses.

Heat-In

The "Heat-In" is the gross calorific value of the fuel determined by the bomb calorimeter, all products of combustion having been cooled down to 60°F and the moisture condensed.

Heat-In Steam

The "Heat-In Steam" is the amount of heat which is transferred to the water to produce steam.

Dry Gas Loss

The "Dry Gas Loss" is the heat lost in the exhaust gases by virtue of their temperature above ambient.

Wet Gas Loss

The "Wet Gas Loss" is the heat lost in the exhaust gases in evaporating and superheating the moisture contained in the fuel and that moisture which is formed by the oxidation of hydrogen during combustion.

Moisture in Air Loss

Air used for combustion contains a small amount of moisture which must be evaporated and superheated only to be rejected in the exhaust gases.

Ash Loss

A small percentage of carbon is physically entrained in the ash and its heating value is therefore lost.

Grit Carry-Over Loss

Lighter particles of fuel are sometimes carried over in suspension with the gases and their heating value is lost. If grit collectors are installed, a portion of the grits can be collected and refired.

Radiation Loss

A boiler is generally insulated to bring the casing temperature down to between 120° and 170° F. Insulating to this temperature introduces a small radiation loss from the surface of the boiler which amounts to about 0.5%–1.5% of the total heat input.

Unaccounted Loss

To make up the final boiler heat balance, a figure of between 0.5% to 1.0% is usually included to cover errors in measurement and small losses such as the sensible heat in ash, etc. which are not measured.

Laid out in Table 2.1 are the approximate heat balances of four 100,000 p.p.h. boilers. Each boiler is designed to burn a specific fuel, i.e. bagasse, wood, coal and oil and has sufficient heat recovery equipment fitted to bring the final gas temperature at the designed CO_2 down to 450° F.

As a rought guide a reduction of approximately 30° F in final gas temperature will be proportional to an increase of 1% in boiler efficiency. Further a reduction of approximately 1% in CO_2 will be proportional to a reduction of 1% in boiler efficiency.

Boiler Blowdown

Modern boilerplant requires careful water treatment control. In addition impurities which collect in a boiler must be blown down. From Fig. 8.2 the percentage blowdown can be calculated for specific conditions. In general blowdown should not exceed 5% of the steam generated. This represents a heat loss which is not normally included in the boiler heat balance. However, since water and not steam is blown down only sensible heat is lost and in most installations in the sugar industry a 5% blowdown rate is proportional to a drop in efficiency of only approximately 0.5%.

TABLE 9.1

BOILER CHARACTERISTICS COMPARISON SCHEDULE

Characteristic	Dump Grate	Bagasse Self Feeding	Moving Grate	Coal	Oil	Remarks
a) Relative cost for 100,000 pph unit based on dump grate furnace .	1	0.91	1.14	1.05	0.73	
b) Floor area occupied by boiler (assumes ID fan and grit collector outside building). . . .	2,400 sq. ft.	2,500 sq. ft.	2,420 sq. ft.	2,600 sq. ft.	1,680 sq. ft.	Allows 10 ft. and 5 ft. clearance in front and at rear of boiler respectively.
c) Height of building required to eaves level	48 ft.	40 ft.	60 ft.	54 ft.	40 ft.	
d) Relative building cost (assuming proportional to building volume)	1	0.87	1.27	1.23	0.58	
e) Approx. installed HP . .	480	390	495	300	95	
f) Approx. power consumption at M.C.R. on major fuel . . .	270 KW	220 KW	280 KW	170 KW	53 KW	
g) Auxillary fuel.	Oil	Oil	Coal	—	—	Gas can be burnt on unit fitted with oil burners.
h) Furnace (i)	No thermal storage	Large thermalstorage	No thermal storage on bagasse reasonable on coal	Reasonable thermal storage	No thermal storage	
(ii)	Refractory sidewalls require annual maintenance	Large refractory areas require annual maintenance	Limited refractory maintenance	Limited refractory maintenance	Limited refractory maintenance	
(iii)	Turbulent thermal conditions. Heavy carry over	Quiescent furnace conditions limited carry over	Turbulent furnace conditions on coal and bagasse. Heavy carry over	Quiescent fnurnace conditions. Limited carry over	No carry over	
(iv)	H.P. secondary air required	Medium pressure secondary air required	H.P. secondary air required	H.P. secondary air required	Medium pressure secondary air required	
i) Superheater	Pendant. Not drainable	Cross flow. Drainable	Pendant. Not drainable	Cross flow. Drainable	Pendant or Cross flow. Drainable	
j) Convection passes	Vertical tubes cross baffled	Vertical tubes cross baffled	Vertical tubes cross baffled	Vertical tubes cross baffled	Vertical tubes cross baffled	
k) Heat recovery equipment required for normal industrial application	Airheater	Airheater	Airheater	Economiser	Economiser	
l) Optimum final gas temperature	420/480° F.	420/480° F.	420/480° F.	320/380° F.	350/400° F.	
m) Other factors limiting lower final gas temperatures	Expensive economiser required for small gain in efficiency	Expensive economiser required for small gain in efficiency	Expensive economiser required for small gain in efficiency	Low temperature deposits. Cleaning severe problem	Low temperature deposits. Cleaning severe problem	
n) Grit Collector	Required	Not required	Required	Not required	Not required	
o) Ash Handling	Intermittant mechanical ashing	Furnace must be cleaned manually once a week	Continuous mechanical ashing	Continuous mechanical ashing	—	
p) Automatic Controls	Fuel feed, air flow and furnace pressure regulation required	Due to self feeding properties of furnace auto controls simplified only air flow and furnace pressure regulation required	Fuel feed, air flow and furnace pressure regulation required	Fuel feed, air flow and furnace pressure regulation required	Fuel feed, air flow and furnace pressure regulation required	
q) Ratio weight of water to M.C.R. evaporation	0.70	0.78	0.65	0.58	0.63	

9.00 Boiler Layout

Boiler design is largely dictated by:

1) Fuel:
   Physical and chemical properties;
   Cost.
2) Final Steam Conditions:
   Pressure;
   Temperature.
3) Feedwater Conditions:
   Temperature;
   Impurities.
4) Application:
   Industrial;
   Power Generation.

Fig. 9.2 illustrates a few of the basic boiler designs used in the Cane Sugar Industry. Their characteristics are compared in Table 9.1 while an indication of the effectiveness of the different types of heating surface used in the design illustrated in Fig. 9.2B is shown in Fig. 9.1A. Fig. 9.1B shows the energy flow diagram for this unit.

Industrial design considerations are as follows:

9.1 *Combustion Chamber*

Furnace geometry is governed by the fuel and type of combustion equipment used. High moisture content fibrous fuels require fairly large furnaces with a refractory belt in the combustion zone to ensure stable conditions. With fuels having higher combustion temperatures, however, such as coal and oil, furnace volumes can be reduced and fully water cooled walls included.

To minimise screen and superheater tube fouling, sufficient heating surface should always be incorporated to ensure that the gas leaving temperature is at least 150° to 200° F below the ash deformation temperature. Reducing the gas leaving temperature much below this point reduces the temperature potential in the superheater zone thus making the superheater itself much larger.

9.2 *Superheater*

Superheaters can be broadly classified into two groups:

(*a*) Drainable
(*b*) Non-Drainable.

Drainable superheaters are less prone to overheating on start-up. The pendant non-drainable type however are readily accommodated in the tall combustion chambers essential with suspension firing.

Tube configuration can be critical. A minimum pressure drop varying from 5 to 25 p.s.i. at M.C.R., is usually necessary to ensure an even steam flow distribution through each tube. Further, by varying tube pitching superheater characteristics in relation to load can be appreciably altered. Increasing the tube pitching increases radiation heat transfer at the expense of convection heat transfer. Conversely, decreasing tube pitching increases convection heat transfer at the expense of radiation heat transfer. By carefully adjusting tube geometry therefore an almost flat characteristic can be obtained over a wide range of boiler loading. Fig. 9.3 illustrates the convection and radiation transfer characteristics in relation to load.

9.3 *Convection Surfaces*

The boiler exit gas temperature can be economically reduced to within 250° and 300° F of the saturation temperature by means of convection heating surface. As boiler pressures rise this surface is less effective and heat recovery surface, in turn becomes more important. Up to about 650 p.s.i.g. convection heating surface plays an important role in the design while at pressures exceeding 1,000 p.s.i.g. it is virtually redundant.

9.4 *Recovery Equipment*

Heat recovery equipment is designed to increase boiler efficiency by still further reducing the boiler exit gas temperature. This is accomplished by either an economiser which transfers heat in the flue gases to the feed water, or an air heater which transfers heat in the flue gases to the combustion air.

In industrial plant the quantity of heat which can be transferred in each case is limited by:

(*a*) The gas temperature can only be reduced economically to within 150° to 200° F of the feedwater or ambient air temperatures.
(*b*) The quantity of heat transferred to an economiser operating under industrial water treatment conditions should be limited to avoid depositing feedwater solids in the tubes which occurs if boiling takes place.
(*c*) The combustion air temperature should not exceed a figure which would induce high stoker and furnace maintenance costs due to excessive combustion chamber temperatures. This figure is usually limited to between 350° and 400° F.

Economisers and air heaters are made of either cast iron or mild steel, the choice being dependent upon factors such as the sulphur, phosphorous and moisture content of the fuel and operating metal temperatures. Wherever possible designs should ensure that operating metal temperatures are higher than the dew point of the gases so as to minimise corrosion. There are a number of techniques used to accomplish this, such as:

(*a*) Recirculating hot air through the cold passes of an air heater.
(*b*) Recirculating hot water from the boiler drum through the economiser to elevate the feed water temperature.
(*c*) Arranging heating surfaces as parallel rather than counter flow exchangers.
(*d*) Adjusting the heat transfer co-efficient by varying mass flow rates so that metal temperatures are closer to the gas temperature rather than the air temperature (see Fig. 5.1).

All these techniques unfortunately reduce the effective temperature difference between the hot gases and the water or air. Larger heating surfaces are therefore required but maintenance and replacements problems are minimised. Where metal temperatures cannot be effectively controlled, e.g. where feed water temperatures to economisers are below 180° F or where high efficiencies are required heat recovery equipment should be made of either cast iron which enables a heavier metal section to be used economically or a specially treated steel.

The choice and extent of heat recovery plant required largely depends on fuel costs and characteristics. As final gas temperatures are reduced to increase efficiency, more and more heat recovery equipment is required until the increase in cost of plant and additional fan power outweighs the saving in fuel. Further, lower final gas temperatures introduce corrosion and fouling problems. The economic limit has therefore to be carefully assessed.

The combustion stability of high moisture content fuels is improved if the combustion reaction temperature is increased. Air heaters are therefore far more advantageous than economisers when burning these fuels.

9.5 *Draught Plant*

In general modern bagasse fired boiler plant operates under balanced draught conditions, i.e. a forced draught fan provides the combustion air and an induced draught fan draws the products of combustion over the boiler heating surfaces and exhausts them to atmosphere.

A high pressure (10″ – 30″ water gauge) secondary air fan is often used to inject high velocity air into the furnace to increase turbulence and hence combustion efficiency.

9.51 *Forced Draught Fans*

These are normally high volume, low pressure fans fitted with horsepower limiting backward bladed impellers. To ensure flexible boiler operating characteristics they should be designed to supply at least 15% more air than that required at the design CO_2 at M.C.R. against a pressure 32% in excess of the M.C.R. draught loss. The fans should be capable of providing 100% of the combustion air requirements.

9.52 *Secondary Air Fans*

These are normally low volume, high pressure fans. Their blade shape depends on the duty which they have to perform and hence can vary over a wide range. Since they normally generate fairly high pressure their operating noise levels are high and care should be taken in the design stages to limit this to 95 decibels.

9.53 *Induced Draught Fans*

These are high volume, medium pressure, high temperature fans fitted with forward curved radially tipped blades having anti-fouling and anti-erosion properties. The exhaust gases from bagasse fired boilers contain a particularly abrasive dust and special precautions must be taken in impeller design to minimise erosion, if a life of more than one crop is to be obtained. Renewable anti-erosion stellite ridges placed normal to the direction of the gas flow have proved satisfactory. Peripheral velocities however, should not exceed 15,000 ft. per minute.

They should be designed to handle at least 20% by volume more than that required at the design CO_2 at M.C.R. against a pressure of at least 44% in excess of the M.C.R. draught loss.

9.54 *General*

Boiler fans are driven by means of either steam turbines or electric motors, the latter becoming increasingly more popular.

Electric motors where installed should be of T.E.F.C. weatherproof design which although more expensive initially are the most suitable for operation in the dusty conditions prevalent in a boilerhouse. Fan drives should be carefully chosen in relation to the high inertia masses which they have to accelerate when starting.

A large number of different types of fan controls are available, varying from simple discharge damper controls to complex variable speed drives. In a well balanced factory where a surplus of bagasse can be produced if necessary, simple damper controls are the most effective.

9.6 *Grit Collectors*

In general grit collectors need only be fitted to units employing suspension firing.

Smut particles are highly inflammable and care should be exercised while operating the plant to ensure that hoppers are always free flowing. Unless high efficiencies are required grit refiring is not recommended due to the abrasive nature of the refired particles which increases maintenance costs and also aggravates the fire hazard problem in hoppers from which refiring is taking place.

9.7 *Boiler Valves and Mountings*

A well balanced complement of boiler valves and mountings fitted to a bagasse fired boiler of 100,000 p.p.h. capacity is illustrated in Fig. 9.4. Provision is made for the future installation of an economiser as well as for water sampling, continuous blowdown and high pressure chemical injection. Two absolute water gauges are fitted as well as a high and low level water alarm and remote water level indicator. The feed valve arrangement ensures maximum flexibility.

