ENERGY CONSERVATION IN THE STEEL INDUSTRY

- BY -

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SUMMARY

The last few years have witnessed an unprecedented increase in the price of energy available to industry in the United Kingdom and worldwide.

The steel industry, as a major consumer of energy delivered in U.K. (8% of national total and nearly 25% of industrial total) and whose energy costs currently form some 28% of the total manufacturing cost, is very much aware of the need to conserve energy.

Because of the complexities of steelmaking processes it is imperative that a full understanding of each process and its interlinking role in an integrated steelworks is understood. An analysis of energy distribution shows that as much as 70% of heat input is dissipated to the environment in a variety of forms. Of these, waste gases offer the best potential for energy conservation.

The study identifies areas for and discusses novel methods of energy conservation in each process. Application of these schemes in BSC works is developed and their economic incentives highlighted.

A major part of this thesis describes design, development and testing of a novel ceramic rotary regenerator for heat recovery from high temperature waste gases, where no such system is available. The regenerator is a compact, efficient heat exchanger. Application of such a system to a reheating furnace provides a fuel saving of up to 40%. A mathematical model developed is verified on the pilot plant. The results obtained confirm the success of the concept and material selection and outlines the work needed to develop an industrial unit.

Last, but not least, the key position of an energy manager in an energy conservation programme is identified and a new Energy Management Model for the BSC is developed.

KEY WORDS: ENERGY, STEEL INDUSTRY, WASTE HEAT RECOVERY, REGENERATORS, ENERGY MANAGEMENT.

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NOMENCLATURE

SYMBOL	DESCRIPTION	UNITS
A	Cross-sectional area	m^2
A_a	Air port area i.e. area subjected to air flow	m^2
Ca	Solid/air capacity ratio	dimensionless
Cg	Specific heat of gas	kCal/kg^{o}C
Cp	Specific heat	J/Nm^3 ^{o}K
Cpa	Specific heat of air	kCal/Kg^{o}C
Cs	Specific heat of material	kCal/kg^{o}C
cv	Calorific value	kCal/kg
D_S	Separation diameter i.e. position at which particles separate from gas	m
d	Hydraulic diameter	m
dq	Quantity of heat received by air from bed	kCal
E	Modulus of elasticity	kg/m^2
F	Frictional force	newton
G_m	Gas mass flowrate	kg/m^2/h
G	Grindability	dimensionless
G_s	Grain size	g/cm^3
g	Gravitational acceleration	m/sec^2
H	Hardness number	dimensionless
H_f	Sensible heat in flue gases	kCal
H_r	Heat loss by radiation	kCal
H_s	Heat to furnace stock	kCal
H_w	Heat loss to waste gases	kCal
ha	Sensible heat of gas	kCal/kg
hg	Sensible heat in waste gas (constant)	kCal/kg
K	Thermal conductivity	kCal/mS^{o}C
Kr	Coefficient of air resistant	dimensionless

NOMENCLATURE

(Continued)

SYMBOL	DESCRIPTION	UNITS
K	Numerical constant	dimensionless
L	Leakage	kg/s
L_A	Load/unit area	g/cm^2
Lg	Sensible heat of gas	kCal/kg
M	Normal area in contact/area available	dimensionless
m	Loss of mass of block	mg
n	Rotor speed	r.p.m.
p_E	Pressure after the turbine	bar abs.
p_G	Pressure in front of furbine	bar abs.
pai	Air inlet pressure	N/m^2
pgi	Waste gas inlet pressure	N/m^2
p1	Inlet pressure to bed	bar abs.
p2	Outlet pressure to bed	bat abs.
Qf	Quantity of flue gas generated by the combustion of 1 kg of fuel	kg
Kv	Volumetric coeff. of heat exchange	$kCal/m^3h^oC$
S	Surface area along a unit path	cm^2
S_1	Specific surface area	m^3
T	Temperature	^oK
T_E	Gas temperature after turbine	^oK
T_G	Gas temperature in front of turbine	^oK
Ta	Combustion air temperature	^oK
Tg	Temperature of waste gases leaving the furnace	^oK
Tm	Ambient air temperature	^oK
Tx	Matrix temperature	^oC
t	Wall thickness between two adjacent holes	m
ta	Air temperature at distance	^oC
tai	Air inlet temperature	^oC

NOMENCLATURE

(Continued)

SYMBOL	DESCRIPTION	UNITS
ta'	Air temperature entering the bed	^{o}C
ta"	Air temperature entering combustion zone	^{o}C
tgi	Waste gas inlet temperature	^{o}C
ts	Average integral temperature in cross-section	^{o}C
ts'	Temperature of bed material	^{o}C
U	Refractory utilisation	w/kg
Ug	Gas velocity	m/sec
Uo	Air velocity at the bed inlet	m/sec
Vt	Tangential component of gas velocity	m/sec
Vv	Radial component of gas velocity	m/sec
V_G	Volume of expanded gases	Nm^3/sec
W_G	Power generated	KW
Wa	Water equivalent of air	dimensionless
Wf	Fuel consumed per unit time	kg
Ws	Water equivalent of gas	dimensionless
w	Speed of combustion zone relative to top of bed	m/sec
X	Functional coefficient	dimensionless
Y	Functional coefficient	diemsnionless
β	Heat loss in cross-section	dimensionless
e	Bed voidage	dimensionless
f	Bed porosity through cross-section (assumed 0.2-0.2)	per cent
ℓ	Length of passage i.e. depth of rotor	m
ℓ_B	Length travelled by block	cm
η	Air/fuel ratio at stiochimetric	dimensionless

NOMENCLATURE
(Continued)

SYMBOL	DESCRIPTION	UNITS
α	Co-efficient of linear expansion	$^{o}C^{-1}$
∞	Ratio of air port area to waste gas port area	dimensionless
ρ_B	Density of block	g/cm^3
ρ_L	Density of liquid	kg/m^3
ρ_g	Density of gas at prevailing condition	kg/m^3
Q_L	Liquid mass flowrate	$kg/m^2/h$
μ_L	Liquid viscosity	centi-poise
σ_{max}	Tensile fracture stress	kg/m^2
θ	Angle of contact	degree
γ_s	Bulk density of material	kg/m^3
Δpa	Pressure drop through the air side	N/m^2
Δpg	Pressure drop through the waste gas	N/m^2
Δx	Distance from top of bed to cross-section considered	m
η_G	Generator efficiency	per cent
η_T	Turbine efficiency	per cent
η_A	Air side effectiveness	per cent
η_G	Gas side effectiveness	per cent
λ_A	Air flux density	kg/sm^2
λ_G	Gas flux density	kg/sm^2
x	Excess air	per cent

ENERGY CONSERVATION IN STEELMAKING

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

The two major sociotechnological concerns that have come to great prominence in the last decade are the energy crisis and environmental crisis both of which are manifestations of the fundamental laws of thermodynamics:

i.e., (a) Energy can be neither created nor destroyed,

and (b) Entropy of a closed system composed of all sources and sinks tends to increase.

It is recognised that if there is not enough energy everything will come to a halt and thus the energy crisis has now become a first order crisis.

In industry energy should be treated just as any other human or material resource and the fact that it cannot be seen is no reason for management to overlook, as frequently happens, its responsibilities to control and adequately supervise its usage.[1-6] All too often the responsibility for energy costs is that of the Chief Engineer, who usually acquires it efficiently only to see it used less so.[7-8]

It is impossible to exaggerate the importance of energy to modern society. It fuels our transport and heats our homes. It is the very basis of the industry on which most jobs in any community directly or indirectly depend.

1.1.1 Historical Background

The progress of the industrial nations of the world in the last century has been characterised by the tecnhological developments based on the constantly increasing utilisation of energy sources. The early industrial age (until the end of the 19th century) was based on coal alone. It was then increasingly replaced by petroleum, which is more versatile and easier to use. In the middle of the 20th century, during a phase of expontential growth, an additional energy source, nuclear fuel was added to these two sources.[(9)]

Until to-day it was recognised that energy consumption can be directly related to the living standards of the populace and to the degree of industrialisation of the country, Fig.1.1[(10)] However this belief is now changing.[(11)]

In the sixties despite warnings from conservationists of a possible 'energy gap' towards the end of this century, little attention was paid to savings.[(12-23)] This was because at the time, with the coming of natural gas along with future prospects of North Sea oil and discovery of nuclear fuel, beliefs ran high that a limitless supply of cheap energy would always be available. In the past this has lead to inefficient utilisation of energy in some cases with disasterous ecological effects. However with the sharp escalating price of energy since the triggering of the price war by the first united action of OPEC Members on crude oil pricing in 1973, energy dependent countries worldwide have recognised the need to conserve it.[24] In addition recent international events have brought many nations to acknowledge that the national security, financial stability and living standards of a nation are intertwined with the energy conservation of the country.

Therefore energy conservation is a MUST not only for this generation but for all the generations to come whilst life lasts on this planet.

This thesis looks at the need and investigates ways to conserve energy in the biggest single industrial consumer of energy in the country. i.e. the steel industry.

Before discussing the effects of the energy crisis on the British Steel Corporation and evaluating various energy conservation schemes, a brief look at the world energy situation and the UK Energy Policy, together with energy consumption in various processes in steelmaking, is presented.

1.2 WORLD ENERGY SITUATION

The World energy consumption has been rising steadily for the last quarter of a century at the rate of five percent per annum (Fig.1.2)[25]. This is equivalent to the doubling of annual consumption every fourteen years.

As more and more of the world's population moves from agricultural pursuits to industrialisation, the demand will continue to expand at an alarming rate. This rapid increase in demand for energy has caused great concern for the adequacy of the long range supply of fossil fuel for the world.[25-28]

Total world conventional energy resources are presented in Tables 1.1, 1.2 and 1.3.[29]

Table 1.4 shows that, despite large reserves of coal when compared with other fossil fuels, because of the intrinsic advantages of oil and gas, there has been a consistent and growing trend away from the use of coal in the past. A persistance in this poilcy would mean an early exhaustion of gas and oil.

To provide the increasing demand for energy both by industrial and developing nations, the state of available energy compels us to look at alternative energy sources and conserve energy wherever possible.[30]

The renewable energy sources that are currently studied are as follows:

- Geothermal Energy
- Solar Energy
- Wind Energy
- Wave, Tidal & Ocean Thermal & Flow Energy
- Biomass
- Hydraulic Energy

It must be recognised that a time span is needed before any of these alternative energy sources can be successfully and ecomonically exploited.

Therefore energy conservation is essential if present resources are not to run out more quickly than new sources can be found or new technologies introduced.

As a part of the world energy scene, the European Community is already facing an increasingly grave situation of not producing enough energy for its needs. At present oil provides 55 percent of the member states energy requirements of which 85 percent comes from abroad, mainly the Middle East.[31] Clearly there are grave dangers in such a heavy reliance on imported oil. Political factors or transport problems could at any time interrupt supply. Furthermore, the control of imported energy is outside the control of member governments and constitutes a permanent drain on the EEC's balance of payments. Therefore, as a first step, the Community has agreed to reduce imports of oil and substitute other sources of energy wherever possible.[32] In addition vigorous energy saving measures are being encouraged.[33-35]

1.3 UK ENERGY POLICY

The primary objectives of any energy policy are

1. The development of an adequate supply of energy at reasonable prices to permit the nation to enjoy a good standard of living.

2. The achievement of relative self sufficiency of energy supply.

3. The maintenance of a safe and healthy environment.

4. The attainment of maximum efficiency in the production, distribution and utilisation of all forms of energy.

These objectives cannot be pursued in isolation. Every energy policy decision is likely to involve some or all of a wide range of other considerations (Fig.1.3). Therefore the UK Energy Policy Green Paper tries to balance conflicting considerations in a wide variety of circumstances while still preserving the structure of a reasonably coherent and consistent energy policy.[36]

1.3.1 UK Energy Scene

During the 20 years up to 1972, the coal industry lost many of its traditional markets as a result of technological developments and competition from cheap oil. Annual production dropped from 220Mt to 140Mt and the industry became mainly dependent on sales to the Electricity Board which in 1976 took over 60% of coal production of 122Mt.[37]

At the same time because of the availability of large quantities of oil at low costs a rapid increase in oil demand occurred. With the discovery of natural gas in the late sixties the demand for gas as a clean and easy to handle fuel has been increasing (Fig.1.4).[38,39]

However, following the first united action of the OPEC members on crude oil pricing in 1973 and subsequent unprecedented price rises it was recognised that measures for energy conservation can make a significant contribution to the balance of payments by reducing dependence on imported oil.[40] In addition, it is recognised that, at the present rate of consumption the availability of fossil fuels will be exhausted by the end of this century. Therefore a positive conservation programme coupled with the development of alternative energy sources is essential.

1.3.2 Energy Conservation

Effective utilisation of energy involves the study of the following interrelated sets of problems:[41-43]

1. The optimal choice of primary energy sources and assessment of their rates of depletion so as to conserve those resources whose scarcity would have adverse social and ecomonic effects.

2. The avoidance of unnecessary waste of energy.

3. Technological changes leading to improvement in the efficieny with which energy is used.

4. Changes towards alternative products or patterns of demand that reduce the rate of growth of energy use.

The energy industries occupy an important and central position in the UK economy both nationally and regionally. They directly employ 650,000 people (approx. 3 percent of the working population) and support 200,000 jobs in process plants (including offshore supplies), construction and other industries. The energy industries contribute about 4½ percent of the gross domestic product at factor cost and in 1976 sold £23.5 billion of energy to its customers raising £2 billion in taxes for the government. The government, recoginsing the importance of not only the need to sustain stable energy industries, published the Govt. policy in the White Paper of July, 1976.[44]

It sees that because of the large oil and gas finds in the North Sea, Britain will be in a strong position and will become a net exporter during the medium term range (Fig.1.5). However it recognises that these finds will probably be exhausted by the end of this century and thus feels that coal must form the major source of energy supplier (coal known reserves 190 billion tonnes offering 300 years life at current extraction) together with nuclear power as a back up fuel. In addition every effort should be made to conserve energy wherever possible.[45]

The heat supplied to final user for 1979 is shown in Fig.1.6 representing both the type and the amount of fuel consumed by industry and for domestic purposes.[39] A corresponding figure (Fig.1.7) shows the movement of fuel prices over the same period.

1.4 ENERGY SCENE IN STEEL INDUSTRY

Fig. 1.6 shows that as much as 8 per cent of the national total and nearly 25% of the industrial total of energy delivered in the U.K. is consumed by the iron and steel industry making it the biggest single industrial consumer (other than energy conversion industries) of energy. Any energy savings in steelmaking can thus show a substantial effect on the national and company economy.[46-49]

As an energy intensive industry the BSC has always been aware of the need to minimise energy consumption, long before the energy crisis.[50-52] A study of the specific energy consumption in U.K. steelmaking over the last decade, Figure 1.8, has shown a steady decrease.

The last few years have witnessed an unprecedented increase in the price of energy available to industry in the United Kingdom and Worldwide following the first united action of the OPEC Members on crude oil pricing in 1973. (Fig.1.8) The average delivered price to the BSC for heavy fuel oil rose three fold from 1973 to 1975 and has further doubled since then, whilst coking coal price rose from an average of £8.7 per tonne in 1973 to £46.8 per tonne in 1979/80. As a direct result of these increases the Corpotation's fuel bill was then running at around £730 million.

Expenditure as purchased fuels and on coal, coke and breeze represented approximately 28% of the BSC's total expenditure for the operating year 1983/84. Furthermore all the available studies and estimates on U.K. energy price trends predict that energy costs will rise at a faster rate than capital and labour costs in the 1980's which will serve to increase the importance of energy conservation in the overall profinability of steelmaking.

There are a number of approaches to the problem of reducing the specific energy consumption in steel manufacture such as:

- use of higher grade iron ore
- better burden preparation
- more sophisticated blast furnace practice
- investment in new plant of high thermal efficency
- increased use of scrap
- improved thermal insulation
- increased emphasis on energy recovery and recycling

Many of the more obvious energy savings options have already been adopted by the Corporation, and indeed exploited to their fullest extent in some cases. One area where there may still be opportunities is that of energy recovery and recycling.

A schematic representation of the distribution of energy in steel manufacture, Figure 1.9, shows that, on average, only 25 percent of the energy supplied is required in reducing iron ore to iron.[53] Strictly speaking the total energy usefully employed does not exceed 30 percent, i.e. energy to reduce iron ore, plus energy sold in by-products, although a figure of 40% is usually taken to allow for the energy consumed in the intermediate heating operations necessary in going from iron to the finished steel product. The remaining 60-70 percent is dissipated from the works to the environment in a variety of forms. Figure 1.9 indicates the magnitude of energy rejection in the various forms. In this distribution it is interesting to note that 29 percent of all the input energy is ultimately rejected in waste gases and cooling water, that 19 percent is rejected by radiation from hot solids, that 17 percent is lost as radiation from the surfaces of process plant, and a further 6 percent is lost in cooling slag.

Furthermore, enhanced energy recycling offers a means of increasing the cost effectiveness of new and existing plant, thereby maximising the return on assets, at a time when major investment into new process plant is not appropriate.

In order to understand the global picture of energy distribution and thus the energy recovery and recycling opportunities it is essential to understand

1. the manufacturing processes of iron and steel making
2. the energy distribution/consumption in each of the major processes.