9.8 *Feed Water System*

To protect boilerplant operating at over 350 p.s.i.g. against oxygen and CO_2 corrosion deaerating plant should be installed in the feed circuit. A simple but effective arrangement which provides sufficient storage to cover most emergency conditions is illustrated in Fig. 9.5. A pressure operated auto cut-in gear should be fitted to the turbo feed pump to ensure that an adequate supply of feed water is always available. This gear should of course be checked once per shift for satisfactory operation. Feed pumps should be sized to deliver at least 15% more than the M.C.R. rating of the plant to cater for load surges.

9.9 *Instruments and Controls*

Boiler instruments and controls, which should preferably be centralised, should be kept as simple as possible. Only those instruments which materially affect boiler operation should be fitted. Log sheets should be designed to ensure that vital operating factors, such as bearings, etc are checked and their condition logged at least once every hour.

A well engineered complement of instruments and controls should not exceed the following:

Instruments:

(*a*) Pressure Indicating:
1) Superheater outlet steam pressure;
2) Boiler drum pressure;
3) Feed water mains pressure.

(*b*) Temperature Indicating:
1) Final steam temperature;
2) Feed water temperature;
3) Economiser water outlet temperature (if economiser fitted);
4) Boiler outlet gas temperature;
5) Air heater outlet gas temperature (if air heater fitted);
6) Final exhaust gas temperature;
7) Ambient air temperature;
8) Air heater air outlet temperature (if air heater fitted).

(*c*) Level Indicating:
1) Local boiler drum level indicator;
2) Remote boiler drum level indicator;
3) Feed tank and deaerator level indicators.

(*d*) Power Indicating:
1) I.D. fan amps;
2) F.D. fan amps;
3) Other fan amps;
4) Feed booster pump amps;
5) Electro feed pump amps;
6) Fuel feeder amps.

Recorders:
1) Combined steam flow/steam pressure recorder.

Audible Alarms:
1) Boiler low/high level water alarm.

Visual Alarms:
1) Feedwater tank low/high level alarm;
2) Deaerator low/high level alarm;
3) Bagasse supply failure alarm.

Analysers:
1) Portable hand-operated CO_2 analyser for spot checks.

Auto controls:
1) Boiler water level control;
2) Feed tank level control;
3) Deaerator level control;
4) Steam pressure control based on simple air/fuel ratio controllers;
5) Furnace pressure controller.

Conclusions

A wide range of boiler plant is available to suit most factory requirements; the choice being dependent upon local conditions. Where positive economic thinking dictates that only bagasse shall be burnt as a fuel, the boiler fitted with a self-feeding furnace very adequately meets these requirements. Further in areas where skilled operators are at a premium, this unit provides broad operating characteristics without the need for sophisticated control systems. In areas where skilled labour is readily available and a maximum degree of mechanisation is required, the air glide furnace with dump grate presents a reasonable solution to the problem. If, however, the burning of auxiliary fuels is to be avoided with this unit an assured bagasse supply must be available. Fig. 10.1 illustrates a possible solution to this problem.

Oil burners for use during the Off-Crop, can be fitted to either of these two units at a very small increase in capital cost.

Auxiliary coal firing can only be accommodated readily in a unit of the type illustrated in Fig. 9.2C. The unit is considerably more expensive and is essentially a compromise design.

Since auxiliary fuels in a well balanced factory should only be burnt during the off crop, the off crop load should be carefully assessed, especially where coal is the auxiliary fuel, in order to establish whether it would not be more economical to install a separate coal fired boiler to cater for this load at maximum efficiency and minimum capital outlay.

Modern bagasse fired water tube boilerplant can be constructed with unit capacities in the range 20,000 p.p.h. to 300,000 p.p.h. This means that factories milling up to 300 tons of cane per hour can be served, if necessary by one boiler. Reference to Fig. 10.2 will indicate that this is in fact the cheapest installation, but it also necessitates putting "all one's eggs in one basket". The final choice must rest with the factory management. However, since reliability of plant is increasing continuously, large unit sizes should be carefully considered.

Summary

The paper is intended to provide a useful reference work on modern steam generating plant used in the Cane Sugar Industry.

The paper is divided into three sections:
The first section deals with the

1.00 Properties and Uses of Steam.
2.00 Fuels.
3.00 Combustion.
4.00 Combustion Equipment.
5.00 Fundamental Heat Transfer Relationships.
6.00 Friction Loss; and
7.00 Circulation Relationships

used in boiler design.

The second section deals firstly with

8.00 Overall boiler heat balance, and secondly, with

9.00 Boiler heating surface disposition and structural design.

In the last section a number of designs are analysed and possible future trends discussed.

10.00 Conclusions.

Acknowledgments

I would like to thank John Thompson Africa (Pty) Ltd. for allowing me to publish this paper and my colleagues D. J. van Blerk and D. A. Jones for their assistance in its preparation.

List of Symbols

A	Area through which heat is transferred normal to the direction of heat transfer, absorbed or reflected normal to the radiating body.	ft^2
C_p	Specific heat of fluid at constant pressure.	BTU/lb.
D	Outside diameter of tubes over which fluid flows.	ft.
F_a	Radiation geometric factor introduced in order to extend the Stefan-Boltzmann relationship to cases other than infinite parallel planes.	dimensionless
F_{e_1}	Emissivity factor introduced in order to extend the Stefan-Boltzmann relationship to bodies other than black bodies.	dimensionless
F_e	Corrected emissivity factor for use in equation 5.8	dimensionless
G	Mass gas flow	$lb/ft.^2hr.$
H_s	Total heat of dry saturated steam	BTU/lb.
h	Total conductive heat transfer co-efficient	BTU/ft.$^{2\circ}$F hr.
h_c	Convective heat transfer co-efficient	BTU/ft.$^{2\circ}$F hr.
h_g	Gas side heat transfer co-efficient	BTU/ft.$^{2\circ}$F hr.
h_r	Radiant heat transfer co-efficient	BTU/ft.$^{2\circ}$F hr.
h_s	Sensible heat in steam	BTU/lb.
h_1	Conductive heat transfer co-efficient through body 1	BTU/ft.$^{2\circ}$F hr.
h_2	Conductive heat transfer co-efficient through body 2	BTU/ft.$^{2\circ}$F hr.
k	Thermal conductivity	BTU/ft.hr. °F
L	Effective vertical length of downcomers	ft.
P	Pressure	P.S.I.A.
Q	Heat transferred	BTU
q	Heat transferred per unit time	BTU/hr.
T	Temperature	°F absolute
T_a	Ambient air temperature	°F. absolute
T_{fe}	Temperature of gases leaving furnace	°F. absolute
T_{fg}	Temperature of furnace gases	°F. absolute
T_w	Temperature of tube walls absorbing heat	°F. absolute
T_1, T_2	Temperatures of source and sink respectively	°F. absolute
$\triangle t$	Temperature difference between hot and cold faces of a body through which heat is flowing	°F.
t_s	Saturation temperature of steam at pressure P	°F.
V	Volume of gas	ft^3
V_s	Specific volume of dry saturated steam	$ft^3/lb.$
v	Velocity of fluid flowing	ft/sec.
u	Specific volume of water	ft^3/lb
u_m	Specific volume of water/steam mixture	ft^3/lb
W_f	Weight of fuel burnt	lb/hr.
W_g	Weight of gas per lb. of fuel in combustion chamber	lb.
x	Distance between hot and cold faces	ft.
t	Unit of time	secs.
μ	Viscosity of fluid	lb/ft. sec.
ρ	Density of fluid	lb/ft^3
ρ_m	Density of steam/water mixture	lb/ft^3
σ	.1723 Stefan-Boltzmann constant	
θ	Unit of time	hr.

Abbreviations:

p.p.h.	pounds per hour
M.C.R.	Maximum continuous rating
N.C.V.	Nett calorific value
G.C.V.	Gross calorific value

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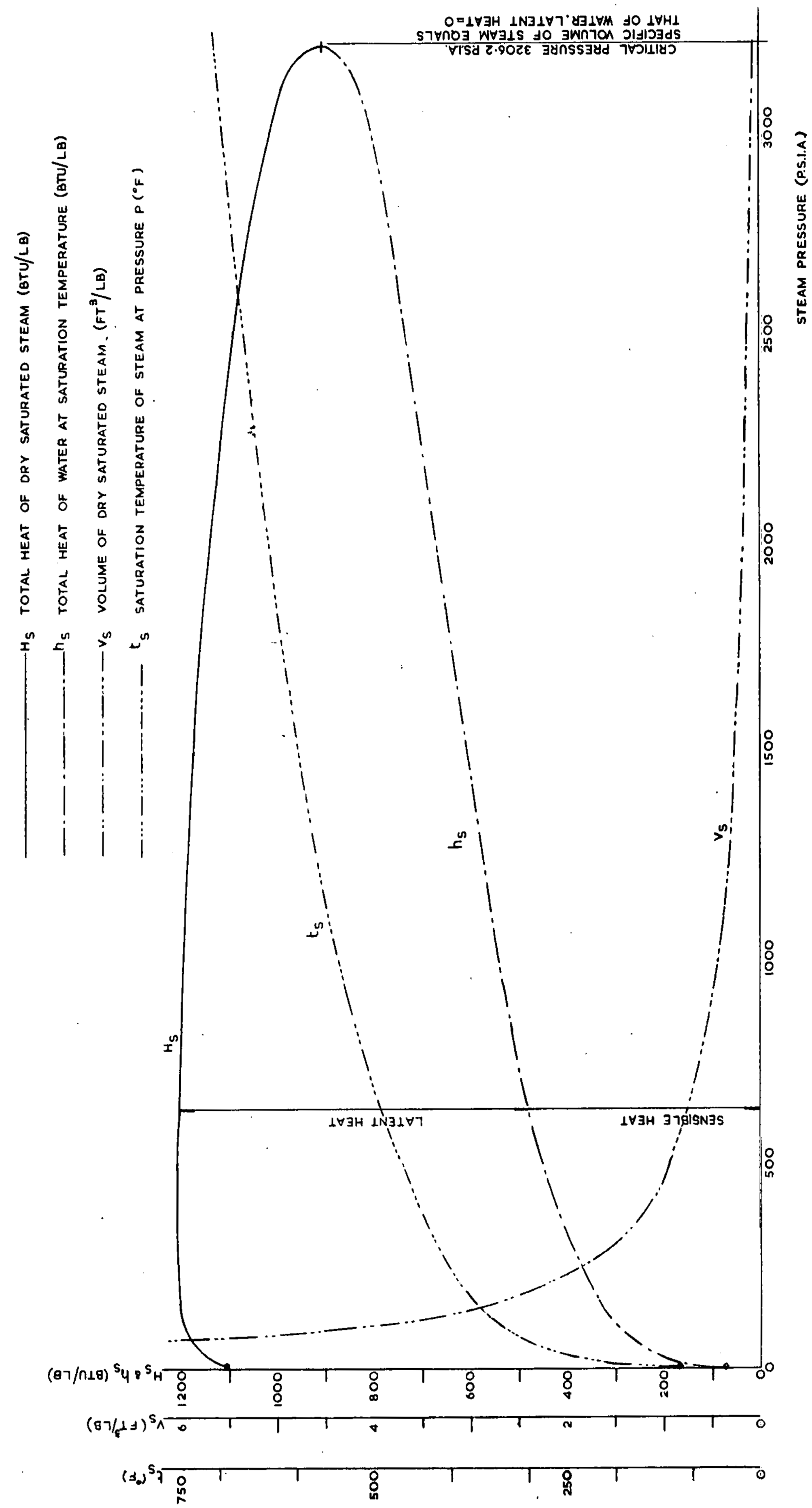


FIGURE 1.1: Curves showing Properties of Steam in Relation to Absolute Pressure.
BASIS: CALLENDAR STEAM TABLES 1939

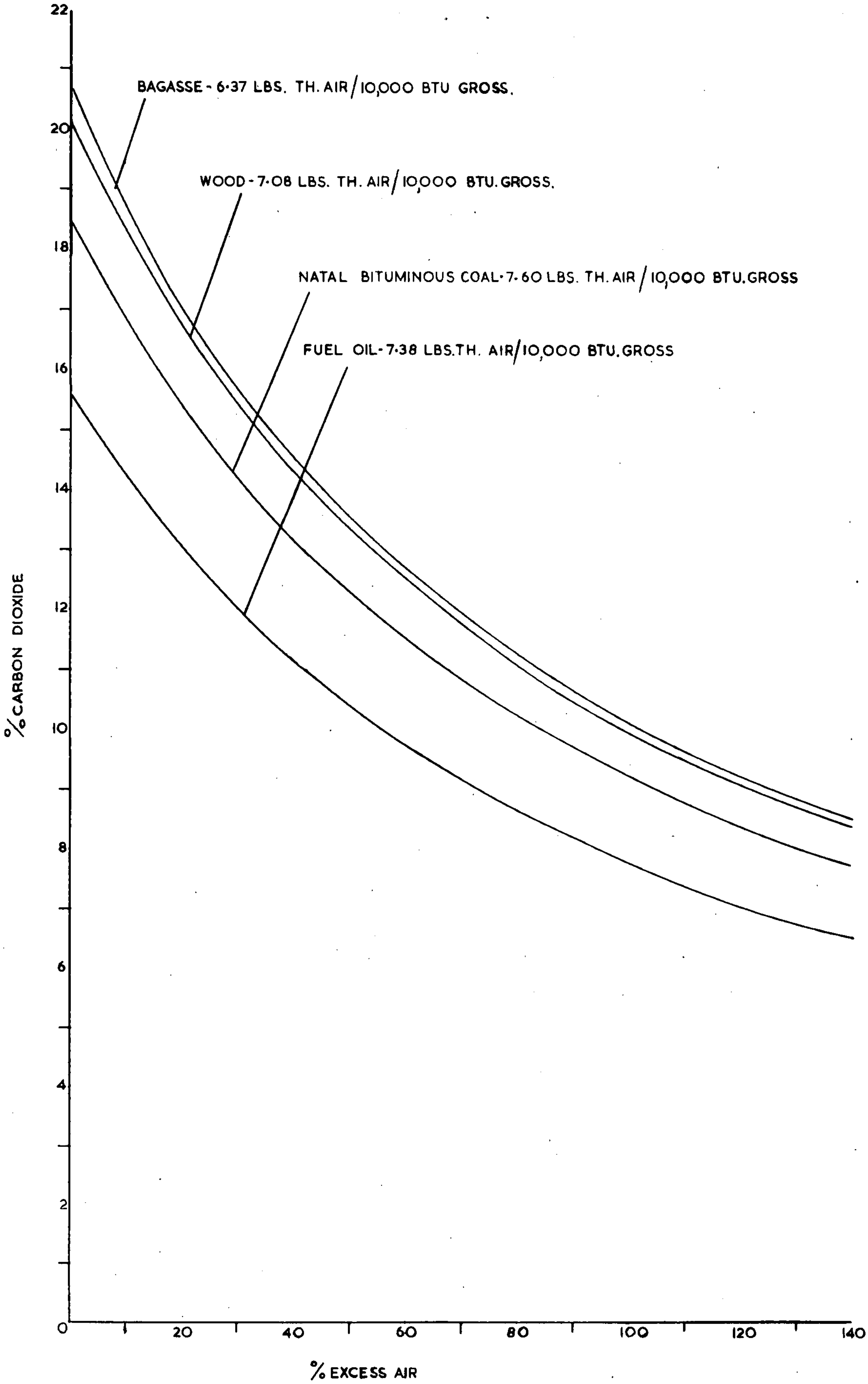
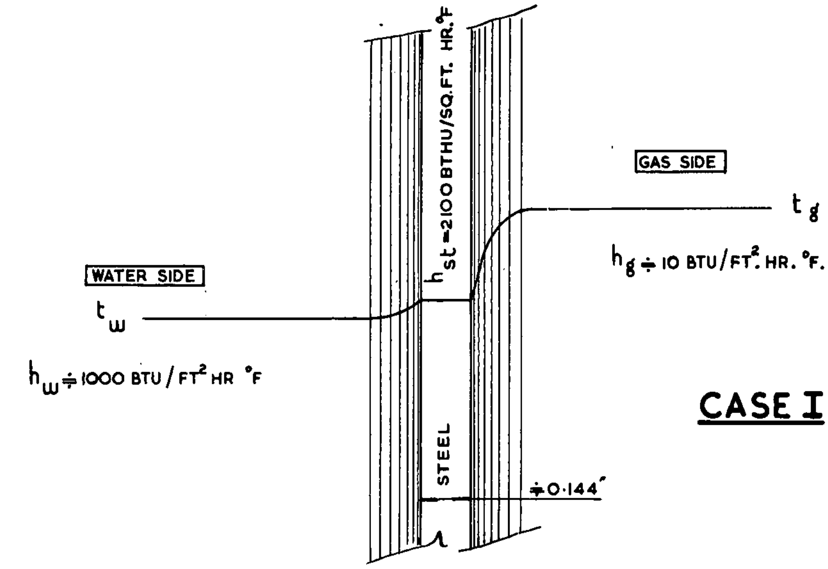


FIGURE 3.1: Curve showing Relation between % Excess Air and % Carbon Dioxide for a Number of Typical Fuels.

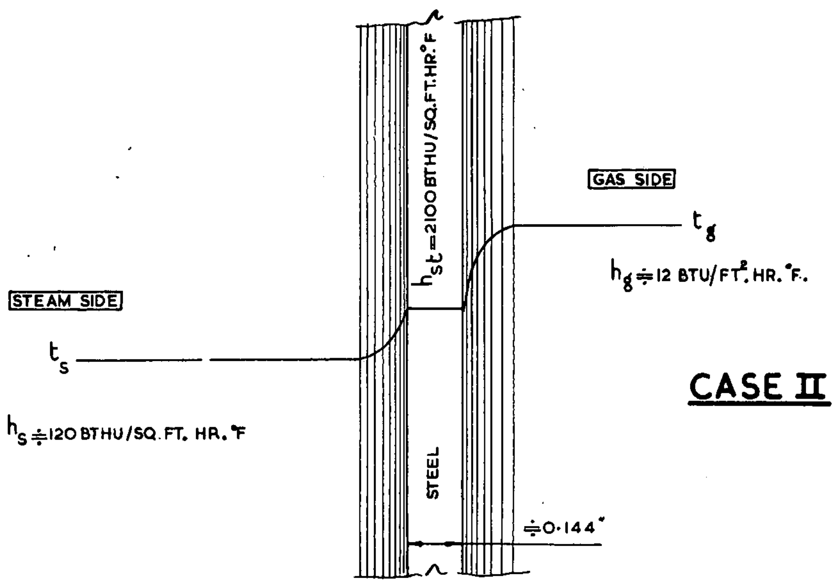


$$\frac{1}{h_{\text{OVERALL}}} = \frac{1}{10} + \frac{1}{2100} + \frac{1}{1000} = \underline{0{\cdot}10148}$$

$$\therefore h_{\text{OVERALL}} = \underline{9{\cdot}85 \text{ BTU/FT}^2 \text{ HR. °F.}}$$

USING GAS SIDE HEAT TRANSFER COEFFICIENT INTRODUCES
AN ERROR OF ONLY 1·5 %

NOTE: METAL TEMPERATURE IN CASE II
HIGHER THAN IN CASE I .



$$\frac{1}{h_{\text{OVERALL}}} = \frac{1}{12} + \frac{1}{2100} + \frac{1}{120} = \underline{0{\cdot}09222}$$

$$\therefore h_{\text{OVERALL}} = \underline{10{\cdot}83 \text{ BTU/FT}^2 \text{ HR. °F.}}$$

USING GAS SIDE HEAT TRANSFER COEFFICIENT INTRODUCES
AN ERROR OF 9·3 %

FIGURE 5.1: Illustration showing the effect of the Heat Transfer Coefficients encountered in Boiler Design.

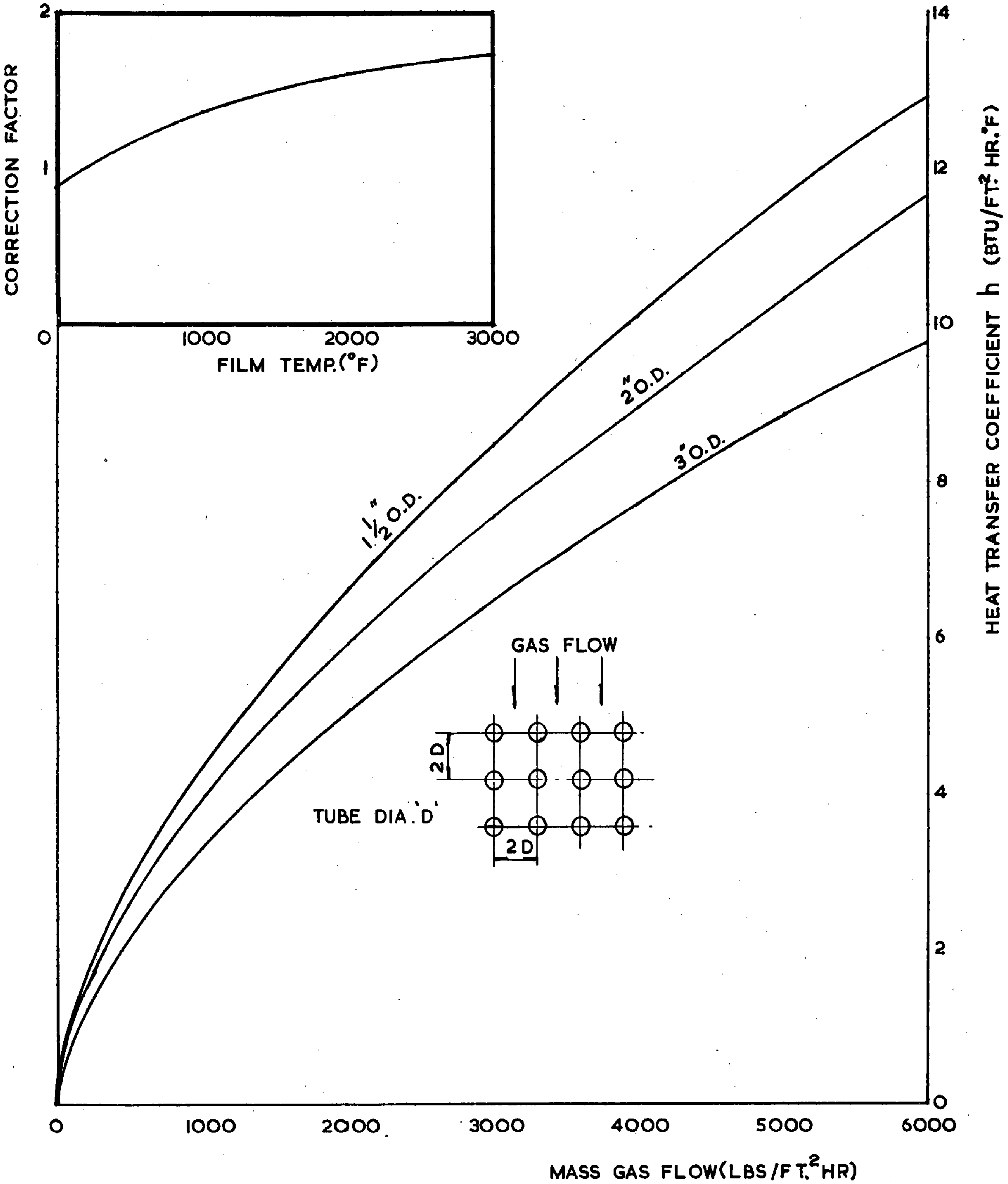


FIGURE 5.2: Heat Transfer Coefficient for Flue Gas Flowing at Different Mass Velocities across a Bank of Tubes (Mean Film Temp. 200° F.) at least Six Rows Deep Squâre pitched at 2D

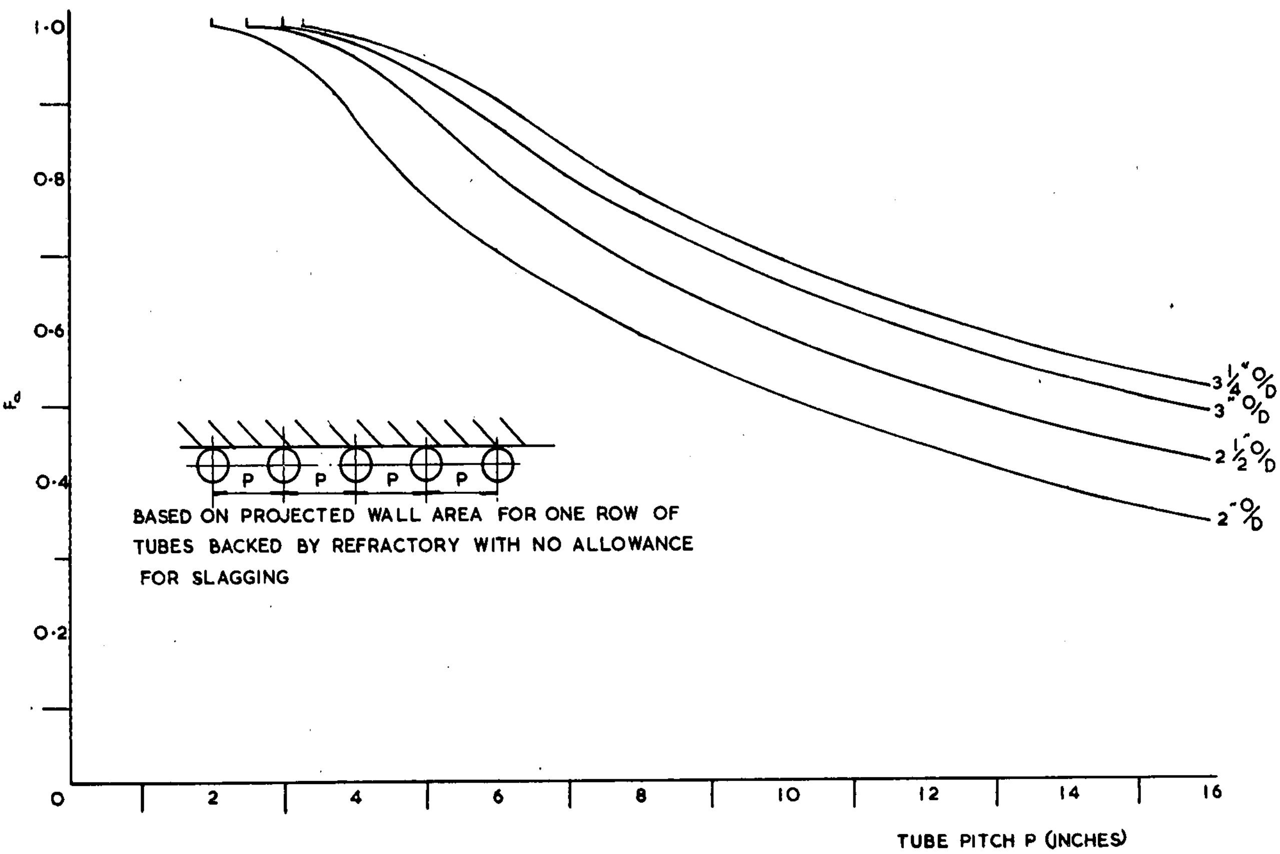


FIGURE 5.3: Factor for Waterwall Heating Surfaces for Bare Tubes.