1.4.1 Iron and Steel Manufacturing Processes

Figure 1.10 shows a simplified iron and steel manufacturing process flow diagram.[54] The main processes can be classified as follows:

A. Coke Ovens

At the start of the process, coal is carbonised to coke in the coke-oven where by-products such as tar, benzole and coke-oven gas are also formed. The coke thus made plays a triple role in the furnace, i.e. as a fuel which in burning raises the temperature in the furnace, as a reducing agent to reduce iron oxide to iron and as a physical support for the 'burden' which is at the same time porous enough to allow the hot gases to permeate to the top of the furnace. The energy utilisation in coke making is presented in Figure 1.11 where it can be seen that an energy flow of as much as 49,000 MJ per tonne of coke output occurs. Energy losses of 6418 MJ/t of coke occurs with the biggest single loss of 2250 MJ/t as the sensible heat loss in the hot coke.

B. Sinter Plant

The ore is usually charged to the blast furnace in the form of sinter which is basically an agglomeration of ore, coke breeze and limestone roasted together in the sinter plant to form a clinker which not only has good physical characteristics to avoid degradation in the furnace but has lost a proportion of unwanted volatile matter in the roasting process. Most modern blast furnaces rely primarily on sinter and coke as their burden whilst some have a variety of other burden materials. Alternatively, some ores are prepared as pellets by bonding very finely ground ore. But this is generally done near the source of the ore.

Fig.1.12 shows that the main sources of energy loss are sensible heat loss in sinter cooling (30%), waste gas losses (28%) and the radiation in ignition hood and other losses (42%) accounting for a total loss of 2287 MJ/t sinter. Energy recovery from sinter cooling can offer substantial savings.

C. Blast Furnace

A modern blast furnace is a steel cylinder which is lined with refractory bricks and is topped by a complex system for the charging of raw materials, and outlet pipes for the gas produced in the process At the base of the cylinder, above the hearth, are a series of tuyeres (nozzles) through which pre-heated air is blown.

The air is pre-heated in hot blast stoves, and the oxygen in this air reacts with hot carbon (coke) to form carbon monoxide, a reducing gas, which reacts with iron oxide to release iron. This leaves the iron free to melt and drop to the hearth of the furnace, producing a heavy lower layer of liquid iron. The gas is drawn off at the top of the furnace, cleaned and then used elsewhere in the works as fuel (including the hot blast stoves). At the same time the limestone reacts with the other impurities to form a liquid slag. This also falls to the hearth, but being lighter than the iron floats on the surface.

As liquid iron and slag build up in the well of the furnace, the slag and the molten iron are tapped off through the holes at the base of the furnace - the slag to a nearby tip and the iron sometimes to a pig-casting plant, but, more often today, straight to the steel furnaces in giant ladles whilst still molten. The whole process is continuous. A schematic diagram is shown in Fig.1.13.

The Sankey diagram (Fig.1.14) shows that the total energy flow through the whole system is seen to be 24,502 MJ/t hot metal. The major areas of energy losses are in gas cleaning, cooling water, gas bleed and blowers.

Major savings in this area can only be achieved through improved practice and above all reduction of coke rate.

However, modern blast furnaces are being designed to operate at higher pressures to get better yields from the furnace. A recovery of potential energy from these top gases can offer substantial energy savings.

D. Steel-making

There are three basic methods of steel-making:

i) BOS Process

ii) Electric Arc Furnace

iii) Open Hearth Furnace

Open hearth furnaces have been on the decline for some time, and currently none of the BSC's steel is produced through this route. More than 60% of the steelmaking is currently produced through the BOS route (Fig.1.15) and the rest through Electric-Arc furnaces. Therefore, it is decided to leave consideration of the open-hearth system out of this thesis.

i) Basic Oxygen Process - BOS:

The Basic Oxygen process is really an enlarged and refined development of the old Basic Bessemer process (now largely obsolete). The major difference is that the new vessel has a solid base and the oxygen (instead of air) is introduced through the neck of the vessel. The BOS vessel also has a much greater capacity and can use a higher proportion of scrap.

Over 350 tonnes of molten metal, containing as much as 30 percent scrap, can be refined in one charge in about 40 minutes with a high control over quality.

The process in the main is exothermic and thus requires very little energy input. Fig. 1.16 represents a Sankey diagram showing the average input and output of energy per tonne of BOS steel made in U.K. in 1978 assuming a hot metal feed of 93% Fe carrying with it the transformation energy of 7.4 MJ per Kg of Fe plus the sensible heat and chemical heat (oxidation of metalloids) per tonne of hot metal.

The figure shows that recovery of high calorific value BOS gas containing some 70% CO can offer major energy savings (475 MJ/t).

ii) Electric Arc Furnace - (EAF)

Steel is made by charging the cold steel scrap in a furnace and melting it with the help of an electric arc. Small additions of other flux materials are added to produce the steel of the right composition. Graphite electrodes are slowly but steadily consumed as a result of the arcing process and, on average, the rate is approximately 5 Kg/t steel.

In assessing the overall energy requirements of the process it is necessary to take into account the energy sequestered in the graphite electrodes and power generation. However, energy input from chemical reactions is considerably less than in the BOS process. The total energy flowing through the EAF process to produce one tonne of liquid steel is about 11,790 MJ (Fig.1.17), including the energy sequestered in the materials input. This, when added with the associated energy rejected at the power station, becomes 16,573 MJ; compared with 12,160 MJ/t for the BOS process.

The main area of energy recovery in the system seems to be from the hot waste gases which can be used for pre-heating of the scrap.

E. Finishing

Primary - Ingot and Continuous Casting

The processing of steel downstream of liquid steel production often involves a variety of manufacturing techniques and for convenience these have been divided into two broad categories:

Primary finishing (primary rolling or continuous casting for the production of slabs and blooms, sometimes referred to as semi-finished steel).

Secondary finishing (secondary rolling and reheating for the production of such products as bar, rod, rail, wire and sheet).

Traditional Ingot Casting Route

In 1978, 16.7 million tonnes of steel was processed via the traditional ingot casting route. This involves the teeming of liquid steel into moulds where it cools and solidifies, thus producing ingots. The latter are then heated up uniformly by burning fuel in "soaking pits" after which they are rolled to the required geometry in what is termed a "primary mill". The overall yield of metal by this route is taken to be 86.8 percent for the purposes of this study.

The energy inputs and outputs associated with the processing of one tonne of liquid steel to slabs or blooms using the traditional casting route have been estimated and are shown in the Sankey flow diagram of Fig.1.18.

It is noted that during ingot casting, enthalpy losses of 700 MJ/tonne in the 1,400 MJ/tonne carried by the hot liquid steel occurs. This is beacuse liquid steel cools in the moulds until, as ingots, they are safe to handle. Some metal is "lost" from the system at this stage, but the assumption is made that this is all recycled as part of the "home scrap". to upstream processes. A considerable quantity of fuel, equivalent to to 1100 MJ/tonne of slab (2100 MJ/t for 100 percent cold charge) and consisting of liquid or gaseous fossil fuel, or even coke-oven or blast furnace gas is used in the soaking pits. Electric power and some steam are used by the primary mills. A further "loss" of metal (known as millscale) occurs during these operations, again which is assumed to be all recycled upstream. (If this is not the case, then the energy content of this metal would represent a loss of 830 MJ from the system).

Neglecting the energy associated with the "lost" metal, nearly half the total loss results from cooling down of the liquid steel. Inevitably, liquid steel must at some stage cool down to the solid state, at a handleable temperature, before marketing. However, in this case the large amount of heat involved does no useful work. Furthermore, the ingots have to be reheated for rolling purposes, hence the greater energy losses from this system compared with the continuous casting route of Fig.1.19.

Continuous Casting Route

An increasing proportion of liquid steel produced in the UK is being cast via the continuous casting route. In 1978, 3.15 million tonnes of steel was processed in this way and the quantity produced annually is expected to increase steadily for some years to come. The process essentially consists of feeding the liquid metal from the steelmaking furnace, via a holding container or tundish to a continuous casting unit (Concast Plant) from which the solidified steel is withdrawn through rolls and cooling sprays, the cast steel finally being cut into the required lengths of slab or bloom steel.

The energy losses associated with the casting of one tonne of steel by this route have been estimated and are shown in the Sankey diagram Fig.1.19. The yield of solid metal is higher than that obtained by the traditional ingot casting route and is taken to be 94.9 percent. A very small quantity of fuel and electric power is required to operate the plant. The energy lost from the system comes from the sensible and latent heat of the liquid steel. Water cooling and water sprays extract 860 MJ, standing in the slab yard results in the loss of a further 540 MJ by convection and conduction to the atmosphere. The "lost" metal accounts for 377 MJ of energy, but the assumption is made for the present purpose that this is in fact all recovered and recycled as "home scrap", to upstream steel-making processes. (In practice, some of this will actually be lost). The continuous casting process thus offers a significant reduction in energy consumption for forming and shaping semi-finished products.

Secondary - Reheating and Rolling

The energy analysis of this representative finishing operation is based on one tonne input of semi-finished steel and the results are presented in Fig.1.20. The various inputs and outputs of energy are shown as arising from one process rectangle representing the operations of reheating, rolling and cooling all combined together. Apart from the assumed 7325 MJ energy content of the metal charge of one tonne of cold semi-finished steel, there is a large input of fuel for reheating which may be oil, gas or by-product gas available on site. The energy content of this fuel is taken as 2,400 MJ, being a weighted average for the relative proportions of material going through the hot strip, plate, heavy section, light section, rod and bar mills of the industry in 1978. The value actually varies between 2,000 and 3,000 MJ. Similarly the electric power for the mills was found to average 280 MJ. The average yield is taken at 0.86 tonnes, covering all finished products. It is assumed that the 0.14 tonnes of metal lost is all recycled to upstream processes, although this may only be partially true. The energy losses from finishing are listed in ranking order below:

	MJ	Percentage
Waste gases from reheating furnace	760	28
Reheating furnace structure losses	610	22
Losses from cooling banks	560	20
Losses from mills, including power	490	17.5
Water cooling at reheating furnaces	350	12.5
Total	2770	100.0

There is no outstanding item of lost energy, although a total of 1,722 MJ (63 percent) of the losses result directly from the reheating operation.

Summarising, it can be said that major energy savings can be achieved by producing more steel through continuous casting. Otherwise, energy recovery from high temperature waste gases leaving the soaking pits and reheating furnaces offer potential areas of savings.

1.5 AREAS OF ENERGY CONSERVATION IN STEELMAKING

The main steelworks processes and the major sources of energy loss from the iron and steel industry have been identified.

For this thesis it was decided to look at energy conservation in an integrated steelworks via the BOS route. Fig.1.21. shows the summary Sankey Diagram for an integrated steelworks In arriving at these figures it is assumed that the plant is operating to a standard practice and the energy consumed in environmental control, conveying materials and losses incurred in the transfer of materials between plants are minimal.

Fig.1.22 demonstrates the relative magnitudes of the energy consumption for the various processes and Table 1.5 compares the way in which energy losses occur.

The areas of energy conservation in the steel industry that need exploring thus may be highlighted as follows:

1. Energy recovery from hot coke cooling - COKE OVENS
2. Energy recovery from sinter cooling - SINTER PLANT
3. Energy recovery from high top pressure gases leaving the blast furnaces - BLAST FURNACES
4. Energy recovery from BOS gas collection - BOS STEELMAKING
5. Energy recovery from waste gases - SOAKING PITS/ REHEATING FURNACES
6. Energy conservation through better process control and management.

The thesis now looks at these schemes in detail.

WORLD ENERGY RESERVES

Source: Survey of Energy Resources 1980

Ref. World Energy Conference, Munich 8-12th Sept.1980.

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Source: Survey of Energy Research 1980.

WORLD ENERGY RESERVES (SOLID FOSSIL FUELS) TABLE 1.2

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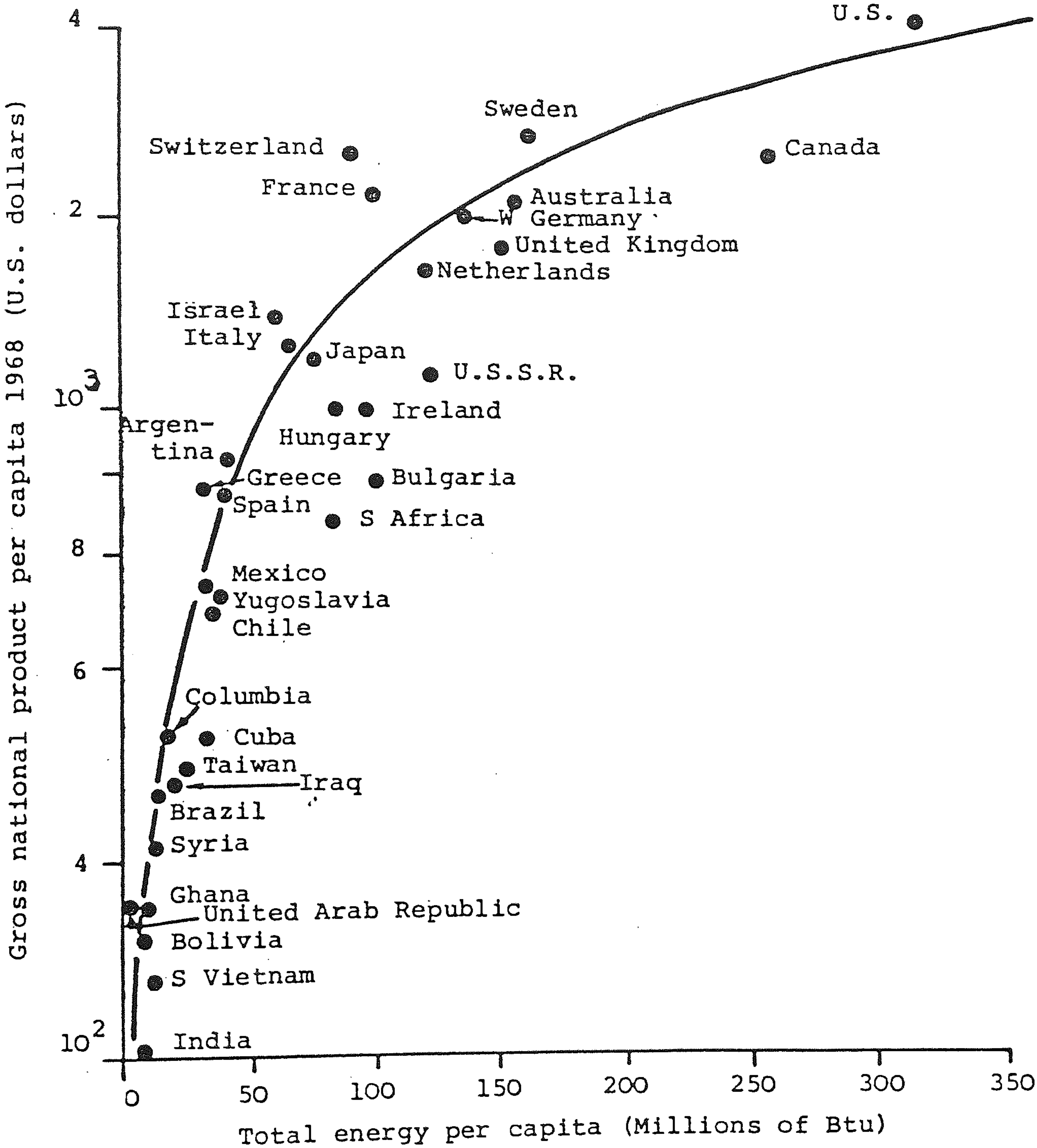
Source: Survey of Energy Resources, 1980.

WORLD ENERGY RESERVES (OIL & NATURAL GAS) TABLE 1.3

WORLD ENERGY CONSUMPTION OF FOSSIL FUELS

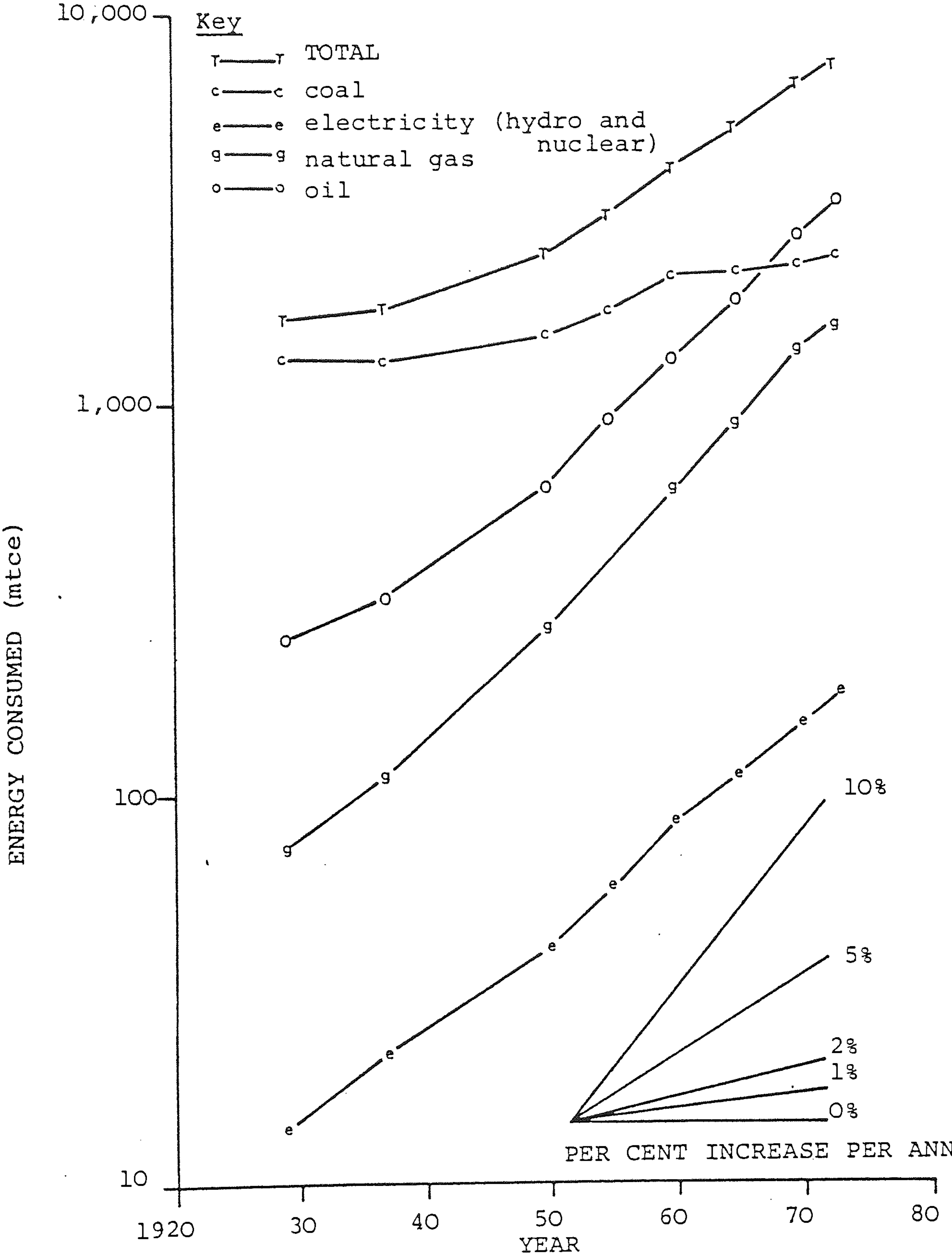
Source: UN Statistical Yearbook.
World Energy Statistics, 1981.