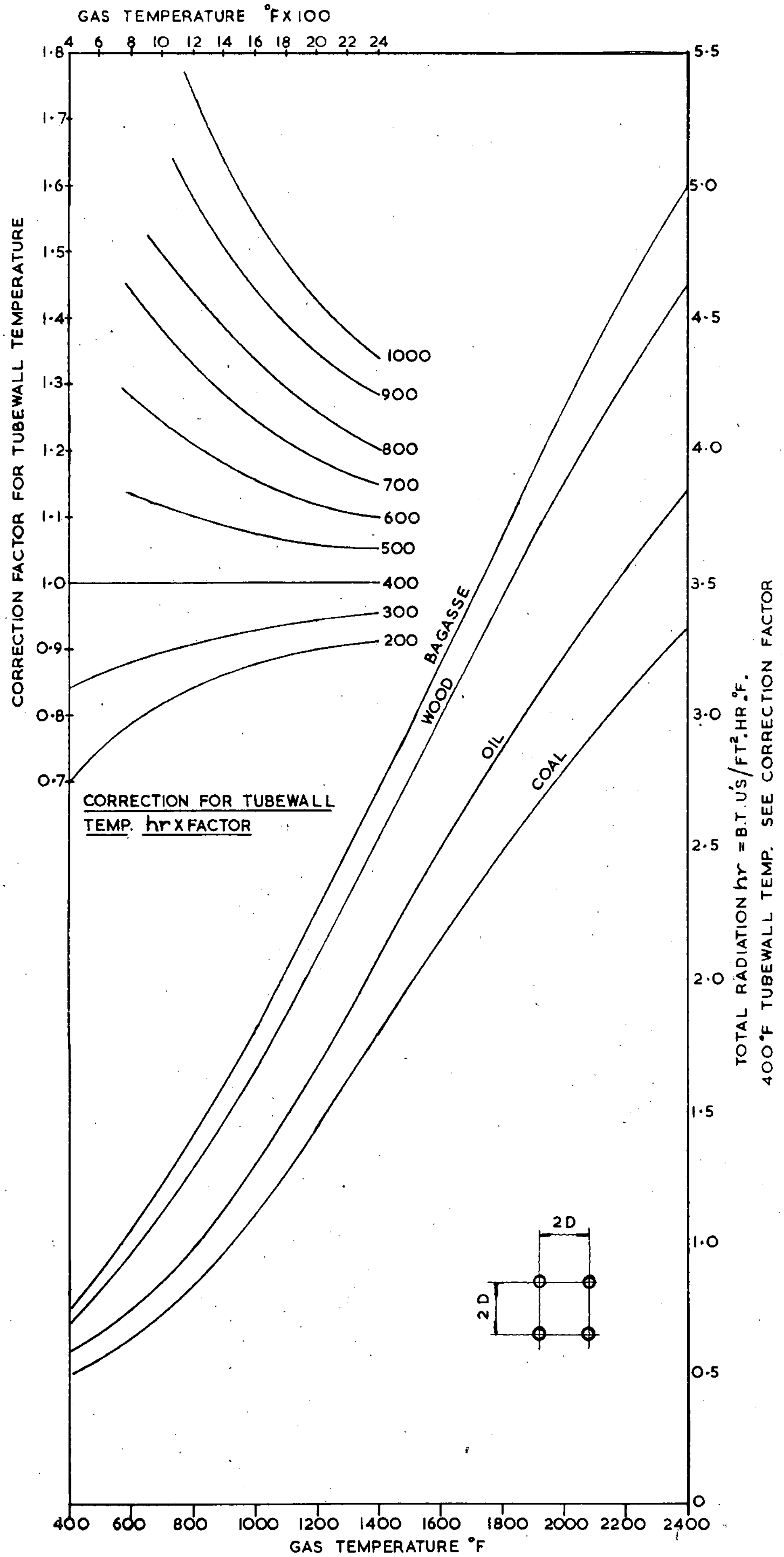
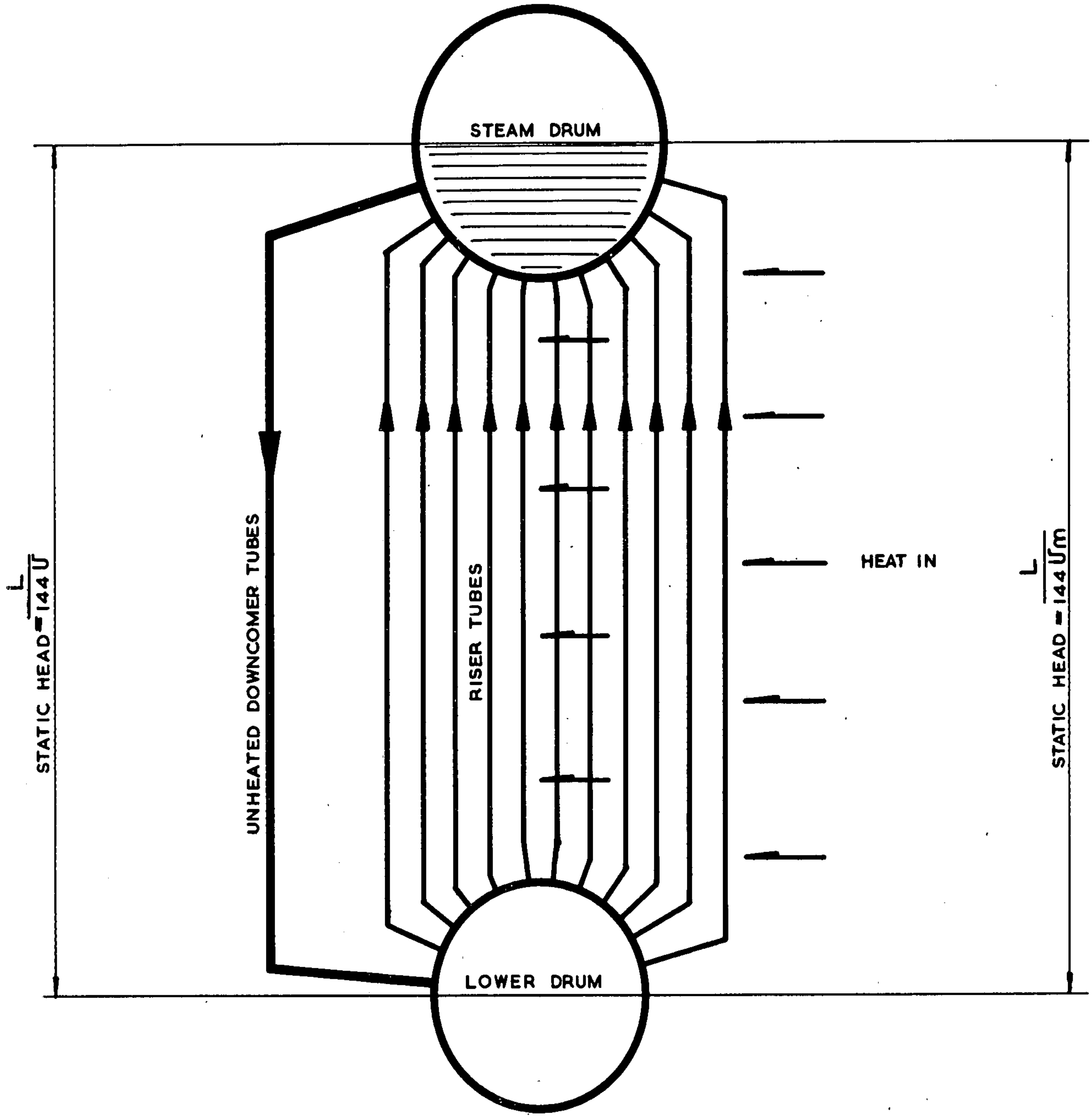


FIGURE 5.4: Curve showing Radiation from Flue Gas (CO_2 & H_2O)



NETT STATIC HEAD = $\frac{L}{144}(U - U_m)$

= FRICTION LOSSES, ENTRANCE & EXIT LOSSES, ACCELERATION LOSSES & BEND LOSSES.

$= f(q, P, P_m)$ ——

IF TUBES ARE NOT VERTICAL $L = L_i \sin\theta$

IE. THE STATIC HEAD PRODUCING CIRCULATION IS REDUCED

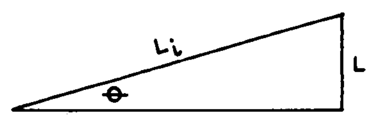
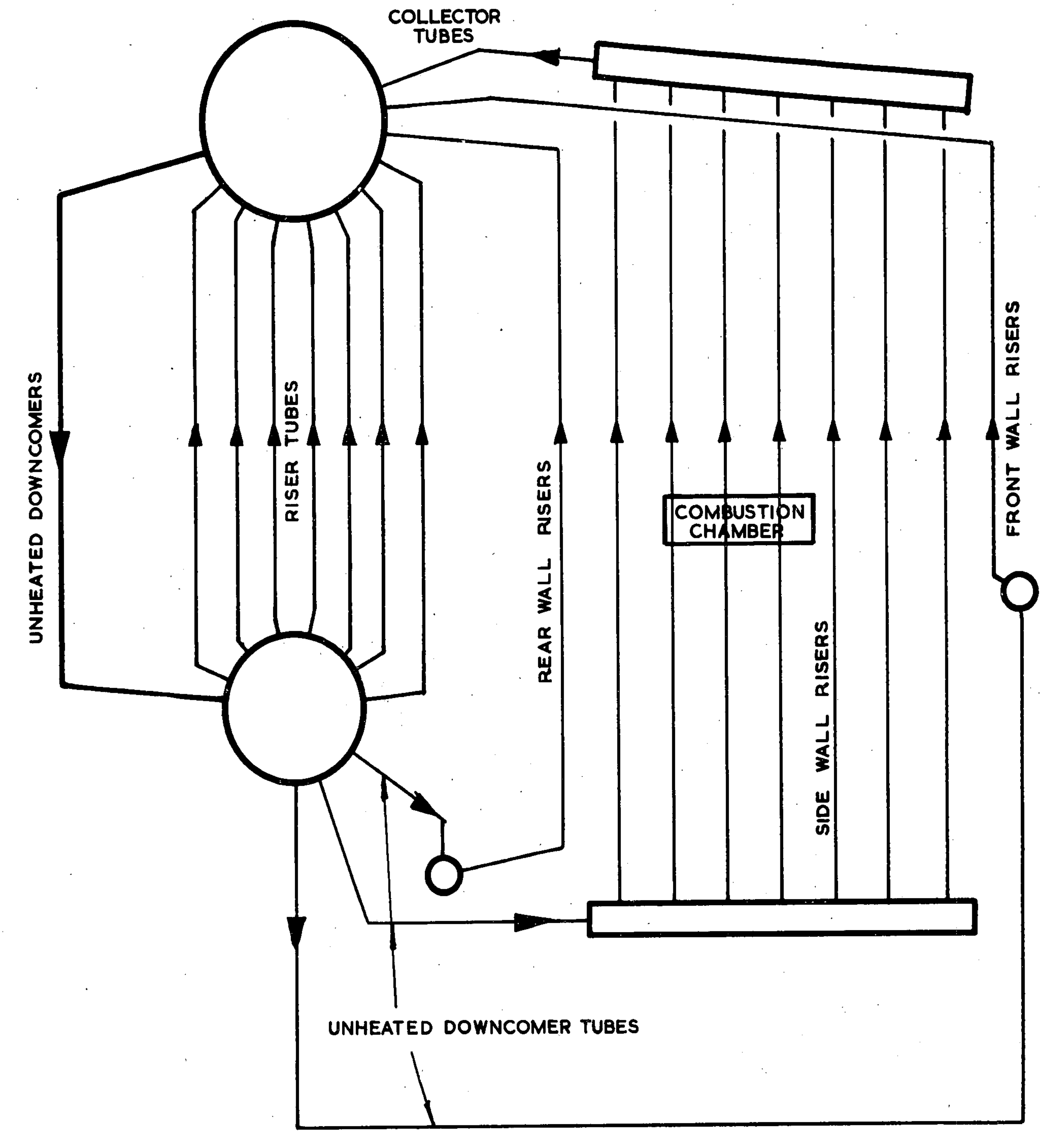


FIGURE 7.1: Simple Boiler Circuit.



The system comprises a number of parallel heated circuits which generate a potential in the form of a static head. The flow through each circuit is controlled by careful sizing of the downcomer tubes to provide the necessary system resistance thus ensuring adequate water supply to each circuit and preventing overheating of the tubes.

FIGURE 7.2—Typical Boiler Circuit

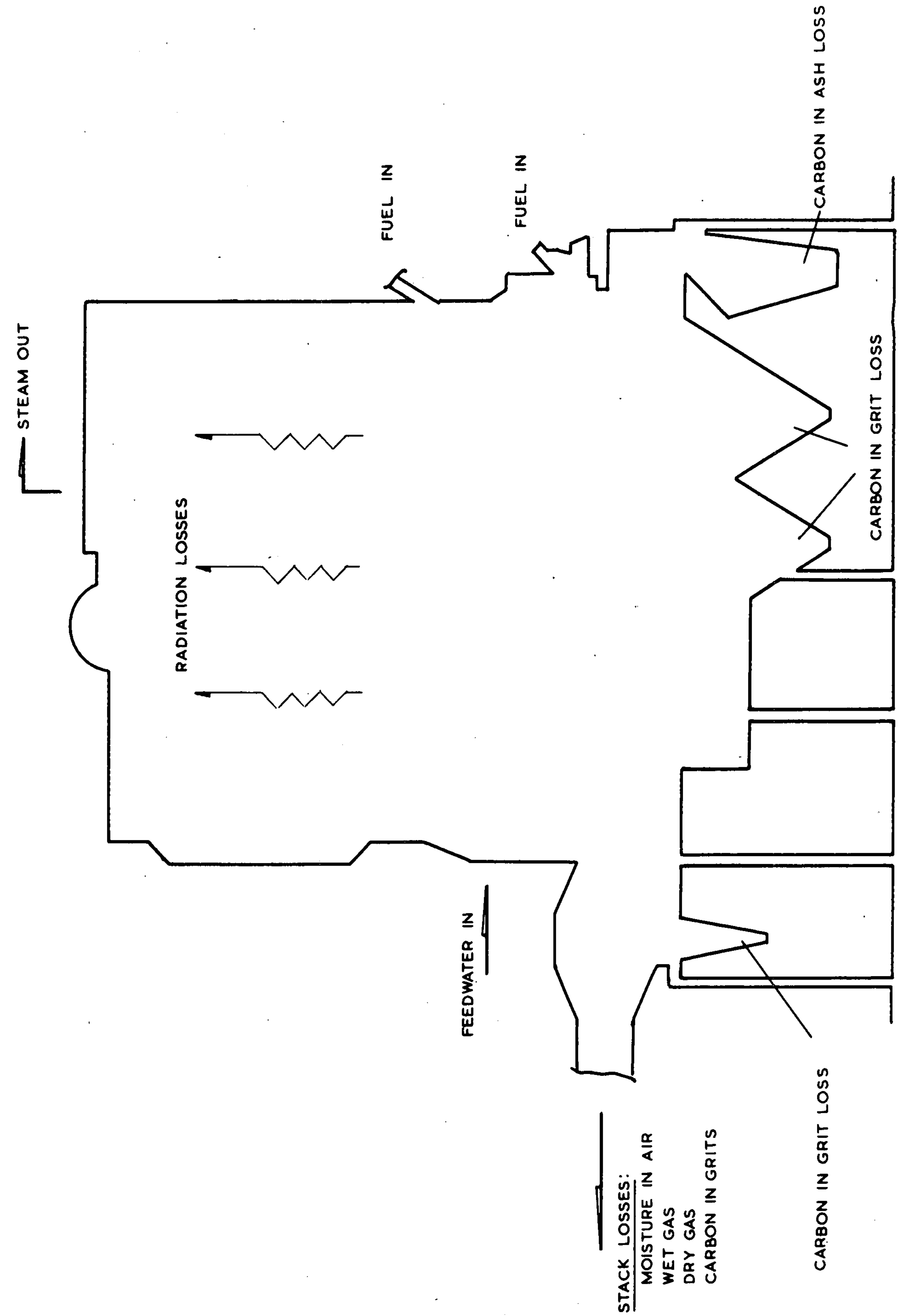


FIGURE 8.1: Factors affecting Boiler Heat Balance.

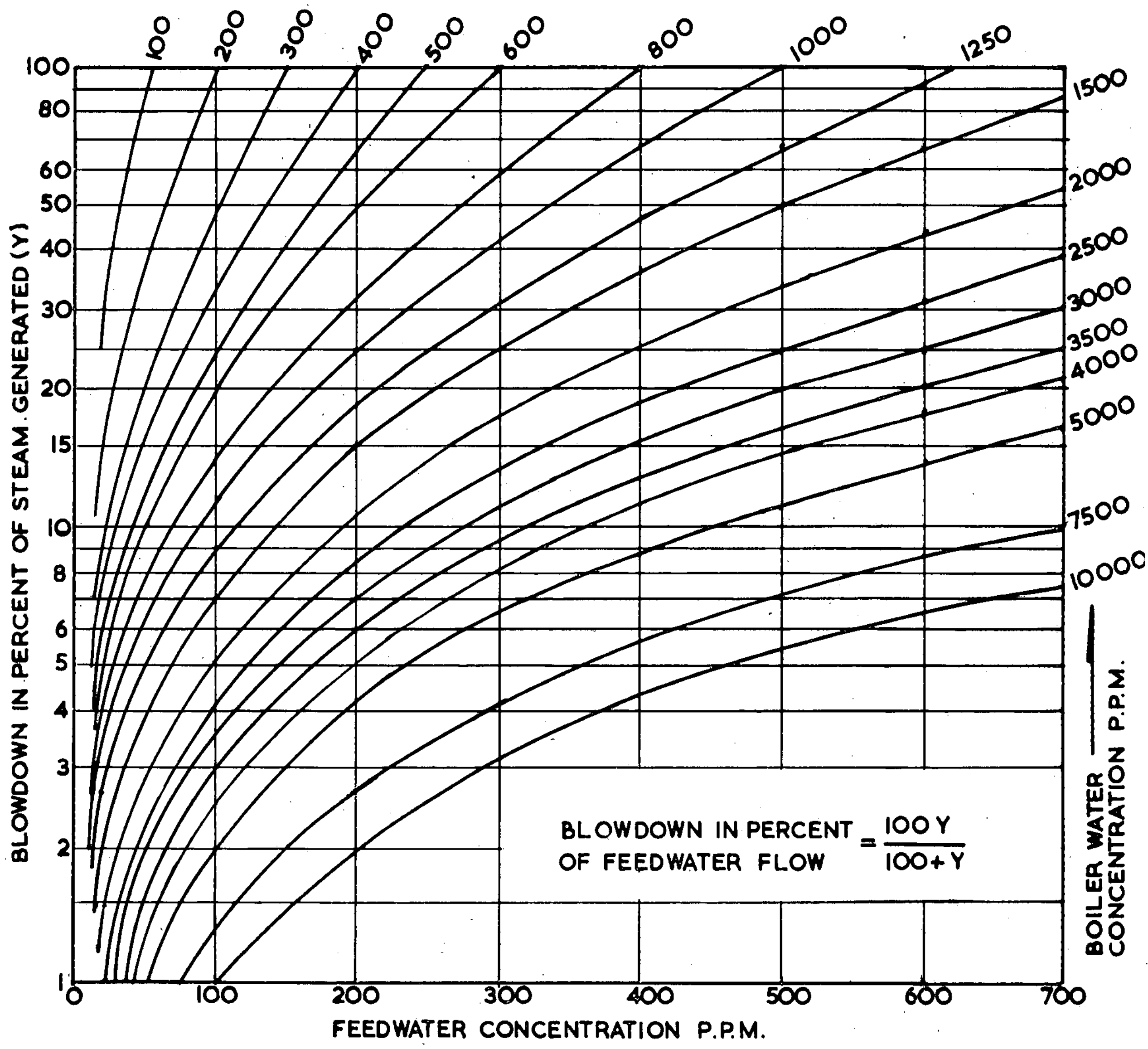


FIGURE 8.2: Blowdown Rate as a function of Feed Water Concentration.

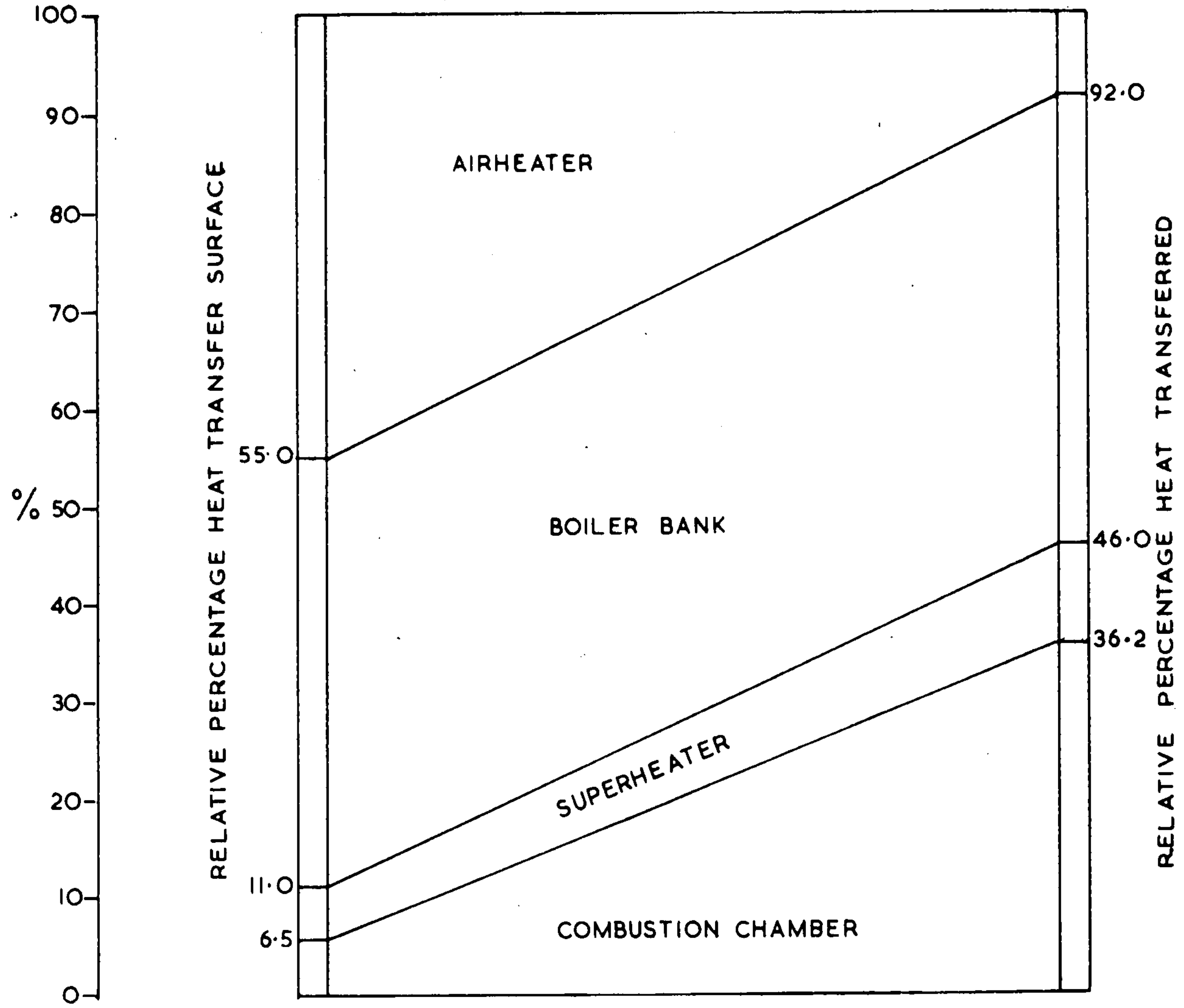


FIGURE 9.1 A: Chart showing the effectiveness of Heat Transfer Surfaces.

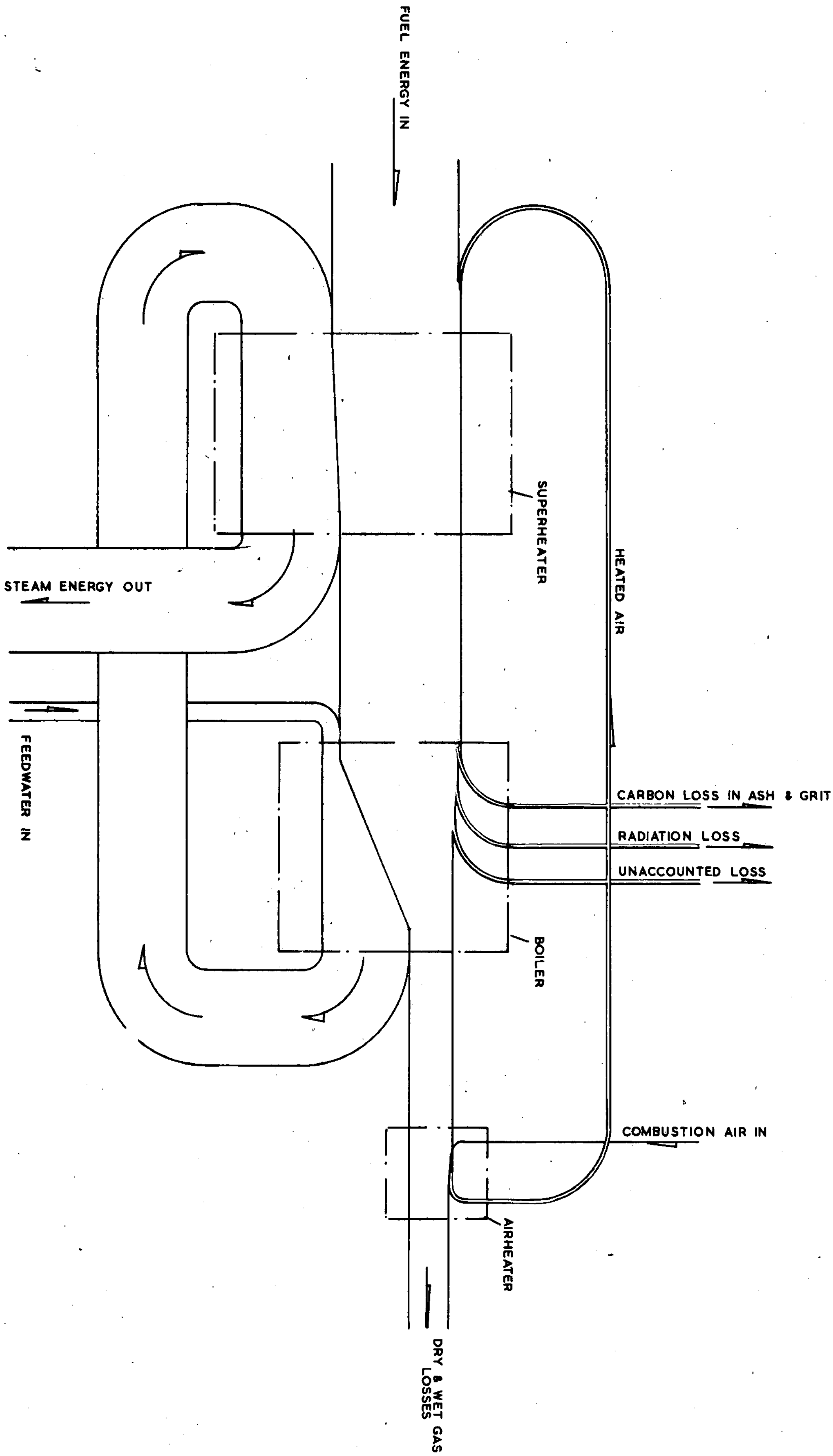


FIGURE 9.1 B: Energy Flow Diagram.