WORLD ENERGY CONSUMPRION OF FOSSIL FUELS   TABLE 1.4



ENERGY PER CAPITA VERSUS THE GROSS NATIONAL PRODUCT PER CAPITA IN 1968

FIGURE 1.1

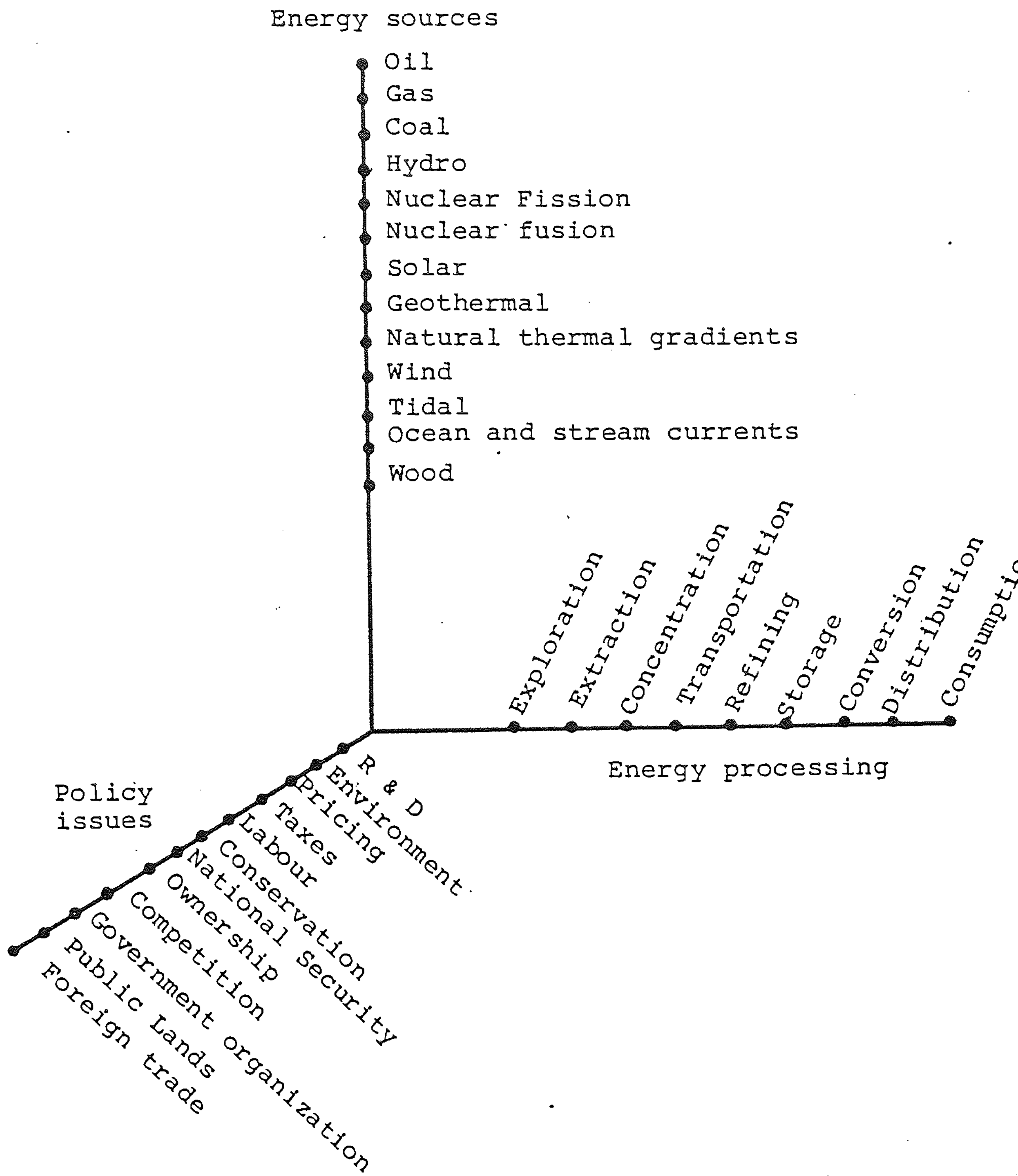


(UNITS: mtce = metric tonne coal equivalent)

Source: UN Statistical Yearbooks
Oil and Gas Journal
Institute of Petroleum Information Service