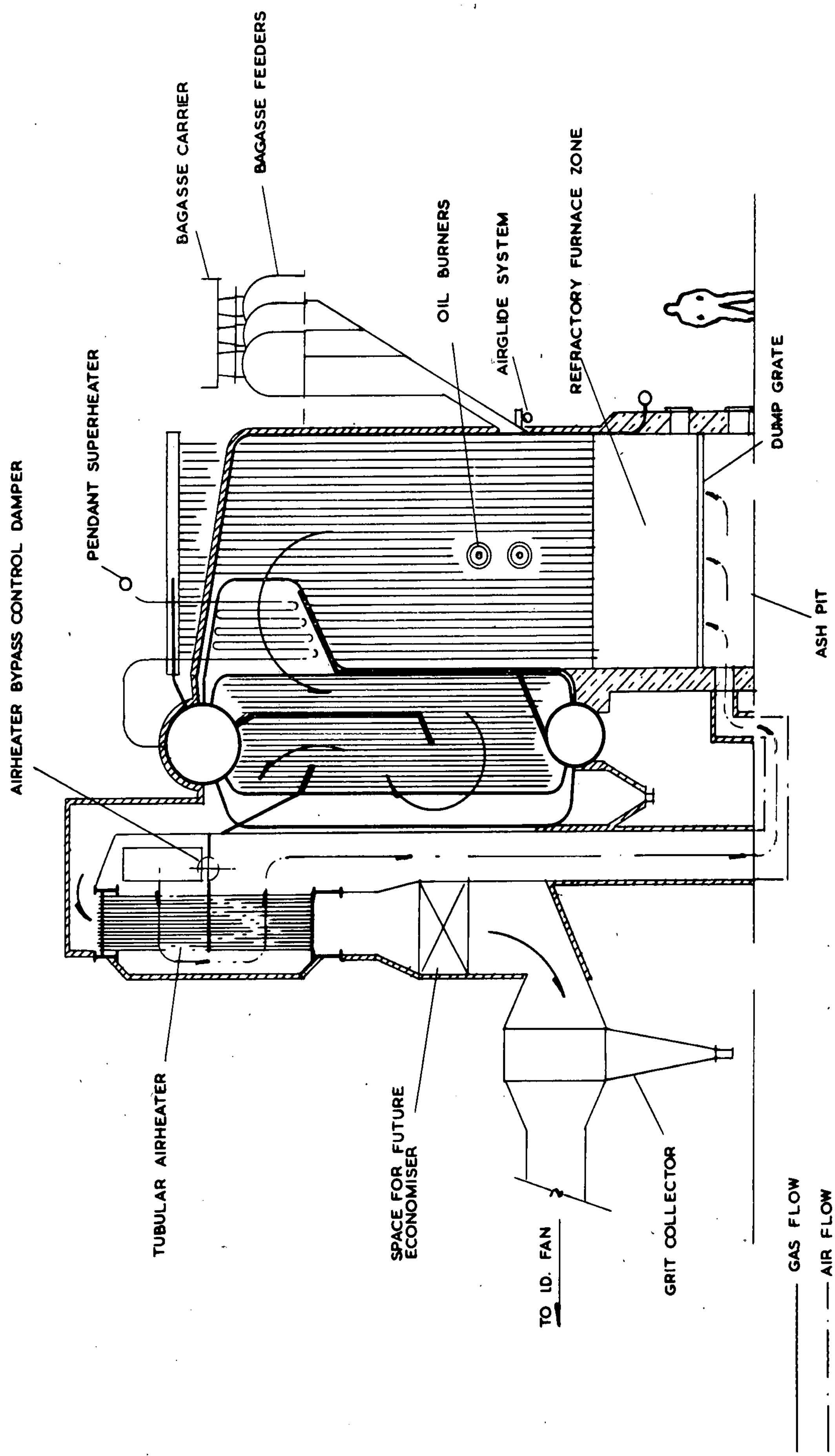


FIGURE 9.2 A: Bagasse Fired Boiler with Dump Grate and Auxiliary Oil Burners.

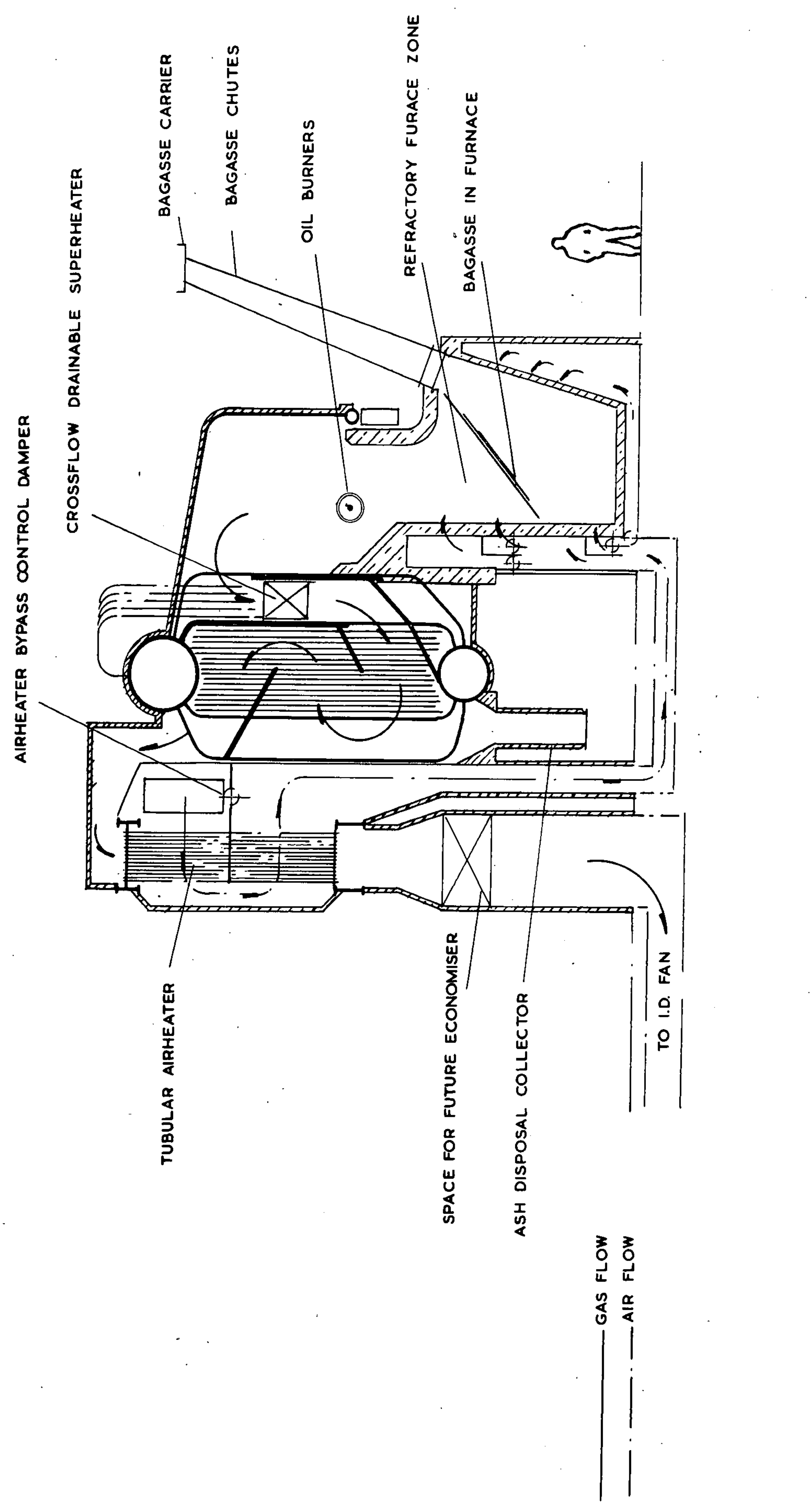


FIGURE 9.2 B: Bagasse Fired Boiler with Self Feeding Furnace and Auxiliary oil burners.

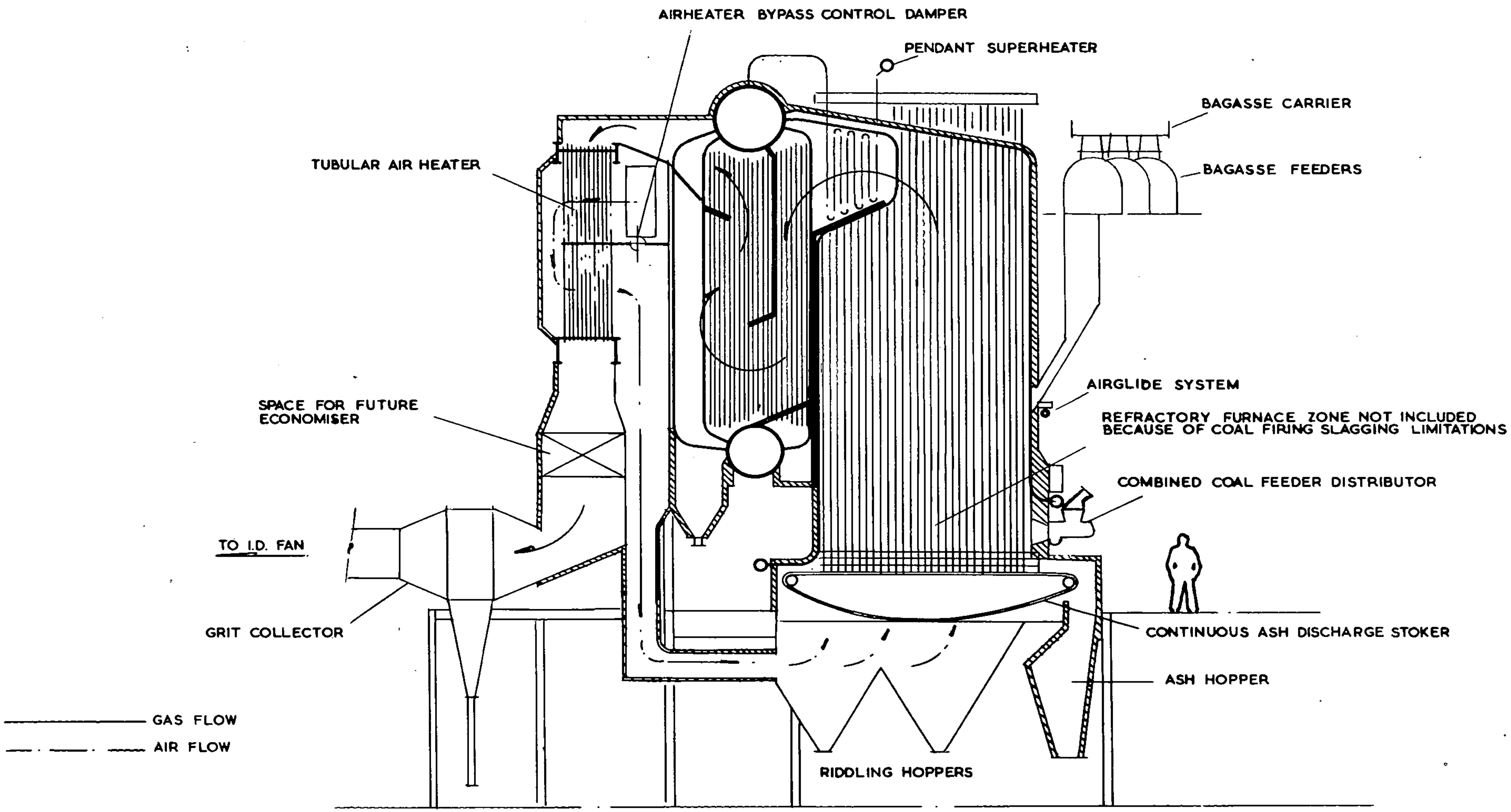


FIGURE 9.2 C: Bagasse Fired Boiler with continuous Ash Discharge Stoker and Auxiliary Coal Firing Equipment.

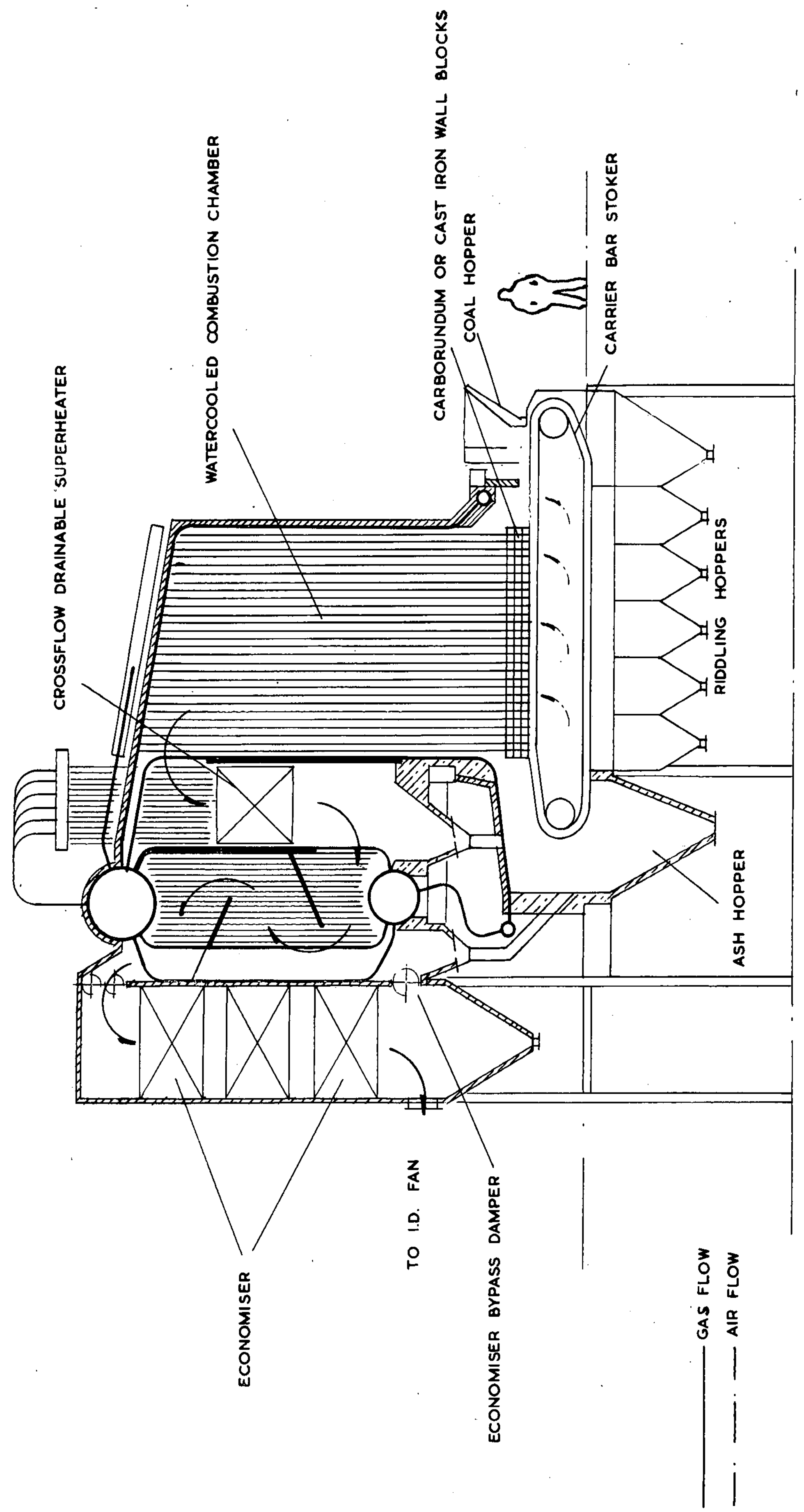


FIGURE 9.2 D: Coal Fired Boiler.

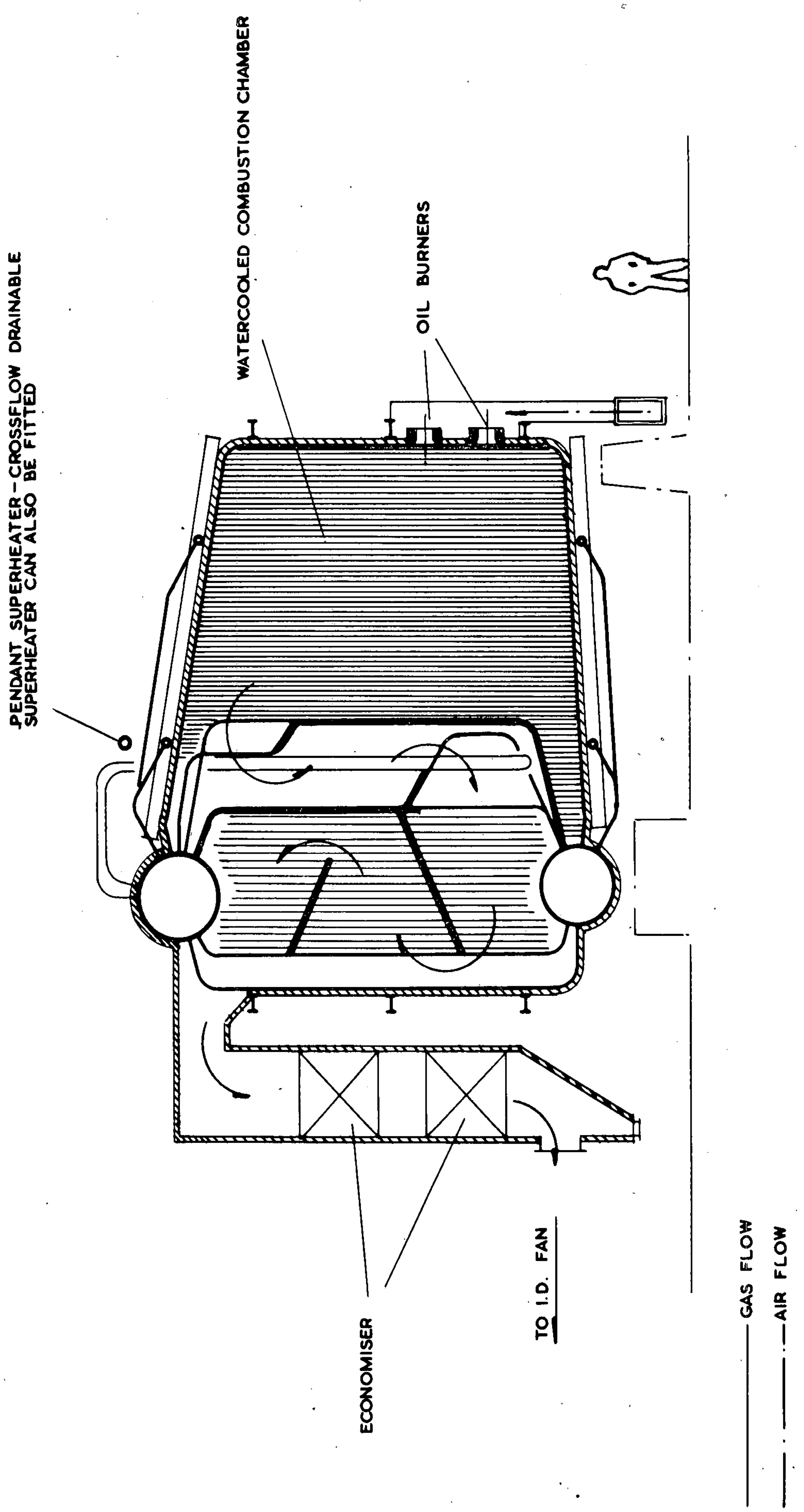


FIGURE 9.2 E: Oil Fired Boiler.

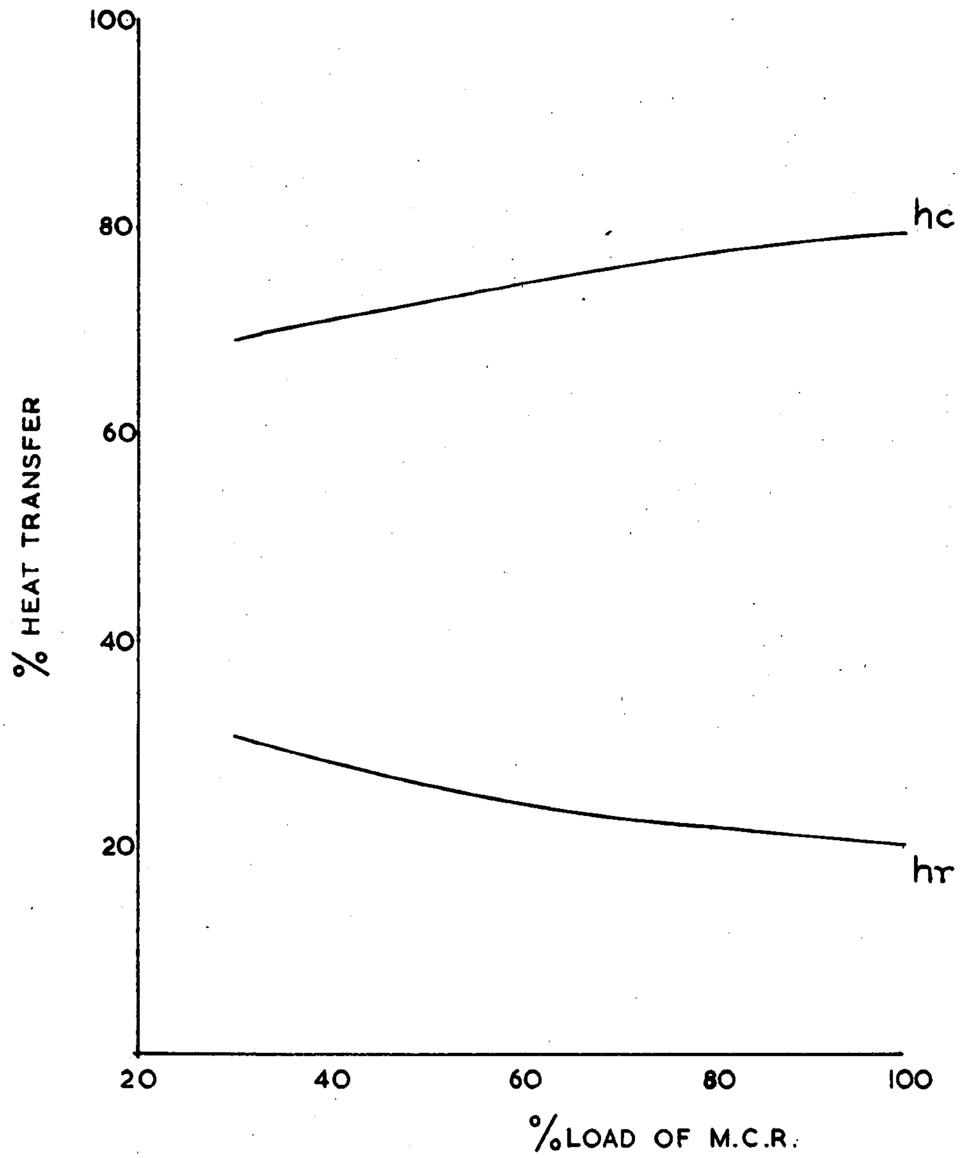
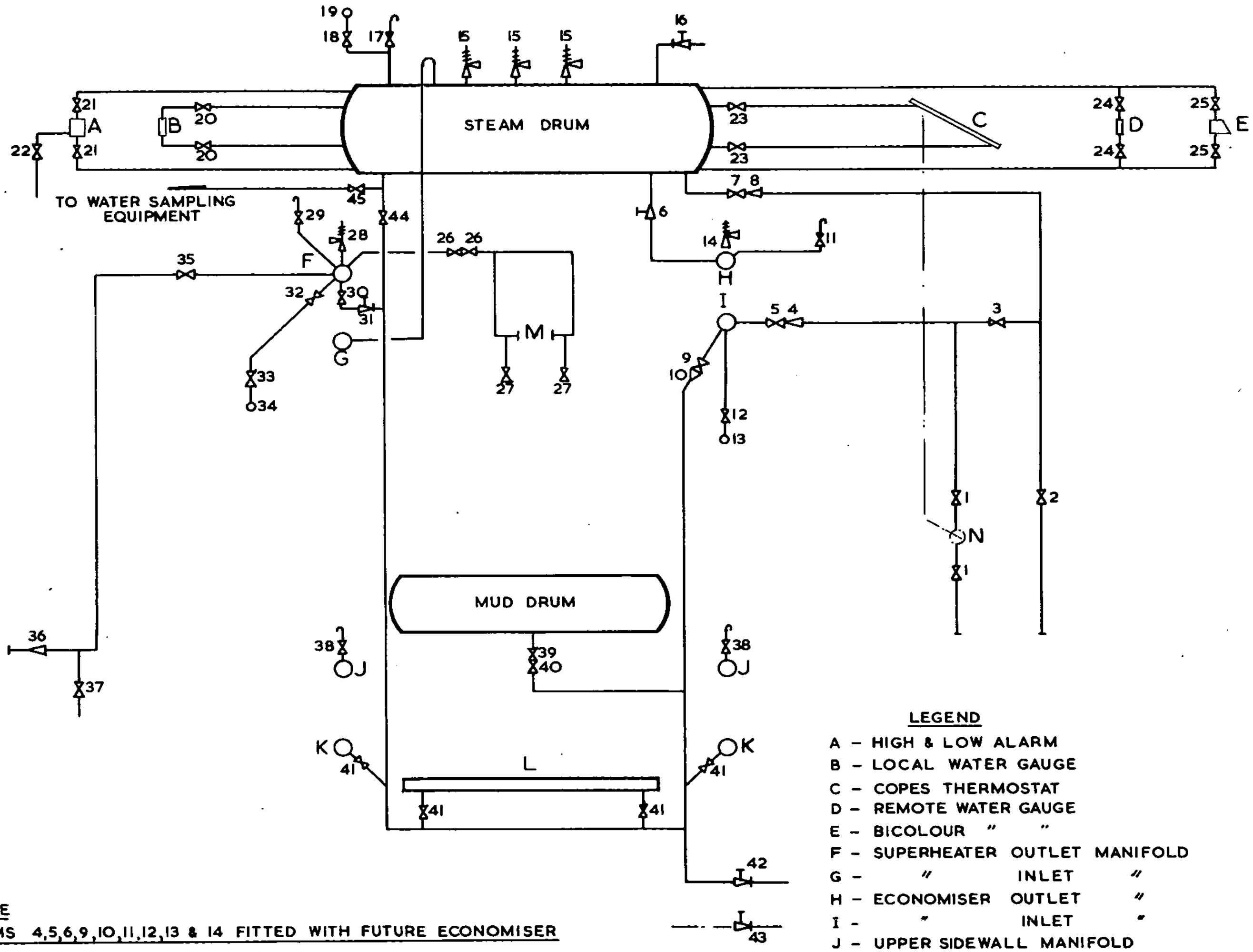


FIGURE 9.3: Effect of load on Radiation and Convection Heat Transfer Co-efficients.