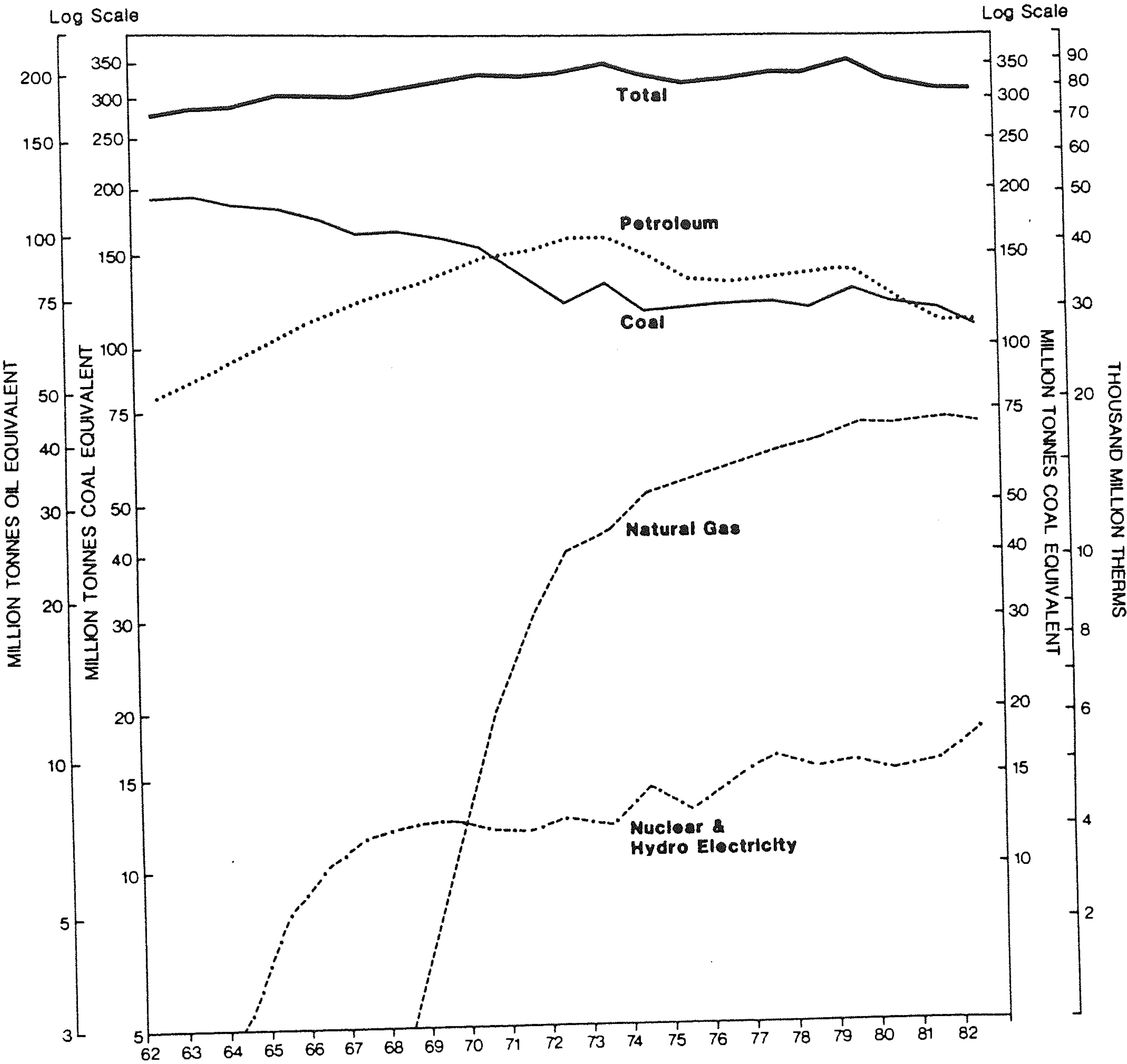
WORLD ENERGY CONSUMPTION

FIGURE 1.2



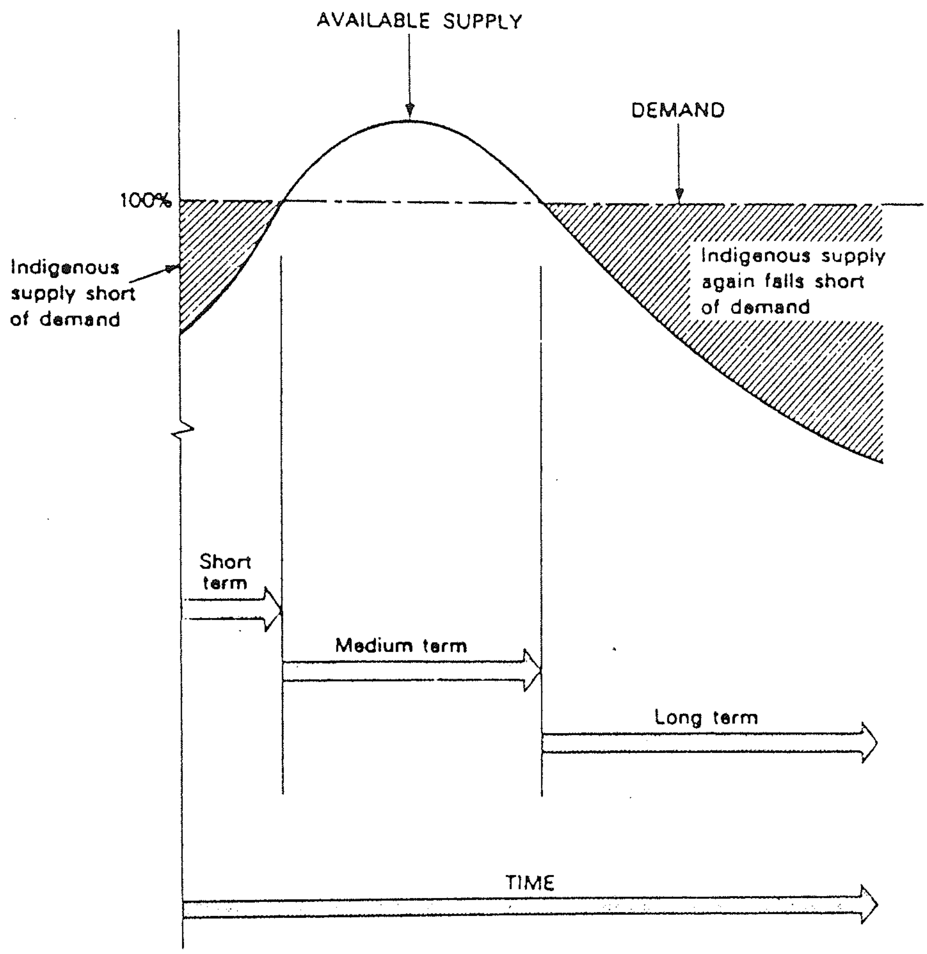
THREE DIMENSIONS OF THE ENERGY ISSUE

FIGURE 1.3



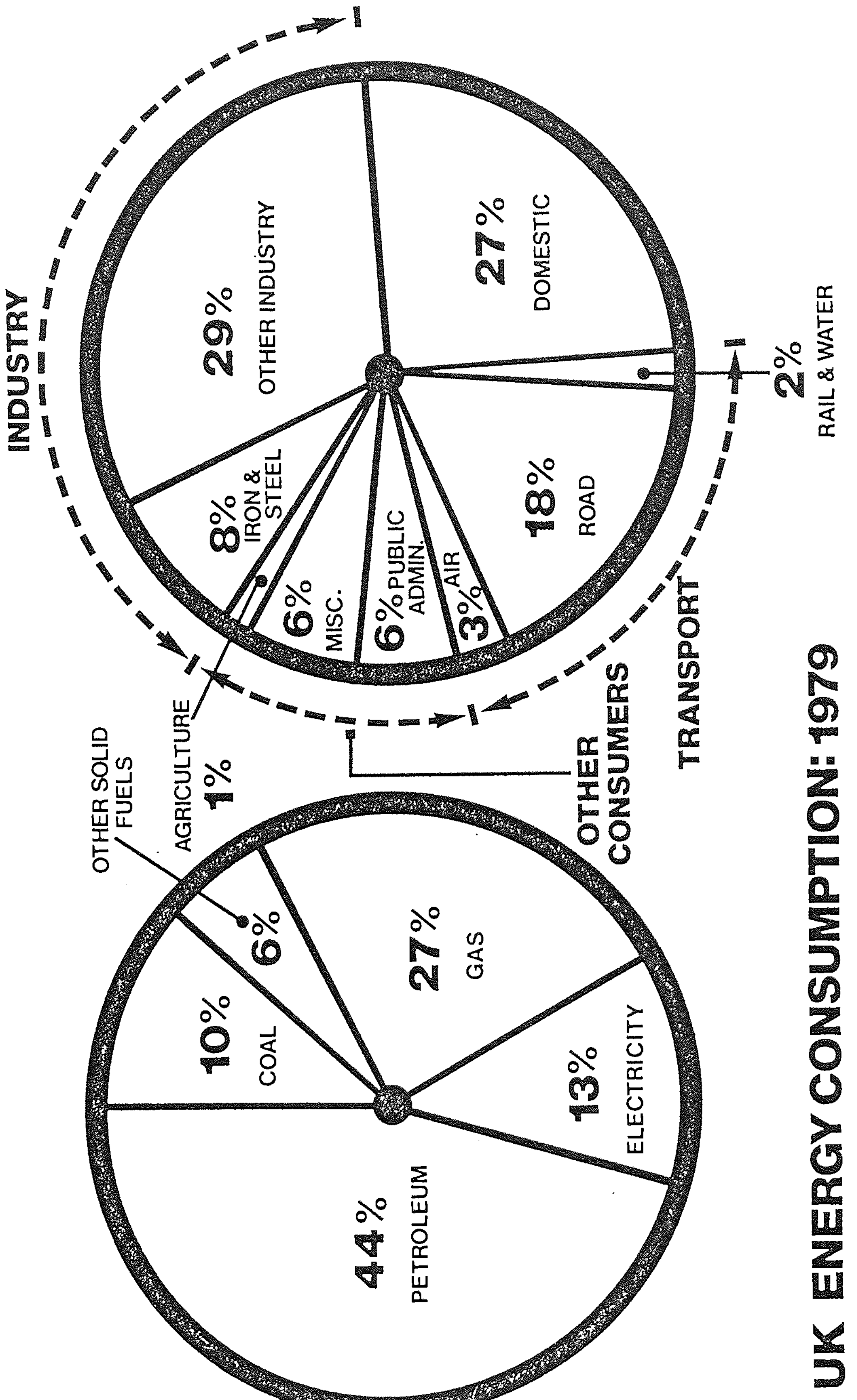
CONSUMPTION OF PRIMARY FUELS IN UK

FIGURE 1.4



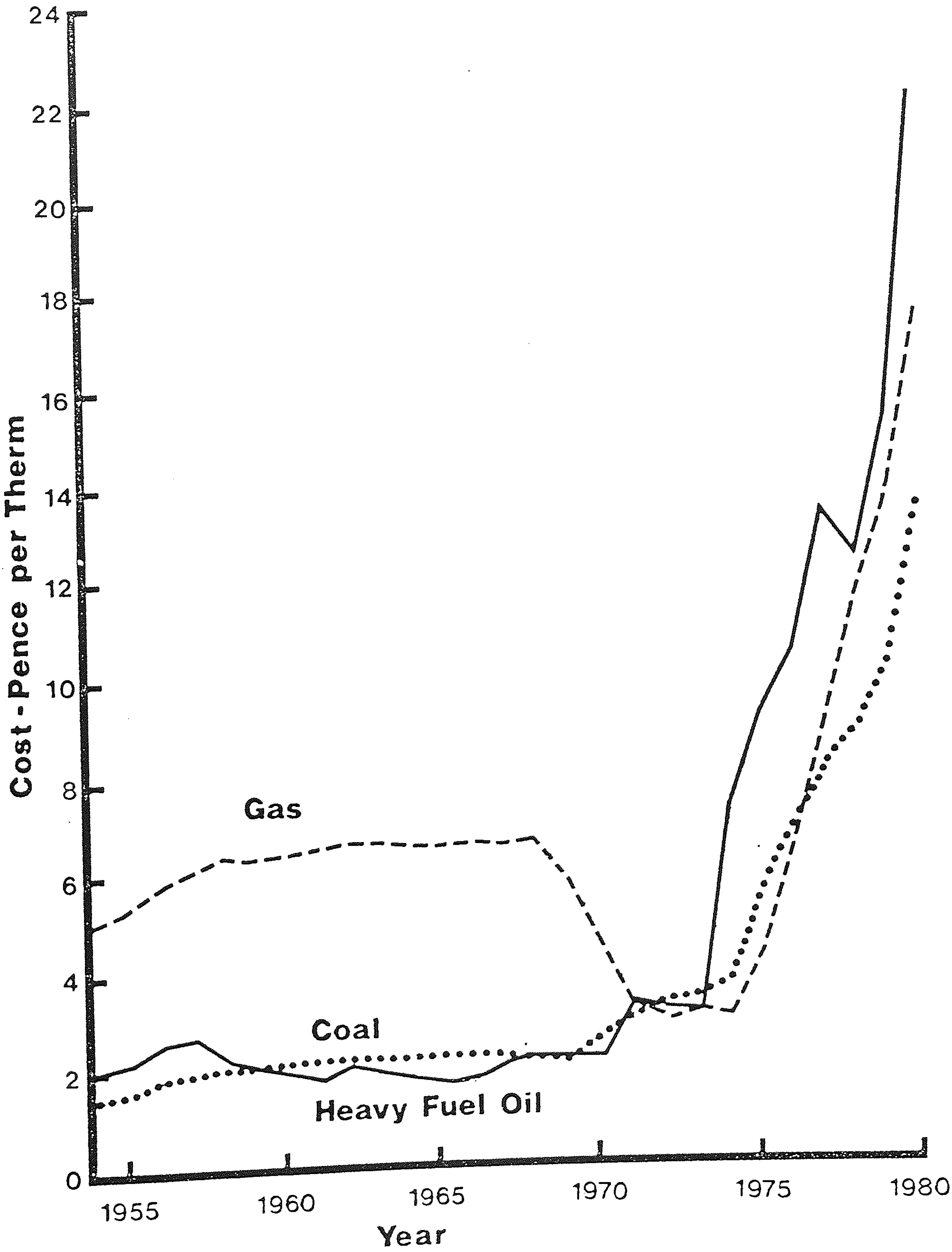
BRITAIN'S ENERGY FUTURE

FIGURE 1.5



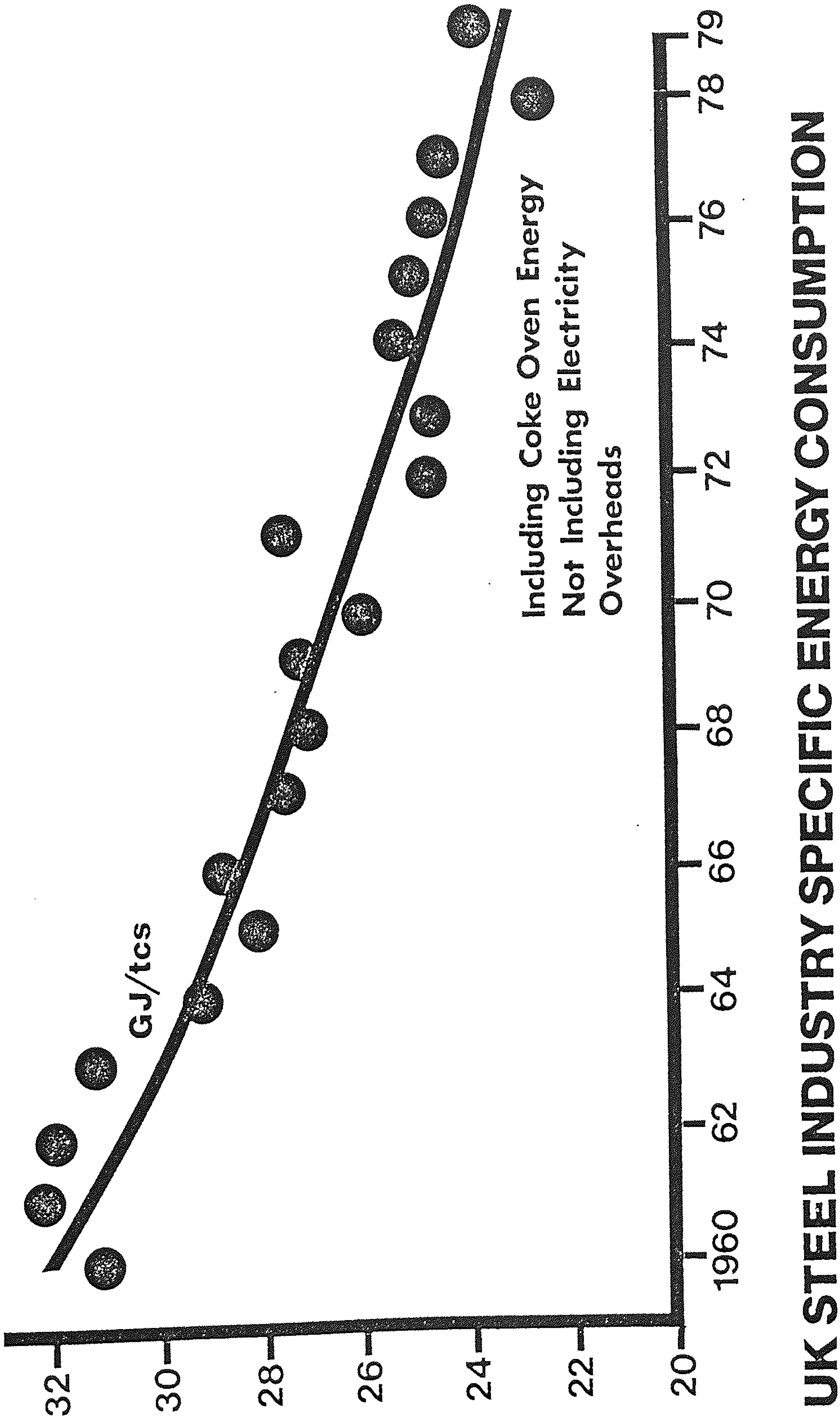
UK ENERGY CONSUMPTION : 1979

FIGURE 1.6



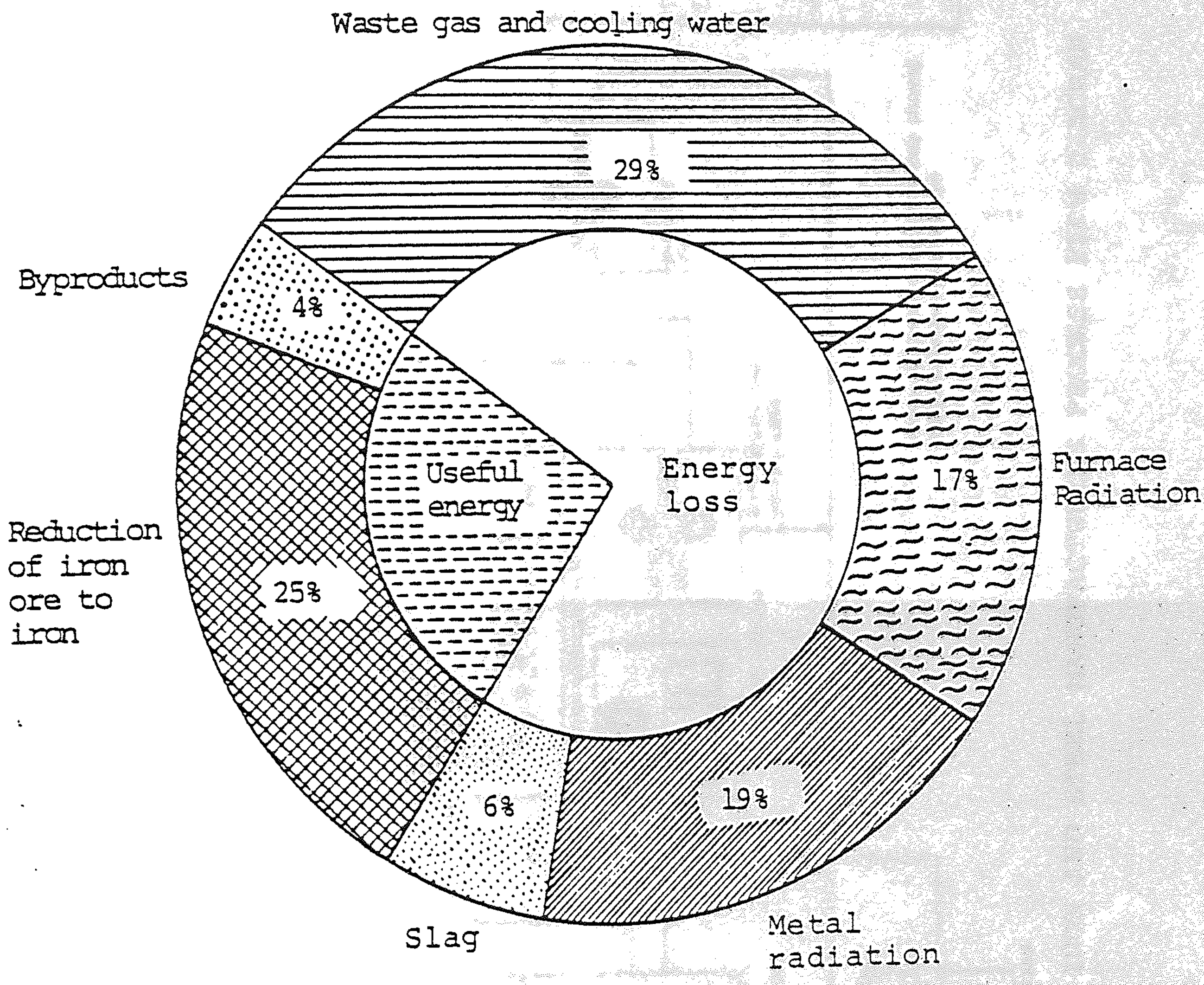
Prices of Fuels used by Industry in the United Kingdom

FIGURE 1.7



UK STEEL INDUSTRY SPECIFIC CONSUMPTION

FIGURE 1.8



ENERGY DISTRIBUTION IN STEELMAKING

FIGURE 1.9

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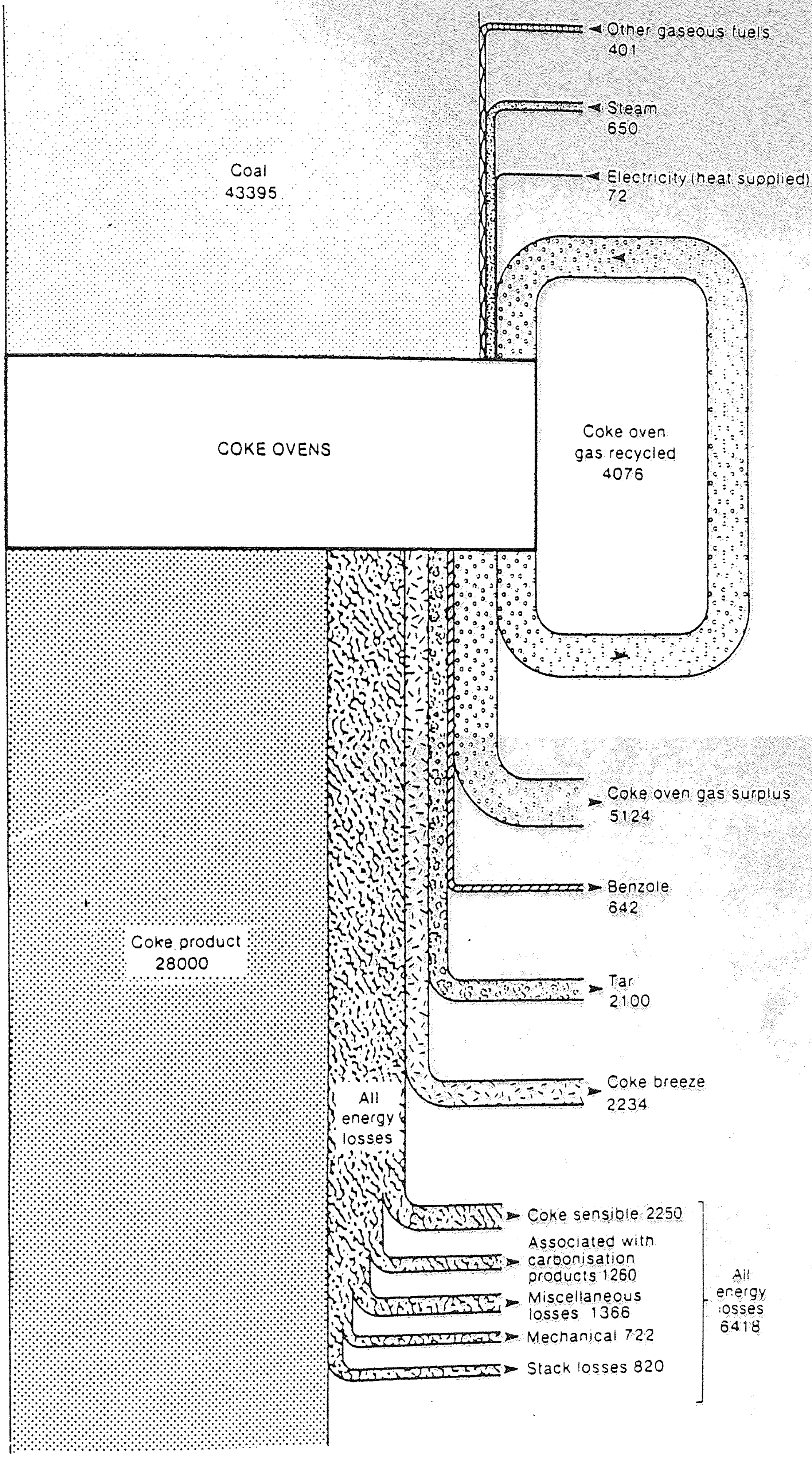
SIMPLIFIED IRON AND STEEL MANUFACTURING PROCESS FLOW DIAGRAM

SIMPLIFIED IRON & STEEL MANUFACTURING PROCESS FLOW DIAGRAM

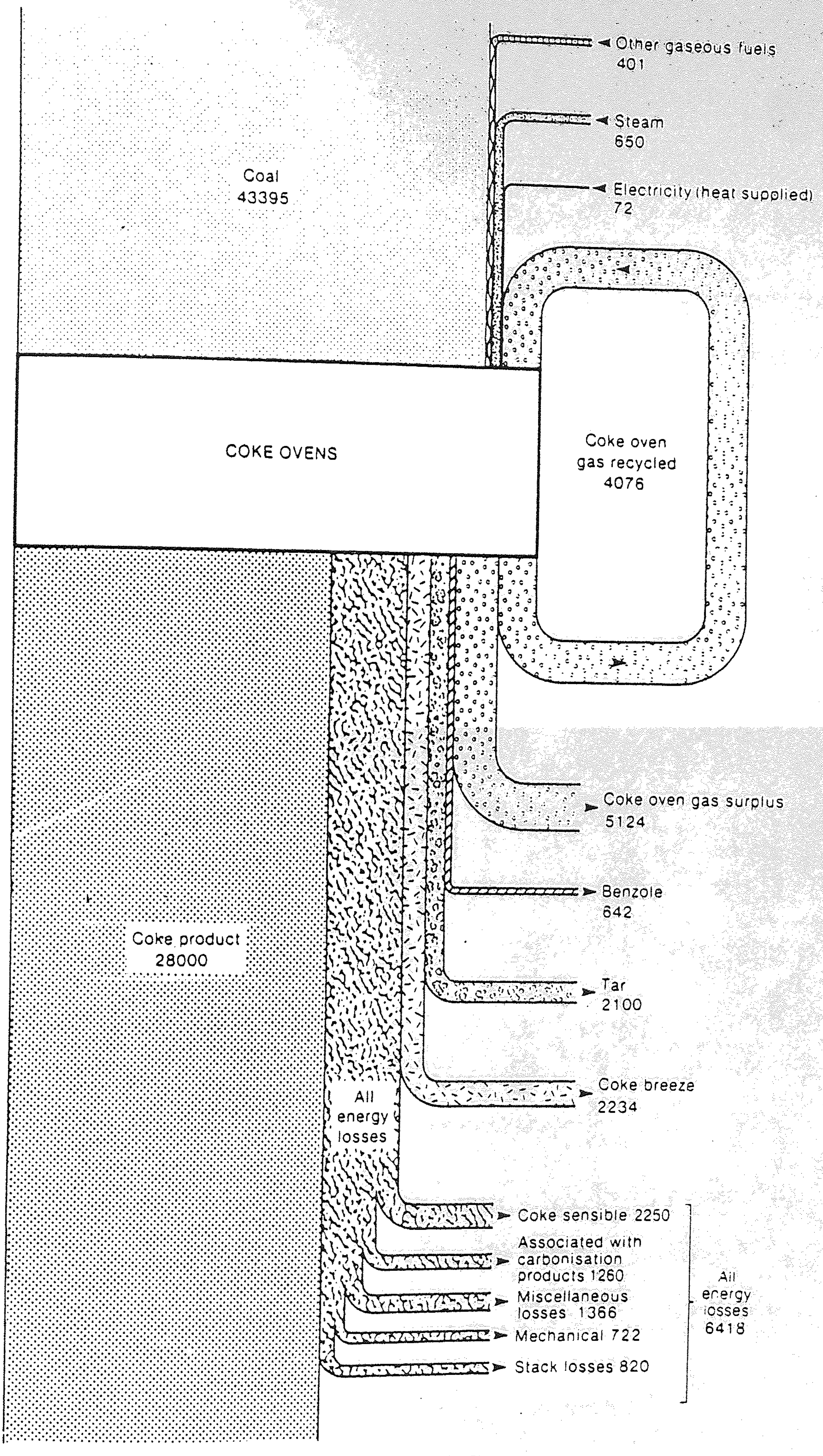
FIGURE 1.10



ENERGY UTILISATION IN COKE MAKING

MJ/tonne Crude Steel (1978)