LEGEND

A – HIGH & LOW ALARM
B – LOCAL WATER GAUGE
C – COPES THERMOSTAT
D – REMOTE WATER GAUGE
E – BICOLOUR " "
F – SUPERHEATER OUTLET MANIFOLD
G – " INLET "
H – ECONOMISER OUTLET "
I – " INLET "
J – UPPER SIDEWALL MANIFOLD
K – LOWER " "
L – FRONTWALL "
M – SOOTBLOWERS
N – COPES FEED WATER REGULATOR

NOTE
ITEMS 4,5,6,9,10,11,12,13 & 14 FITTED WITH FUTURE ECONOMISER

ITEM	DESCRIPTION
1	COPES ISOLATING
2	AUXILLIARY FEED
3	COPES BYPASS
4	ECONOMISER INLET N.R.V.
5	ECONOMISER ISOLATING
6	MAIN FEED ISOLATING
7	AUXILLIARY FEED
8	" " N.R.V.
9	DRAIN VALVE
10	DRAIN N.R.V.
11	AIR RELEASE
12	P.G. ISOLATING
13	PRESSURE GAUGE
14	SAFETY VALVE
15	SAFETY VALVES
16	CHEMICAL DOSING
17	AIR RELEASE
18	P.G. ISOLATING
19	PRESSURE GAUGE
20	WATER GAUGE ISOLATING
21	HIGH & LOW ALARM ISOLATING
22	" " " DRAIN
23	COPES THERMOSTAT ISOLATING
24	REMOTE LEVEL GAUGE ISOLATING
25	BICOLOUR LEVEL GAUGE ISOLATING
26	SOOTBLOWER CONTROL
27	" " DRAIN
28	SAFETY VALVE
29	AIR RELEASE
30	SUPERHEATER DRAIN
31	" " S.D.N.R.
32	P.G. ISOLATING
33	" "
34	PRESSURE GAUGE
35	MAIN STEAM STOP
36	" " N.R.V.
37	" " DRAIN
38	AIR RELEASE
39	BLOWDOWN ISOLATING
40	BLOWDOWN
41	"
42	H.P. DRAIN S.D.N.R.
43	L.P. " "
44	CONTINUOUS BLOWDOWN
45	WATER SAMPLING ISOLATING

FIGURE 9.4: Diagrammatic Arrangement of Boiler Mountings and Valves.

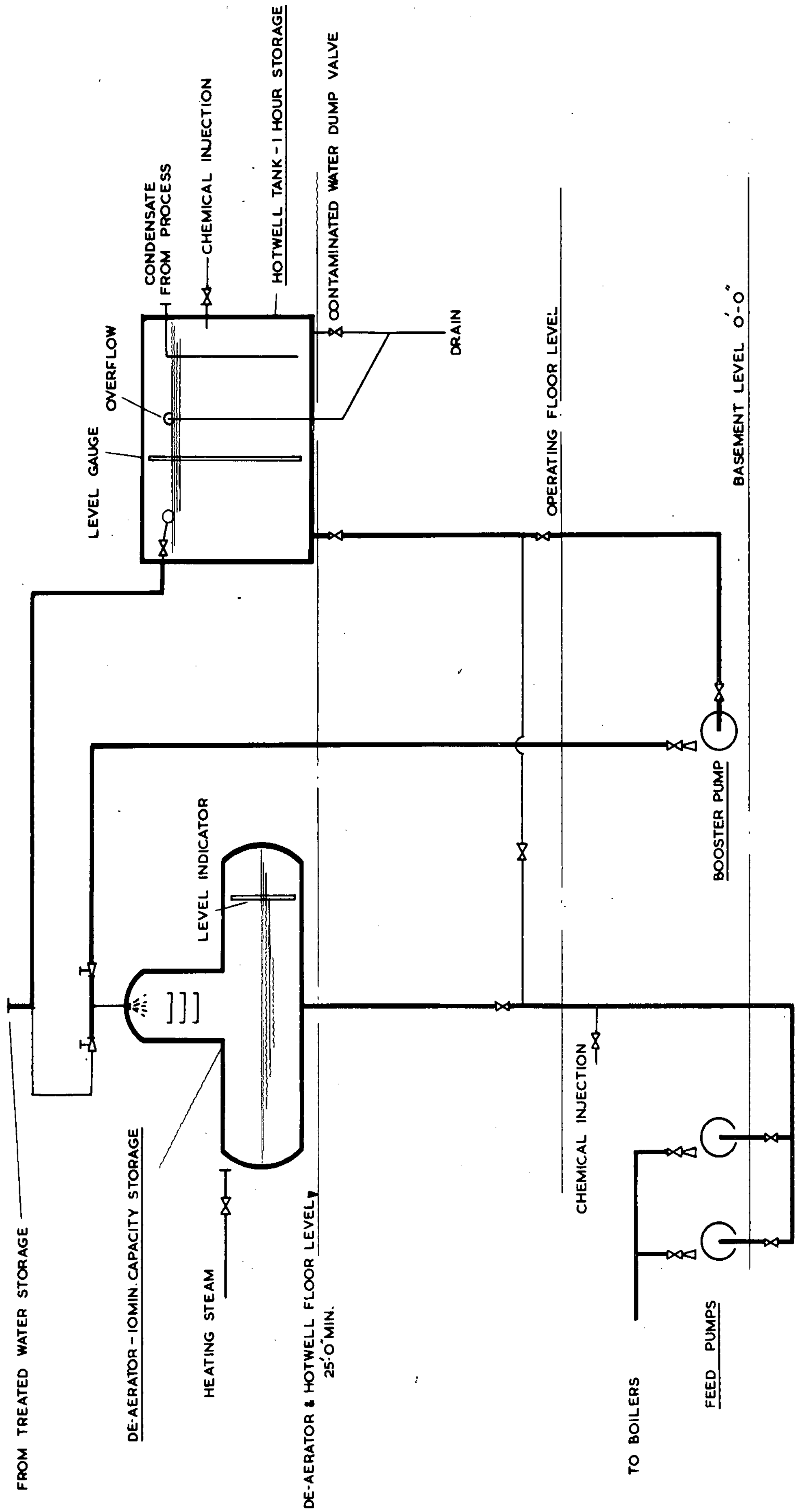


FIGURE 9.5: Pre-Boiler Feed System.

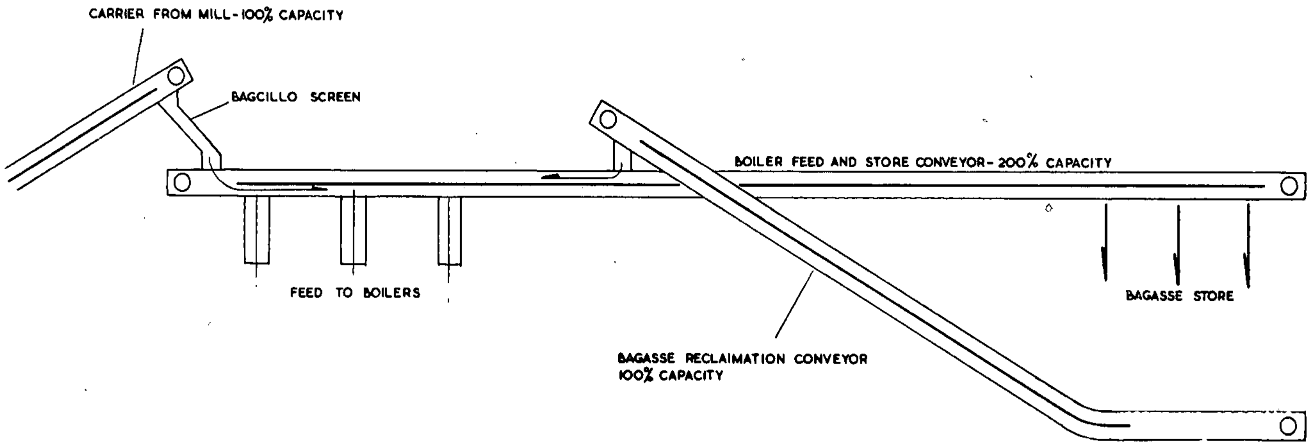


FIGURE 10.1: Arrangement of Bagasse Handling Equipment to ensure continuous supply to Boiler Plant.

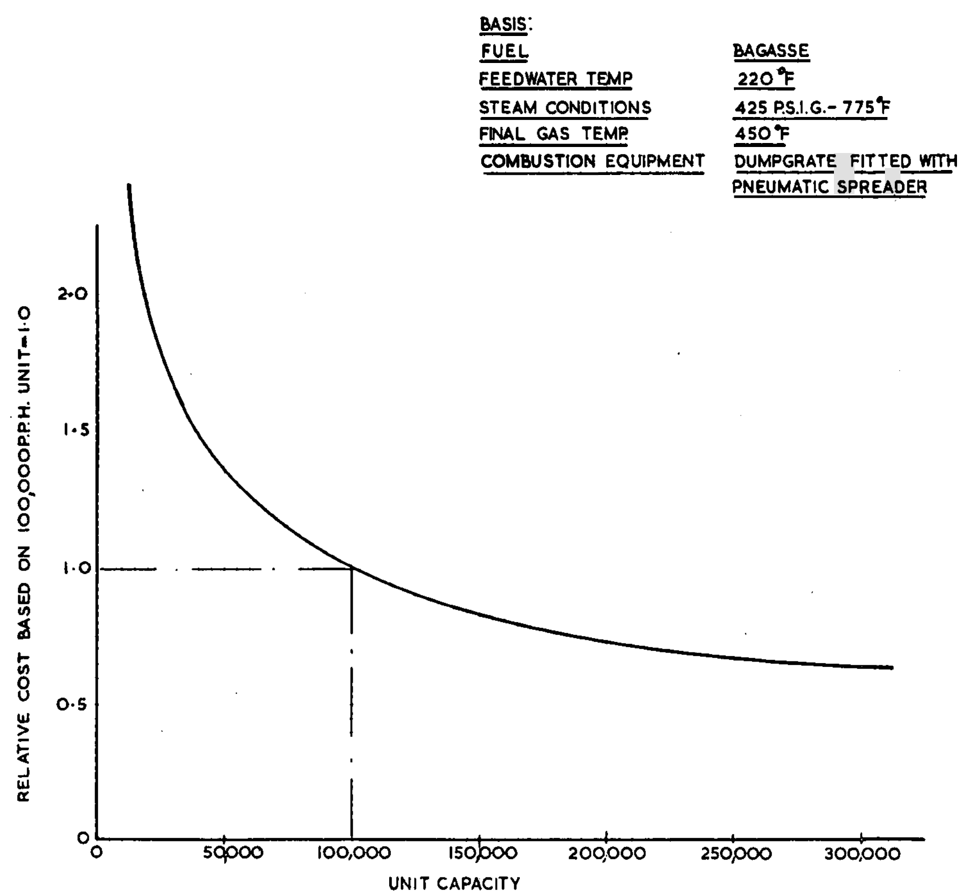


FIGURE 10.2: Curve showing Relative Cost per pound of Steam for Various Sizes of Boilers.

Mr. Ashe: It is indicated in figure 10.2 that the cost per pound of steam is much higher for smaller boilers than for larger boilers but this has not been our experience with a 50,000 lb. per hour boiler.

Mr. Magasiner: With small boilers it is possible to simplify the design and thus effect economies.

Mr. Hurter: How does the availability of boiler plant compare to the availability of turbo-alternators and turbines?

Mr. Magasiner: The availability of boiler plant is improving rapidly. Provided the boiler water treatment is adequate and attention is paid to the plant while it is operating there is no reason why a boiler should not steam throughout the crop without a loss of production. It should be possible to operate on one boiler only.

Mr. Hurter: It is not feasible to operate on one boiler if both bagasse and coal are being burned.

Mr. Magasiner: If it is running either on coal or on bagasse there is no reason why a factory should not operate on one boiler.