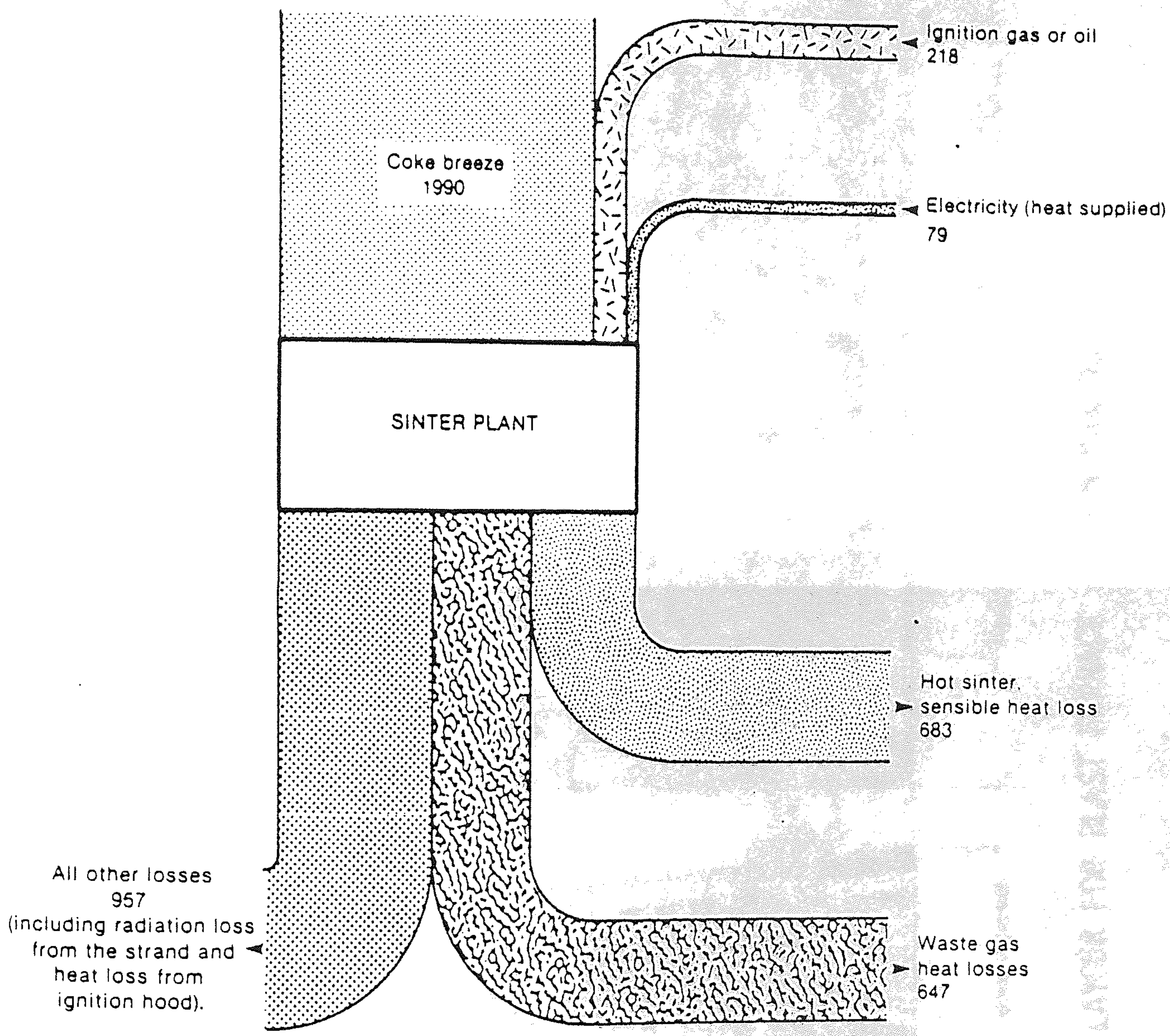
FIGURE 1.11



ENERGY UTILISATION IN COKE MAKING

MJ/tonne Crude Steel (1978)

FIGURE 1.11



ENERGY UTILISATION IN SINTER PRODUCTION

MJ/tonne Crude Steel (1978)

FIGURE 1.12

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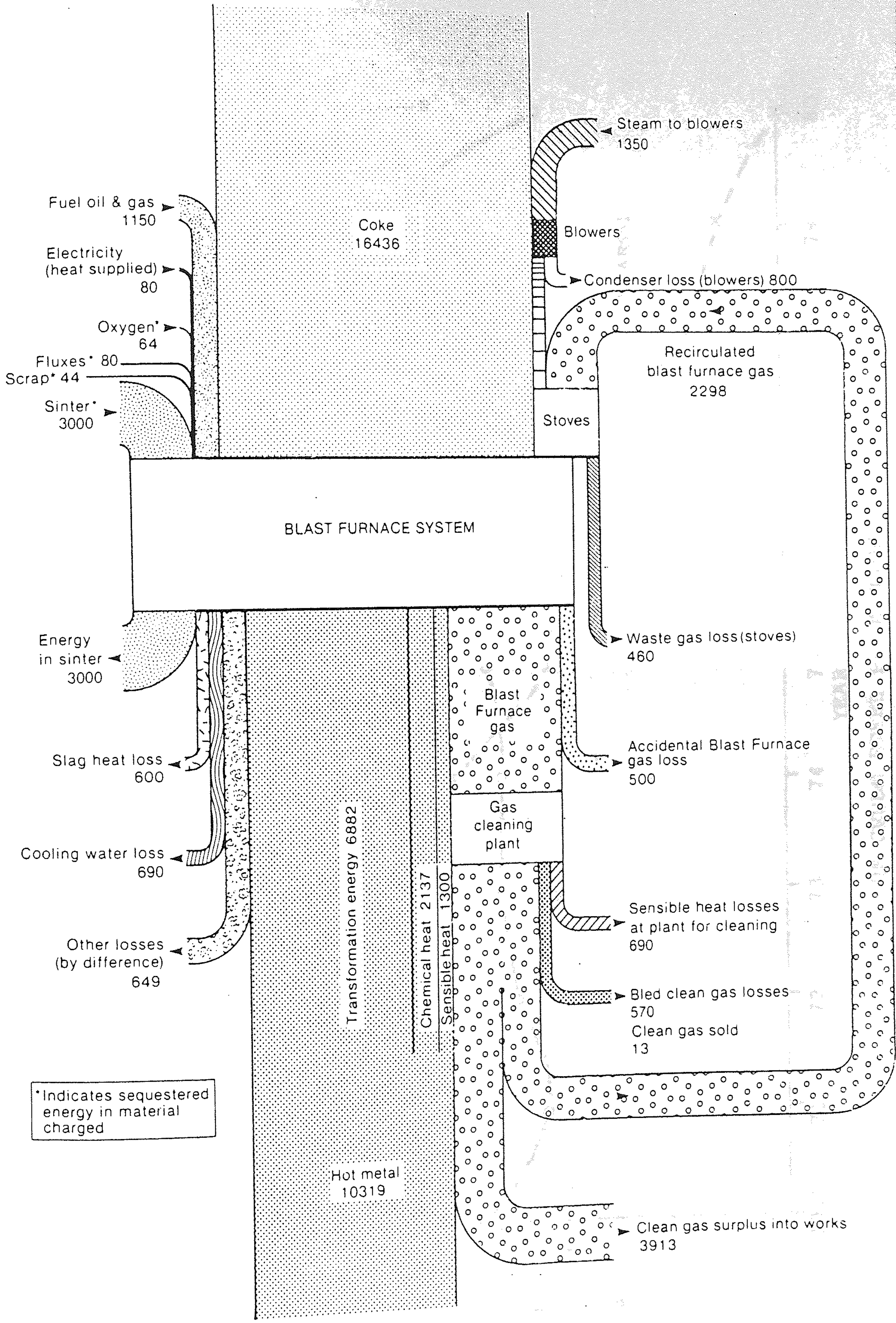
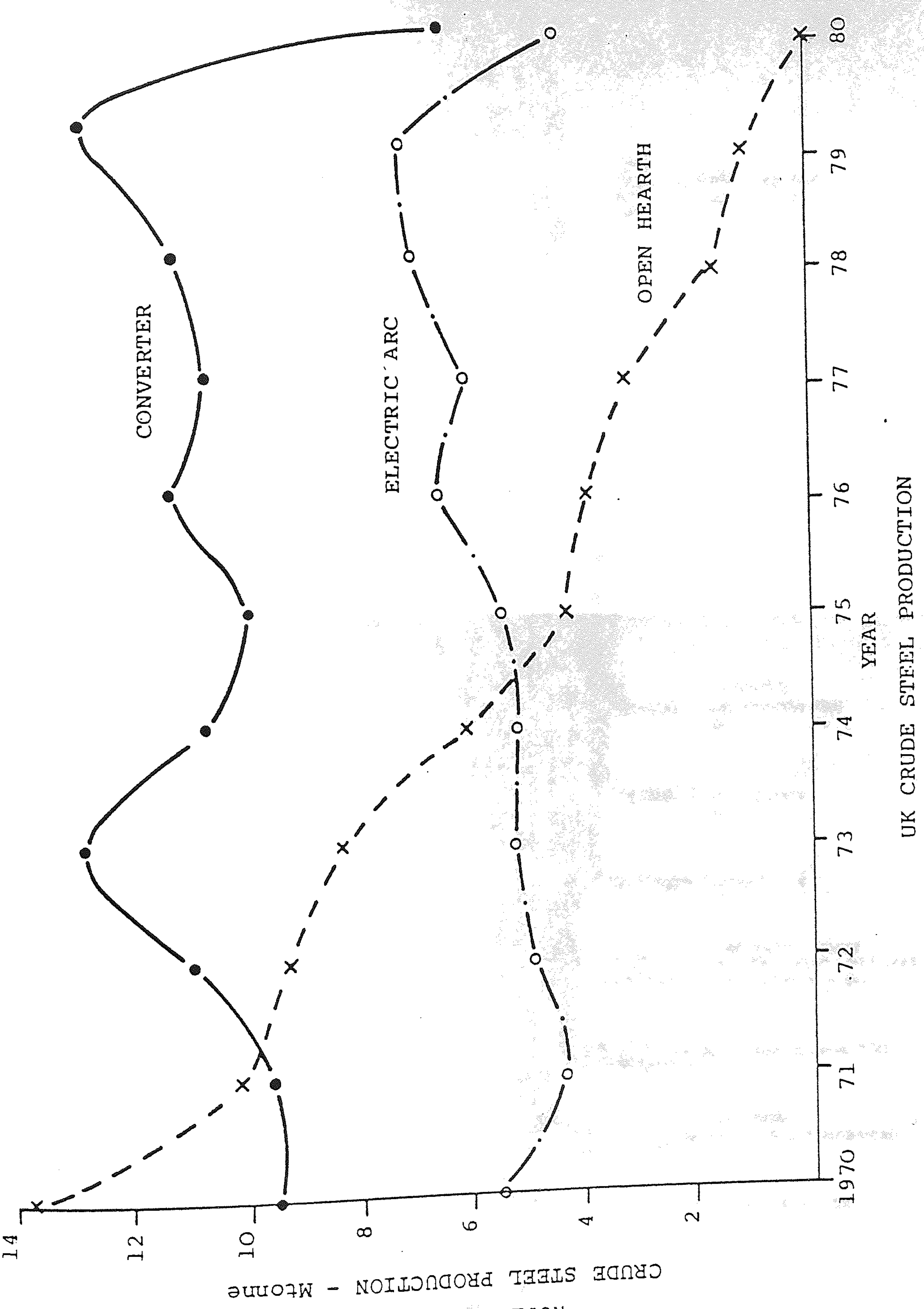
SCHEMATIC LAYOUR FOR BLAST FURNACE : IRON MAKING

FIGURE 1.13



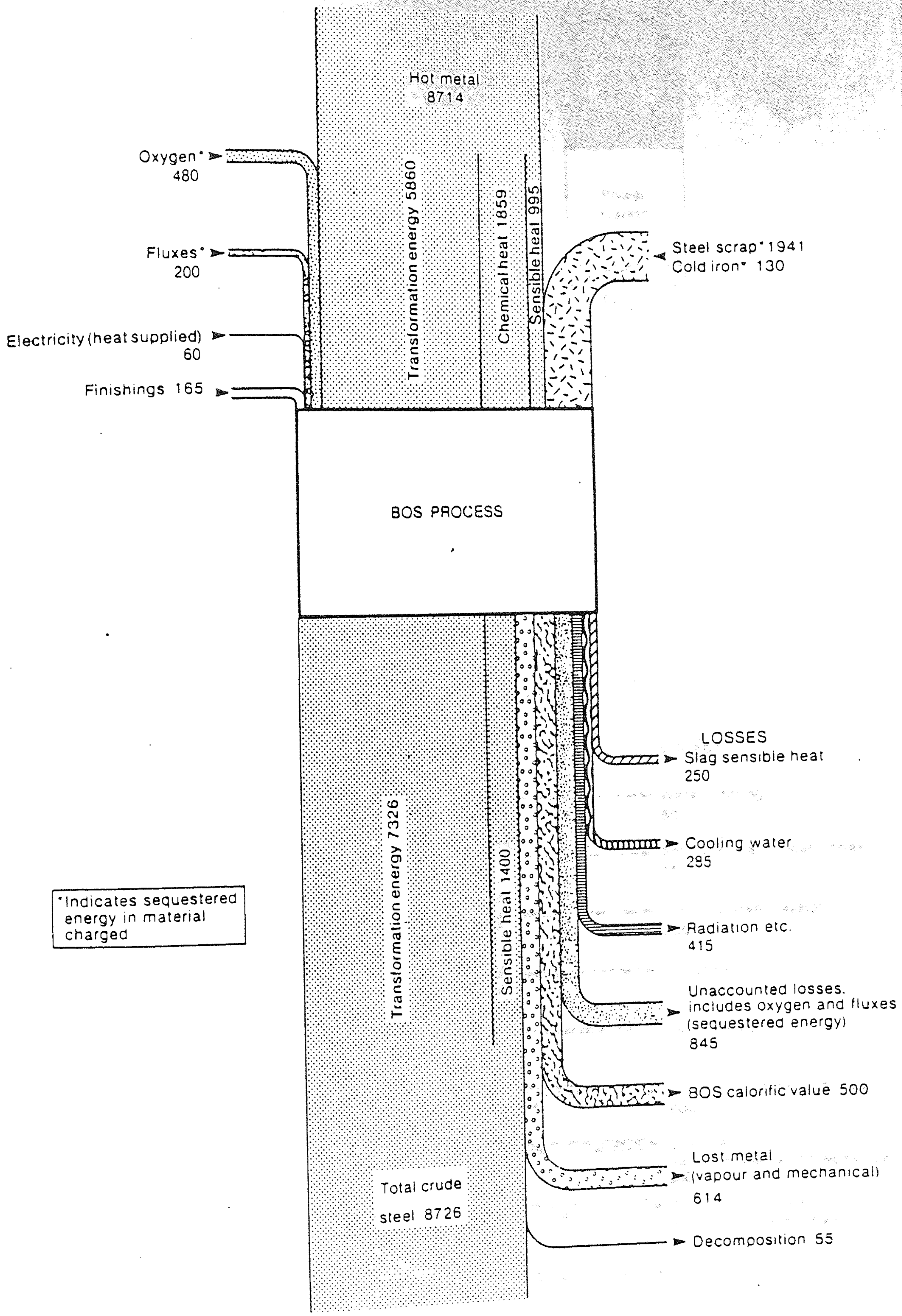
ENERGY UTILISATION IN IRONMAKING

FIGURE 1.14



UK CRUDE STEEL PRODUCTION

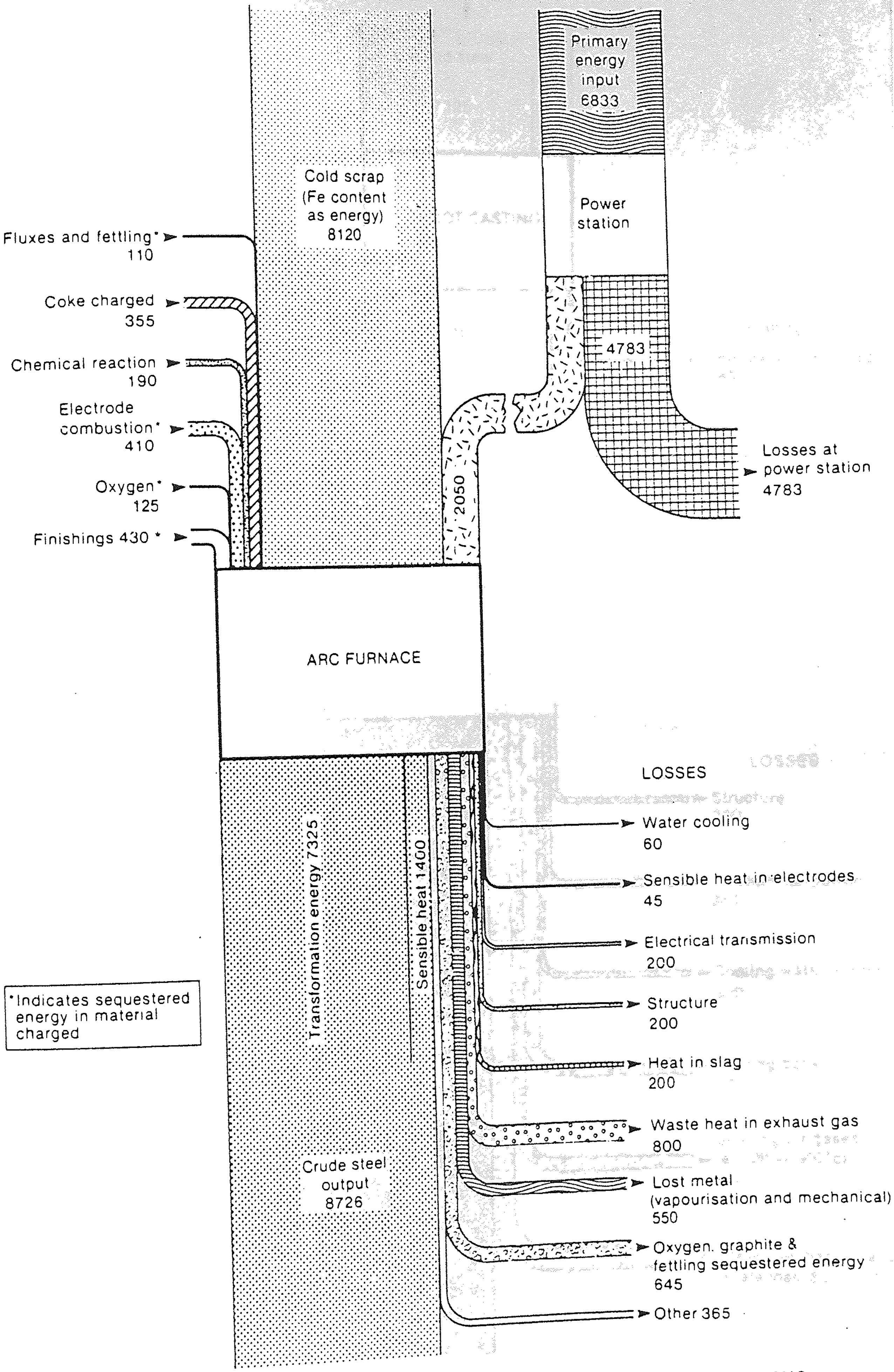
FIGURE 1.15



ENERGY UTILISATION IN BOS STEELMAKING

MJ/tonne Crude Steel (1979)

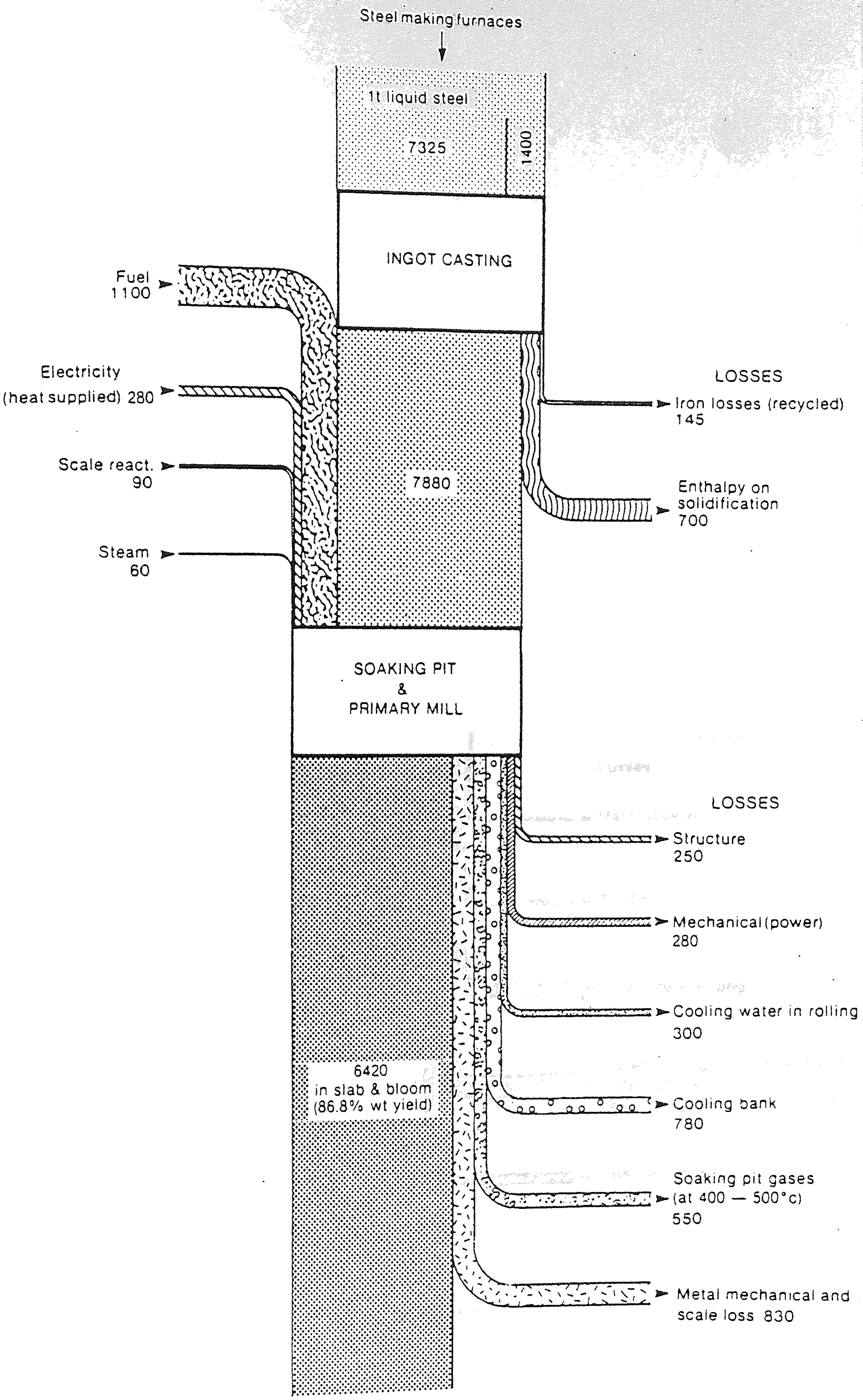
FIGURE 1.16



ENERGY UTILISATION IN ELECTRIC ARC FURNACE STEELMAKING

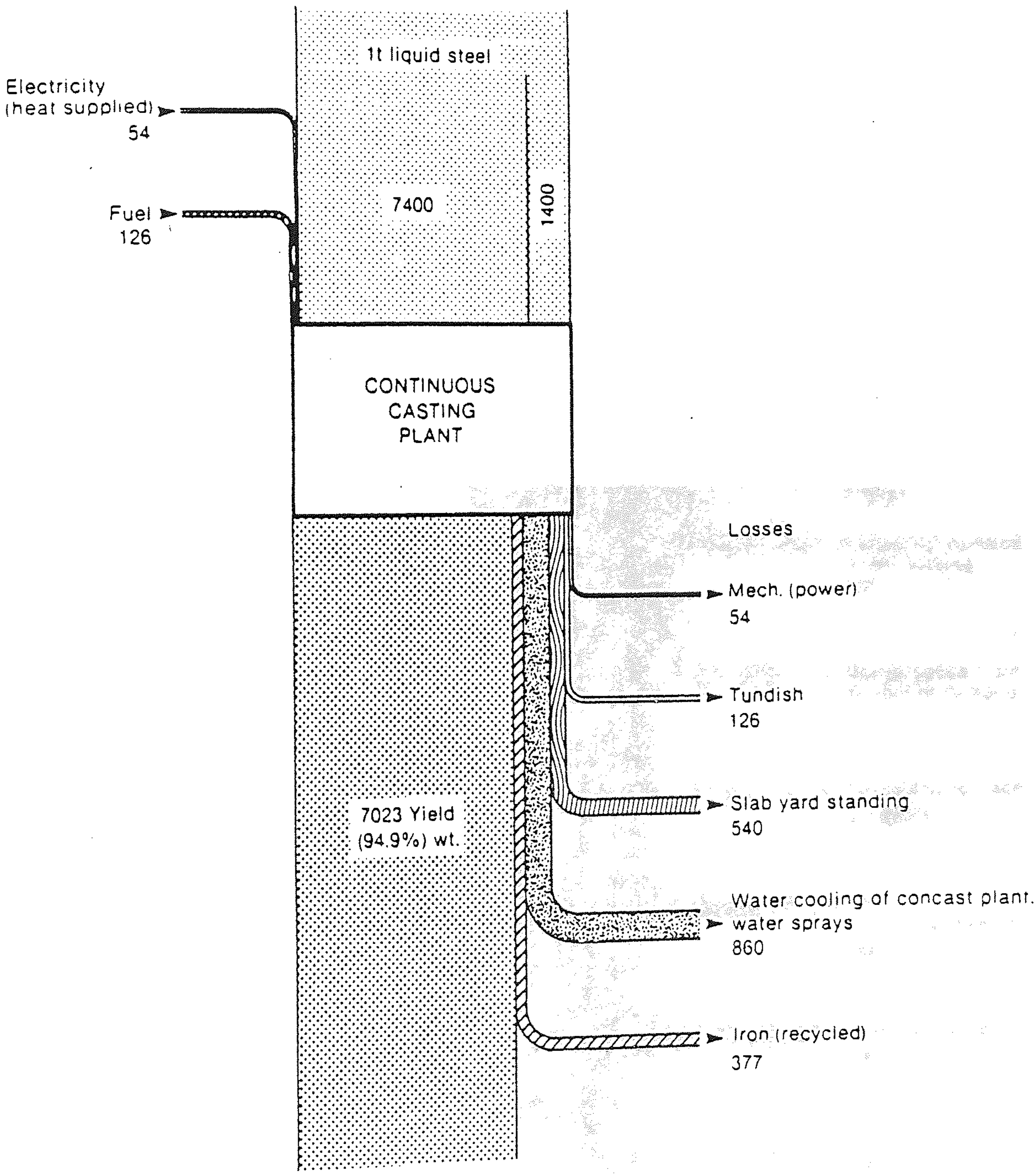
FIGURE 1.17

MJ/tonne Crude Steel (1979)



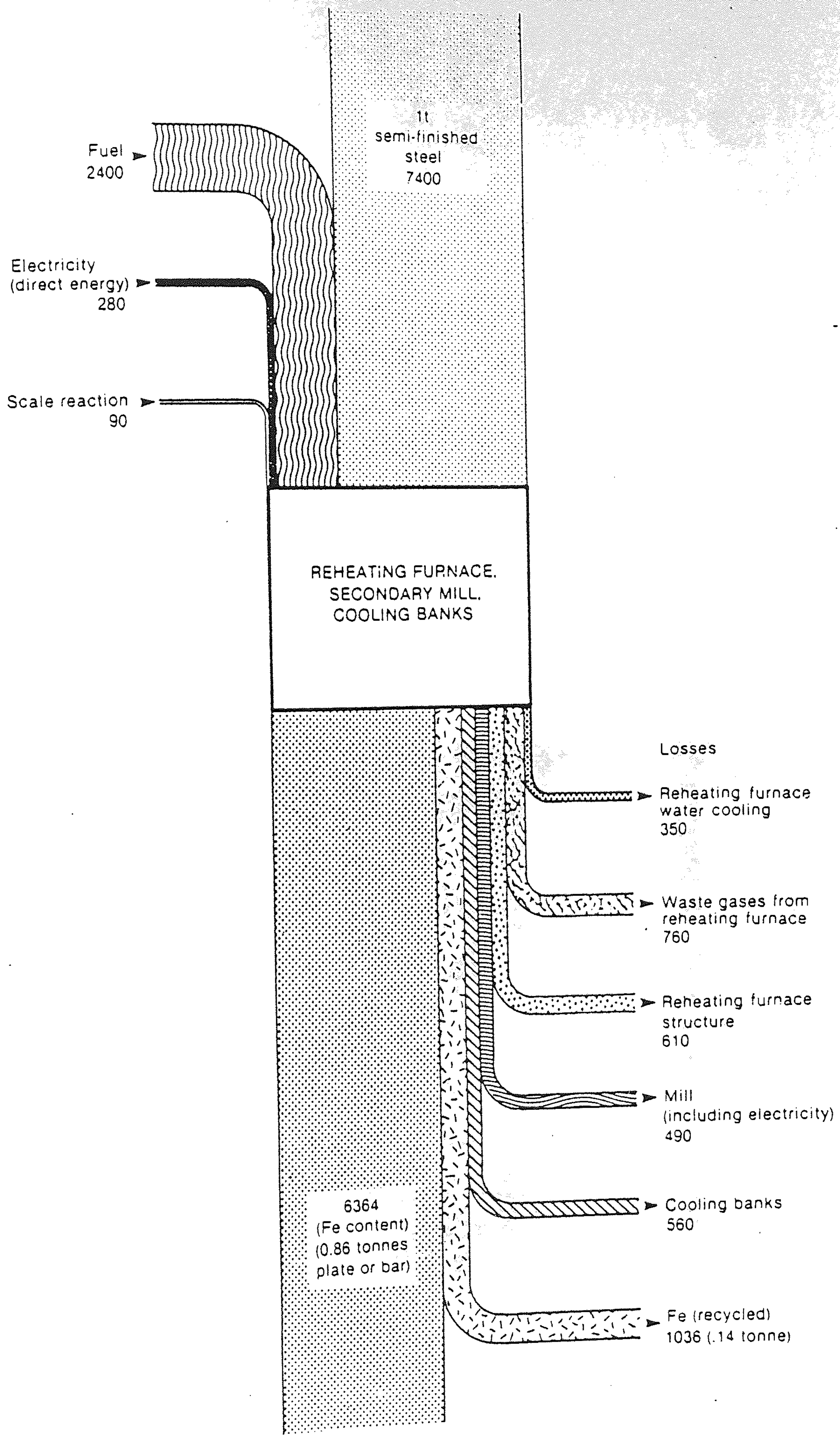
ENERGY UTILISATION IN SEMI FINISHING
(TRADITIONAL INGOT CASTING ROUTE)

FIGURE 1.18



ENERGY UTILISATION IN SEMI FINISHING
(CONTINUOUS CASTING ROUTE)
MJ/tonne Crude Steel (1979)

FIGURE 1.19



ENERGY UTILISATION IN SECONDARY FINISHING

MJ/tonne Crude Steel (1979)

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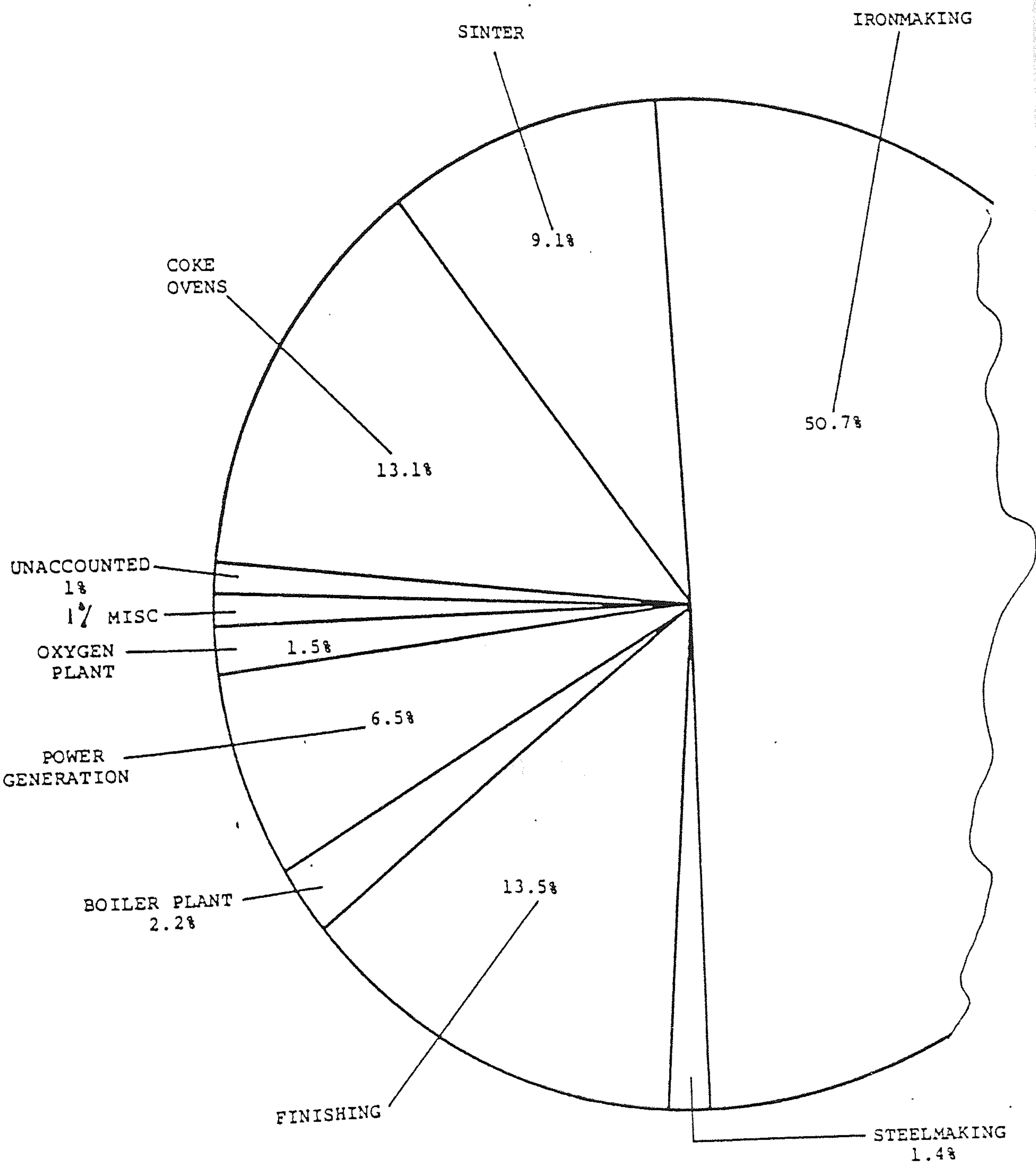
SANKEY DIAGRAM FOR AN INTEGRATED STEELWORKS

FIGURE 1.21



Energy Consumption in Steelmaking

FIGURE 1.22

CHAPTER 2

ENERGY SAVINGS: COKE OVENS

Coke ovens have been an intergral part of the steel industry for a long time. For every tonne of pig iron produced in the blast furnace almost half a tonne of coke is needed. This represents a process energy consumption of 206 million GJ for BSC for the year 1982/83 representing some 3 percent of total UK primary energy consumption.(55-56)

Blast furnace and foundry grades of coke do not simply produce heat but also provide carbon for reducing iron oxide to make pig iron or cast iron. Therefore, in order to achieve this extra quality, the coke must possess the following characteristics:(57-58)

- provide sufficient mechanical strength to support the burden charge
- low sulphur content so that sulphur is not transferred to the hot metal resulting in an increased need for hot metal desulphurisation.
- the size an porosity be such that a suitably porous bed is created in the blast furnace or cupola.

These quality parameters thus constrain the choice of coals that can be used for coke making.

For efficient blast furnace operation it is important for the coke charged to retain its original size and strength as long as possible inside the furnace.(59) Traditionally it was accepted that a good quality coke in UK could only be produced from rank* '301' coal. An estimate

---

*Rank number is coal classification system used by the National Coal Board. It is a function of the percentage volatile matter and the coking capacity of the coal.

of the workable coal reserves in the UK indicated that this traditionally favoured metallurgical coke feedstock '301' represented approximately 10 percent of the potentially available coal for use in coke ovens. Therefore, to extend the range of coals,blending has been used for many years now. In addition,the development of formed coke allows the use of lower grades and a wide range of coals as acceptable feedstock.[60-63]

Great effort and much research has been devoted to the development of processes to improve the quality of coke, to use a wider range of coal quality, to reduce the pollution in coke ovens and to decrease the overall coke consumption in blast furnaces.

An assessment of coke consumption in UK blast furnaces shows a steady decrease (Fig.2.1) over a number of years.[56]

This part of the thesis now assesses coke technology in detail and high lights areas of energy conservation.

2.2 COKE MANUFACTURE

Fig. 2.2 shows a cross-section through a typical coke-oven currently in use. The ovens essentially consist of three main parts, namely the coking chambers, the heating flues, and the regenerators, all constructed of refractory brick. The ovens are constructed in batteries to conserve space and heat. The coal is charged from a charger (or larry) car through openings in the top of the oven. After the coal has become coke, the coke is pushed out from one end by a power-driven ram or pusher. During the coking period, the ends of the coking chamber are closed by refractory- lined doors, which are constructed to seal completely the ends of the ovens.

To permit the escape of volatile matter driven from the coal during coking, an opening is provided at the top of the oven at either one or both ends of the coking chamber. Each such opening is fitted with an offtake pipe, which connects the oven with the gas-collecting main for the battery. The combustion chambers consist of a larger number of flues which permit uniform heating of the entire length of the coking chamber. Modern practice utilises the regenerative principle to preheat the combustion air to achieve higher thermal efficiency. In all modern oven batteries, individual pairs of regenerators are provided for each heating wall, or part wall, and are located under each oven. This permits separate control of the flow of preheated air for combustion to individual flue walls, and allows close control of heating.[64]

The size of the coking chamber is usually a compromise of many interrelated variables that will best suit the expected production requirements while making the best quality of coke within practical limits.

e.g. (a) coke size desired by blast furnace operator

(b) the time required to carbonise coal in chambers.

The coke once produced must be quenched to stop any further combustion. There are two methods for quenching the hot coke pushed from the ovens namely, wet quenching and dry quenching.

In wet quenching, the charge of hot coke from the ovens is received in a quenching car and transported to a quenching station where it is sprayed with water. The car is then taken to a coke wharf where the coke is discharged. It must be recognised that coke with a low moisture content is desired. This is accomplished by arranging the sprays and the time of quenching so that sufficient heat will remain in the centre of the individual coke lumps to evaporate excess surface water. The usual practice is to aim at an average moisture content of 2½% in the coke after screening.

In dry quenching the sensible heat of the coke is used for the production of steam for general plant use. This is done by dumping the hot coke into a closed system where the recirculation of inert gas conducts the sensible heat from the coke to a boiler until an equilibrium is reached within practical limits and the coke temperature is below its ignition point in air. The coke is then discharged for screening and loading.

2.3 ENERGY REQUIREMENT: COKE MAKING

The following parameters contribute to the energy flows in a coke oven.[65]

A. Energy in the Coke

For complete carbonisation reactions it is necessary to raise the coal blend to at least 1000°C. In practice this temperature is exceeded to ensure that all the coke ...tely heated. Published results from a number of coke ovens indicate that average coke temperatures range from 942-1103°C. Typically, published values indicate that the sensible heat content of coke prior to wet quenching is approximately 1.6 GJ/tonne coke.

B. Energy in the distillation products

Temperatures in the range 625-828°C have been measured in the ascension pipe. A typical value is 650°C. Energy loss of the by-products is approximately 1.0 GJ/tonne coke.

C. Energy in the waste gases

This comprises the energy content of the combustion products and the excess air. Typically, approximately 30% excess air is used to ensure complete combustion; however, the overall range reported is 20-40%. Energy loss from the flue-gas exhaust is approximately 0.4 GJ/tonne coke.

The waste-gas temperature at the regenerator exit should be as low as possible but not low enough to cause condensation in the lower part of the regenerator.

D. Energy losses from battery surfaces

These have never been measured accurately and published values have been obtained by different calculations from overall energy balances. Values within the range 0.305-0.488 GJ/tonne dry coal have been reported.

E. Heat of reaction of coal

It is generally agreed that the carbonisation reaction is exothermic. However, the extent of this is not so clear, with values in the range 0.045-0.48 GJ/tonne of dry coal being reported. The majority of the values are around 0.26 GJ/tonne and this is considered typical.

2.4 POTENTIAL AREAS OF ENERGY SAVINGS

A typical case showing the material and energy flows in a BSC coke-oven is presented in Fig. 2.3.[66]

The energy that is essentially available for recovery is that provided by the underfiring and is made up principally of stack losses and the sensible heat of the coke. However, the waste gases are rejected to the stack at less than 300°C and can only be considered as low grade waste heat source. The coke-oven gas evolved requires cooling to remove condensable constituents prior to cleaning and recovery as fuel gas. Hence this does not provide a convenient source of waste heat.

BCRA show (Fig. 2.4) that for a typical coke-oven battery over 50% of the heat from underfiring gas is normally dissipated as sensible heat of the coke and is lost in wet quenching.[67] In addition, conventional coke ovens, if incorrectly operated or maintained, can cause severe local air pollution during coal charging, coke pushing and wet quenching of coke.[68]

In the modern 'tall oven' designed coke-ovens,[69] energy ... pollution control can be achieved using 'Pipeline charging of preheated coal' and 'Dry coke quenching'.

2.4.1 Coal Preheating

Coal preheating is now a well established procedure in commercial operation around the world.[70-74]

Three systems developed are all based on similar principles for thermal treatment of the coal and differing only in the method of conveying and charging the preheated coal to the oven. The method of pipeline charging of preheated coal is described in the section below.

A. Pipeline Charging of Preheated Coal

The method consists of preheating sized and screened coal in a Rosing Type or Cerchar type preheating unit. The preheated coal is fed through a distribution hopper to charging bins. When an oven is ready to be charged, the bin is pressurised with steam and coal is propelled by steam jets to the oven. The coal enters the oven through a closed stationary system and therefore there is no opportunity for dust to escape.

Energy balances (Fig.2.5) indicate that there is a net saving in energy compared with normal coke-oven practice in which all the heating of the coal is done in the oven. Over the range of preheaters and preheating temperatures, this saving is estimated as 130-230 MJ/tonne of wet coal input or 120-330 MJ/tonne coke at the same coal moisture and coke yield. In general terms this is about one fifth of the energy savings in dry cooling.

The major advantages ofpreheating are improved thermal history of the coal charge in the oven and increased bulk density of the charge. Mainly as a result of the latter there is a significant improvement in the quality of the coke. Extensive studies carried out by the BCRA using a wide range of British and foreign coals and commercial blends (gravity-charged) have shown that the physical characteristics of coke are much improved, the degree of enhancement being generally greater for the lower rank of the coal. In addition, the consistency of the coke and the nature of its porous structure are improved, and the proportion of coke in the useful size range is generally increased. These effects are of considerable importance to the iron and steel industry because they enable the quality of blast furnace coke to be either improved (as is necessary for the larger blast furnaces) or, when using more of the poorer coals in the blend, the quality maintained at an acceptable level. Another practical consequence of preheating is that the charge need not be levelled, particularly in gravity charging.

Preheating of the charge leads to a substantial increase (between 30 and 50%) in the oven throughput, and this implies a similar percentage decrease in the number of ovens for a given battery throughput. The actual decrease depends on the method of charging (being greater for gravity charging) and also on the type of coals used. If a new coke-oven battery for preheated charging is planned for optimum machine utilisation, it will have fewer ovens than for the equivalent production by wet charging. Therefore, the capital and some operating charges per tonne of coke will be reduced.

Additional benefits of preheating include improved emission control and possibly increased refractory life when pipeline charging is used.

The main disadvantage of preheating is the danger of exposing hot coal to the atmosphere and the carryover of coal dust to the collecting main during the charging period. This is more severe with pipeline than with gravity charging.

B. Pipeline Charging: BSC Steelworks

The Corporation, fully recognising the potential, has invested in the system at three of its steelplants.

REDCAR: A schematic diagram of 'Coaltek' preheating and pipeline charging installed at Redcar Steelworks is shown in Fig.2.6.(75-76) Two batteries of 66 x 5.3 m ovens are fed with preheated coal via pipelines from 5 Cerchar preheaters. The major factors influencing the decision to utilise coal preheating was the necessity to produce coke of the quality to enable the new 14 meter furnace to achieve design production of 2.2 t/m^3/day. The system was commissioned in 1978-79.

Unfortunately due to operational difficulties ever since commissioning, the plant never achieved its designed production rate. Therefore a decision was taken in 1983 to dismantle the system and replace it with a conventional wet quenching system. The wet charging system is due to come into operation in the Spring of 1984.

SCUNTHORPE: At Dawes Lane, Scunthorpe, the blast furnace operation does not require coke of as high a quality as at Redcar. With wet charging, a blend of 88% high volatile and 12% low volatile coal produced satisfies the coke quality requirements. With preheating, coke quality could be maintained with only 6% of low volatile coal in the blend, representing a considerable saving in blend costs. At Scunthorpe three batteries of 25 x 5.3 m ovens have been constructed to be supplied with preheated coal via a pipeline from 3 x 80 T.P.h Cerchar preheaters. The system is still to be fully commissioned.

ORGREAVE: At Orgreave Works, following preliminary trials with a Rosin preheater and gravity charging via a larry car at Brookhouse, a battery of 43 x 42 m ovens was constructed, fed with preheated coal from 2 x 60 T.P.h Rosin preheaters. As at Scunthorpe the low volatile content of the blend can be reduced by 50% with preheating with no loss of coke quality.

2.4.2 Dry Quenching of Coke

A. INTRODUCTION

In the wet quenching method, the hot coke is sprayed with water to reduce its temperature and suppress combustion. The resulting steam is let to the atmosphere and the water is recirculated without any effort being made to recover the heat. Wet quenching also results in a large scale pollution problem since fine particles of carbon are carried away with the steam.

The heat content of a tonne of wharf coke is approximately 1.6 GJ. Dry cooling operates on the principle of the recovery of the major part of the sensible heat of the coke by taking the coke to a chamber through which inert gas is circulated. The inert gas takes up the heat from the coke, which is thus cooled before discharge from the chamber. The heat in the inert gas is utilised in a waste-heat boiler to generate high-grade steam, and the gas thus cooled is returned to the dry cooling chamber. The coke temperature on discharge from the chamber is usually about 200°C.[77-90]

In the operation approximately 0.3% of the coke is burned off prior to quenching and owing to extra handling the production of breeze is approximately 1% greater. Dry cooling at this level leads to the net recovery of approximately 1240 MJ/tonne of wharf coke, that is some 76% of the enthalpy above 0°C of coke of 10% of ash discharged from the ovens at 1050°C.

The materials and energy flows when practising the dry cooling of coke are shown in Fig. 2.7 which is based upon typical operating conditions and indicates that with dry cooling the coking plant becomes a net exporter of steam. Figure 2.8 indicates that, with dry cooling, 42% of the underfiring energy is recovered as steam and only 10% is dissipated as sensible heat of the coke.

In addition dry cooling eliminiates virtually all the atmospheric and quenching water pollution problems associated with the wet quenching[91] of coke and may offer an increased coke yield due to the reduction of burn-up in the coke car exceeding the burn off in the dry cooling chamber.

In addition, manufacturers of DCC equipment claim that Dry Coke Cooling (DCC) offers an improvement in the physical and chemical properties of the dry cooled coke due to improved thermal and mechanical treatment during the process and a slight reduction in coke rate in the blast furnace when compared against coke cooled by conventional wet quenching. However, no data is available to substantiate this claim.

The realisation of the full economic potential of dry coke cooling and payback period depends on complete utilisation of the recovered energy. This means that all the steam generated must be used effectively for power and/or process and/or other purposes. While this is perfectly possible to achieve on a new plant or major plant extension the same may not be possible on the existing plant.[92-93]

The major drawbacks of the DCC application to the BSC works can be summarised as follows:

(a) Very high investment costs which may only be acceptable where environmental control is necessary.

(b) Increased dust in the screening plant thus a need for improved/new cleaning plant.

(c) Introduction of new technology; i.e. a lack of handling experience.

(d) The need for complete utilisation of recovered steam in the works.

B. DCC - Technologies Available

The three main designs commercially available for the dry coke cooling system are:

i) GIPROKOKS - (C D Q) System

ii) The Waagner Biro - DCC System (94-99)

iii) Otto Dry cooling system (100-101)

The basic concept of each of the systems is very similar, where the hot coke passing through a cooling chamber is cooled by an inert gas moving counter current to the downward movement of coke. (50-55) The inert gas is thus raised in temperature to around 710-850°C and coke is discharged at 200°C. Using the hot gas stream, steam is generated through a waste heat boiler. Depending on the quantity of coke produced, DCC plant is designed on unit system consisting of one or several cooling chambers. The size and design of chamber can vary in each of these systems. Typical design data of a cooling chamber is given below:

Coke throughput per cooling unit	50 - 60 t/h
Coke temperature at inlet	1050-1100°C
Coke temperature at discharge	200°C
Cooling (re-circulating)gas flow	90-100,000 m^3/h
Gas temperature - Inlet to cooling unit	130°C
- Inlet to waste heat boiler	700-850°C
Energy recovered as superheated steam	0.4-0.45t/t of coke

It should be recognised that most of the installations currently operating in the world are based on GIPROKOKS design. A list of these installations is presented in Table 2.1. Therefore only GIPROKOKS system is discussed in this chapter in detail.

Other designs are available with good recommendations. Sulzar/Waagner Biro design was used at the Ford Dagenham plant some years ago but with the plant becoming obsolete it is now out of commission. The salient feature of the Waagner Biro process is the bunker by-pass line which branches off the pressure side of the recirculating fan to provide a constant steam production.

GIPROKOKS Dry Coke Cooling

Fig. 2.9 shows a line diagram of the GIPROKOK system. In this system the hot coke from the coke oven is transferred to a position under the Dry Coke Cooling plant hoist shaft in a coke charging basket where it is lowered in a pre-chamber. The purpose of the pre-chamber is to hold a considerable reserve of coke, at a uniform temperature, which allows for interruptions to the hot coke feeding operation and contributes to the stabilisation of the temperature of the re-circulating gas, ensuring a steady steam raising condition. The hot coke is gravity fed into the cooling chamber. The cooling of the hot coke is achieved with inert gas circulated by means of a fan in an enclosed gastight system. The hot gas, on reaching the upper section of the cooling chamber passes through a series of sloping flues where coarser coke breeze particles, entrained in the hot gas, are removed thus protecting equipment downstream from erosion. The hot gas then passes from the dust separator on into a waste heat boiler where steam is generated.[102-109]

The now cooled gas passes from the base of the waste heat boiler and is directed through cyclones where the fine particles still entrained are removed. The gas is thus ready to duct back to the suction side of the main circulating fan to repeat the above cycle.

The cooled coke is discharged at the bottom via a special automatic discharging system onto a conventional trough belt conveyor system to existing coke wharf.

A burner is supplied with an amount of combustion air and coke-oven gas to assist and maintain a continuous burn off of any combustibles in the recirculating gas to avoid any explosion hazards. The position of the gas burner at this point offers utilisation of heat of combustion at the waste heat boiler with an additional advantage of a temporary heat source during excessive interruptions in hot coke supply.

A Sankey diagram for a 52 t/h C.D.Q. unit is presented in Fig. 2.10.

C. DISCUSSION

An extensive literature survey together with first hand discussions with the manufacturers and users of the DCC equipment abroad was carried out. The conclusions drawn from the survey are presented below.

From the work carried out by Nippon Steel and Nippon Kokan in Japan there seems little doubt that dry cooled coke does offer better quality coke in terms of high strength, low volatile content, low reactivity and uniform grain size distribution when compared with wet quenched coke. In addition, the claims made by manufacturers with regards to energy recovery and improved environment, are also met.

The trials with dry cooled coke carried out in the Soviet Union in blast furnace of 2300 m^3 capacity state that the specific coke rate is reduced by 2-3 percent due to the high quality indices of the dry cooled coke, as compared with those of wet quenching. However, no further work to substantiate this has been carried out and it is doubtful that this reduction in coke rate in the blast furnace can be achieved solely due to dry cooled coke.

Reports published in USSR talk of poor performance of some of the DCC installations. This is usually attributed to low standards of maintenance, allied to the psychological buffer of reliance upon reserve wet quenching facilities. It is generally accepted that, where this buffer does not exist, a high standard of performance is automatically achieved. This is illustrated by Cherepovets & West Siberian Iron & Steelworks. It is also Russian practice for an installation to have very substantial reserve capacity possibly to cover a chamber failure. Because of high capital costs and a different method of economic assessment, the Western world will be reluctant to accept this practice and certainly it is not accepted by the Japanese plants. Ogishima [104] and Tobata [80] dry cooled plants provide smooth operation under high driving rate conditions.

One recurrent theme in C D Q development is the necessity to improve the anti-pollution standards achieved by the operation of DCC installation. However, no attention appears to have been given to the foul and dirty bleed which results from "after-burning", this being itself a consequence of the use of a pre-chamber and the tendency for an explosive mixture of gases to accumulate in that part of the system. It is clear that additional equipment would be required to deal with the bleed should a DCC system incorporating a pre-chamber be adopted in UK practice.

A recent British Patent of Giprokoks introduces a by-pass in the gas circulation system ahead of the boiler. The steam output from the boiler is monitored so that, when the energy supply falls as a result of increase in the concentration of combustibles, a small proportion of the circulating gas is directed through a combustion chamber supplied with fuel gas and air. This arrangement which (as far as is known) has not yet been applied to any production module, is claimed to avoid uncontrolled admission of air for after- burning to eliminate combustible components of the recirculating gas and to keep constant the rate of heat supply to the boiler.

The cooling chambers of the installations at Chiba and Tobata are of the standard Russian size, giving a throughput of 56 t/h. The modules at Ogishima, Keihin have larger chambers, of rated throughput 70 t/h. There appears to be a feeling in Japan that even 70 t/h is too small for chambers serving high capacity coke-oven batteries. Nippon Steel have stated that they are working on the design of a 150 t/h chamber (and on different heat recovery systems) and IHI (Ishikawajima-Harima Heavy Industries) Co. Ltd., have indicated their interest in the design of 120 t/h chambers. There is no theoretical reason why the Russian design should not be extended to chambers of larger capacity (and this of course has happened at Ogishima. However, no known

Russian plant is yet operating with chambers of capacity greater than 56 t/h; though a 72 t/h is being commissioned at Tyssen.

Waagner-Biro AG of Vienna, and its wholly owned subsidiary, the American Waagner-Biro Company Inc. (Pittsburgh), continue to offer designs of various rated throughputs, up to 120 t/h, with no pre-chamber. They have recently introduced and patented designs for a compact combination of the dry cooling chamber and the steam generator. Their most important recent innovation (patented) consists of a chamber bypass line (including a calalytic inert gas conditioner). The cold gas flow in the bypass is controlled and mixed with the hot gas coming from the cooling chamber, so that the temperature of the gas entering the steam generator and thus the steam output, can be kept constant in spite of irregular coke charges and/or withdrawals. This effectively eliminates the marginal advantage which the Russian design, with pre-chamber, might have maintained even in a multi-modular installation, in regard to constancy of output. In relation to the recent modification by Giprokoks in this area, it must be observed that the Waagner-Biro approach still seems simpler and cheaper. Clearly, the absence of a pre-chamber substantially reduces the capital cost of the installation; it also must reduce the maintenance problems and costs, the wear on the hanging firebrick lining of the pre-chamber being so high that the repair costs form a substantial proportion of the total maintenance costs for DCC in the USSR.

The Dravo Corporation has patented a system combining dry cooling and wet quenching: this involves dry cooling to 350-400^{O}C, followed by cooling to 100-150^{O}C by water sprays, and is claimed to reduce the volume of gas to be circulated in dry cooling, to avoid the need to cool the gas to low temperature and to yield coke which is not so dusty as completely dry cooled coke. This may be taken as indicative of growing interest in the potential market for DCC.

In 1976 Waagner-Biro, in a worldwide study of DCC plants and quotations, found an average pay-off time of 3.35 years (and some of the schemes embraced measures for the control of the emissions on discharge of the coke from the ovens, which is costly and shows no financial return). Russian experts in recent conversations have said that the cost of DCC plant is recoverable in three to five years. though in practice it could be longer (upto eight years).

It is clear that, under the best conditions, there remains an irrefutable economic case for DCC, a case which can only be strengthened by any consideration of the environmental factors and/or the natioanl need for energy conservation. It should also be emphasized that none of the assessments mentioned above makes any allowance for improved blast furnace performance (essentially, lower coke rate) which may result from the use of dry cooled coke. In practice, the case must be modified by considerations relating to the cost of plant modifications necessitated by the addition of DCC to existing coke-making facilities, the achievable steam utilisation level and the nature of the fuel saved by the production of steam from DCC (to which the potentialities for the production of high-grade steam and the use of this steam for the generation of electricity constitute an important corollary).

D. CASE STUDY: ECONOMIC ASSESSMENT OF DCC APPLICATION IN BSC

In the new BSC reorganisation, assessments of all six major coke-oven plants ie. Scunthorpe (Dawes Lane), Port Talbot (Morfa Bank), Llanwern, Orgreave, Ravenscraig and Redcar based on 1979 prices, is made. Only in one case, Llanwern, it is concluded that there was no economic case for DCC installation (consideration of the environment and energy conservation being ignored). For the remaining five plants, economic cases of varying degree of compulsion emerged from the studies. These five studies are therefore summarised (in plant alphabetical order) in the succeeding section of this chapter.

It is clear that the availability of any reasonable capital subsidy would strengthen the arguments for DCC at any particular plant. A major subsidy which could possibly be obtained for such a project at certain BSC plants (e.g. Ravenscraig) would be in the form of a Development Grant, which might amount to as much as 40 percent of the capital requirement.

The government through the Department of Energy has made available £20 million for an expanded programme of demonstration projects in energy-savings over the next few years. The objectives of these schemes will be to achieve an annual energy saving of at least £5 for each £1 of government contribution and to achieve an overall replication of at least six for demonstration projects, though the final decision of assessment is left at the discretion of the programme committee. Such capital subsidies may amount to 25% of the requirement. Following discussions with the D.O.E. programme committee it was established that D.C.C. would qualify for such support.

A government subsidy for as much as 100% could also be available to assess the feasibility of coal preheating/ D C C and as much as 50% for developing the system.

Another possible source of capital subsidy is the E.E.C., which is operating a scheme of support for novel projects with energy-saving potential. It seems unlikely that D C C per se would qualify, because of its lack of true novelty, but an integrated scheme for coal preheating/ D C C might have a good chance of success.

SCUNTHORPE STEELWORKS

Although there are four separate coke-making plants in Scunthorpe, the high capital cost of dry coke cooling plant and the age of most of these batteries means that the only logical location of a D.C.C installation is at the Dawes Lane plant. The scheme giving the best return would be a two-chamber unit with a wet quench as standby. The boilers would then generate steam at 13.8 bar and feed directly into the process steam main.

Assuming 80% utilisation and a steam recovery of 0.5 tonne per tonne of coke cooled and further assuming the recovered steam can all be absorbed in the process steam distribution system, the scheme offers a fuel oil saving of £1.825m pa. Taking into consideration the operating and maintenance cost of £600,000 p.a the scheme offers a net return of 15% on a capital cost of £8.4 million ignoring depreciation.

An alternative means of utilising the D.C.C recovered steam would be to generate electricity in an adjacent generating plant operating in parallel with the grid. Under these conditions the steam recovery would be 0.45 tonnes per tonne of wharf coke equivalent to a steam make of 48.6 t/h providing a gross electrical generation of 11.0 MW. This valued on 1978/79 electricity tariff generates a revenue of £1.53m pa. Taking operating and maintenance cost of £747,000 pa this offers a net return of 7.1% (ignoring depreciation) on a capital cost of £10.96 million, (this includes a generating plant cost of £2.5 million).

Another alternative to these would be coal preheating which technically has never been applied. The Dawes Lane ovens are equipped with a coal preheating plant for which the heat source is CO gas. The replacement of this fuel gas by DCC could offer a gross annual fuel saving of £11.41 million.

PORT TALBOT (MORFA BANK) STEELWORKS

This study is based on the new development at Morfa Bank assuming a 1982 coke output of 25,000 tonnes/week. Assuming 100% dry coke cooling, the system will require standard modules and a standby unit (ie 2 x 100 t modules with a standby unit).

The economic assessment shows that if the recovered steam could be absorbed in the process steam distribution system, the scheme would offer a net saving of £1.63 million per annum for an estimated capital cost of £14.0 million giving a return on capital of 11.7% with utilisation factor of 80%.

However, the recovery of power generation offers a net saving of £1.75m pa for an estimated capital cost of £16.0 million giving a return of 11% on capital.

It can be concluded that there is a case for the installation of D.C.C. at Morfa Bank and should be further investigated.

LLANWERN STEELWORKS

Any scheme which produced steam at Llanwern would lead to a reduction in steam raising at the boilers and consequently an increase in B.F. gas bled as there is no other alternative use for this gas. Therefore steam recovered by Dry Coke Cooling will have to be accompanied by either a direct steam consumer e.g. electrical generation or a consumer for the B.F. gas not required at the power plant.

Thus the economics of dry coke cooling installation at Llanwern has been assessed using the electrical generation route and assuming 1977/78 standards of 23,940 tonnes of coke/week throughput.

A power generation of 7.25 MW can be achieved by the installation of a D.C.C. unit on No.5 batteries and a total of 15.25 MW can be recovered by installing a further D.C.C. unit on No. 1-4 batteries. Assuming an 80% utilisation, this offers a gross savings of £1.87 mpa and a net savings of £1.22 mpa. The discounted cash flow (at 10%) at constant price level would indicate that in order to achieve a nill return on capital outlay, the value of total capital that can be spent would be £9.35 million. With the alternator costing in the region of £2.5 million, the maximum amount available for purchashing the D.C.C. unit is £6.85m. This seems insufficient to finance even the cheapest D.C.C. scheme of 3 units entailing one unit on each battery plus a common standby, which geographically may not be possible.

It is concluded, therefore that based upon energy savings alone D.C.C. cannot be justified at Llanwern.

ORGREAVE WORKS

There appears to be sufficient space to site a D.C.C. unit in a convenient place close to the coke oven batteries but this needs to be checked by detailed engineering drawings.

A single module, 90 tonnes/hour coke capacity, D.C.C. unit could be capable of handling the coke output at Orgreave and steam produced by the unit could be wholly utilised either as process steam or in power generation.

An economic evaluation of the case for D.C.C. has been carried out, based on a direct comparison between a D.C.C. unit and a conventionally fired boiler to raise the same quantity of steam. In this study it was recognised that overall availability of D.C.C. plant might be as low as 85% but credits were allowed for full utilisation of steam produced with coke oven operation at 100% (560,000 t.p.a. wharf coke). The saving in fuel for steam raising was calculated at £591,000 per annum but additional maintenance and electricity costs estimated for D.C.C. operation reduce overall saving in operating costs to £454,000. The capital cost of the D.C.C. unit including boiler and water treatment is estimated at £6 million, but this is partly offset by £1 million in boiler plant requirement. These figures show that the saving in operating costs would almost cover the 10% interest charged on the capital but would not be sufficient to pay back any of the initial capital outlay.

There would, however, be an economic case for D C C. If an interest free loan could be obtained or if some of the other potential advantages of D.C.C. were to be taken into account (i.e. improved B.F. operation and use of dry breeze as a blend additive) then the pay back period is calculated at 11 and 8 years respectively.

REDCAR COKE OVEN BATTERIES 1 & 2

An economic assessment on batteries 1 and 2 show that a net fuel saving of £2.9 million p.a. can be achieved which will offer a 15% return on an investment after depreciation of 5% should the maximum capital cost be £14.5 million. Additional savings could be made if the plant installed generated steam at sufficiently high pressure to provide power generation and further reused at lower pressure for process steam.

It must be noted that the installation of a D.C.C. plant to serve batteries 1 and 2 would be difficult and involve considerable civil engineering work and could be expensive. The case is made to look worse by the presence of temporary excessive steam raising capacity where the local people may exert pressure to retain the temporary package boiler plant as a viable standby facility.

RAVENSCRAIG STEELWORKS

At Ravenscraig there is a surplus capacity at the power station generators of 15.5 MW. This surplus could be taken up by using the steam produced from D.C.C.

Using the power produced from D.C.C. steam would reduce the electricity requirement from the Scottish Electricity Generating Board by up to 11.4 MW at 85% of the fully developed capacity at Ravencraig. This is equivalent to a net saving of £1.4 million per year after taking into consideration operating costs but before depreciation.

Preliminary examination suggests that there is a financial case if the capital costs of the installation can be kept below £11 million, at which stage a DCF of 9.75 per cent is anticipated. To reduce the capital costs, a DCF of 14.5 per cent could be obtained.

Further examination, based on the MRF of 38.000 tonne hot metal per week reveals a net return before depreciation of £1.0 million per year and a DCF of 4.4 per cent. But if area Development Grant is allowed for in the capital costs a DCF of 8.5 per cent is anticipated.

It is appreciated that there could be serious site constraints at Ravenscraig and therefore it is proposed that a more detailed engineering examination be undertaken to assess the feasibility and overall viability of such a scheme.

2.4.3 Utilisation of D.C.C. Recovered Energy for Coal Preheating

It has long been recognised that the energy recovered in D.C.C. could be used in ways other than as power and/or process steam. Between 20 and 30 years ago, Sulzer Brothers Ltd. (the pioneers of D.C.C.), published papers dealing with alternative methods of utilisation of the energy, including the application of hot-air turbines, gas turbines and combined steam/gas-turbine processes. Even earlier (1932), a U.S. Patent outlined means for preheating coal using energy recovered in D.C.C.

It is the integration of D.C.C. and preheating of the coke oven charge which today seems the one attractive alternative.(110-114) A broad proposal in this area was patented by BSC in 1973. A report in 1975 indicated that pilot-scale work on integration was planned in the USSR but no further information has yet been released. A major design effort has been made by Waagner-Biro AG(115) and the American Waagner-Biro Co. Inc., resulting in numerous British, American and other Patents. Their schemes, and a patented contribution from Babcock-BSH AG(116) are further considered below.

To put the idea into perspective, it may be noted that, on average, 52 percent of the heat input from the underfiring gas to the coke ovens is normally disspiated as sensible heat of the coke going for wet quenching; with DCC, 42 percent is recovered as steam and only 10 percent is dissipated as sensible heat of the coke. Consider now that some of the energy recovered in DCC were used for preheating of the coke-oven charge; on average (at about 9.5 percent coal moisture), about 70 percent of the recovered energy would be required for preheating the coal to 220°C (coal temperature on exit from preheater) in either a Büttner-Rosin type entrainment preheater or a fluidized-bed preheater, leaving about 30 percent (13 percent of the input) available for other uses. If it were available as reasonably high-grade steam, the latter quantity would approximately match the requirements of the by-product plant. Moreover, with preheating absorbing 70 percent of the recovered energy, the minor variations in the steam output from DCC would in no measure impair the necessary consistency of the energy supply to the preheaters. Thus the integration of DCC and preheating of the coal charge can lead to an excellent balance and would represent a significant step towards a closed energy cycle on coke ovens.

Waagner-Biro have prepared three schemes which would permit the realisation of the integration in commercial practice. The simplest approach (and the one also adopted by Babcock-BSH AG) would dispense with the waste-heat boiler of the normal dry -cooling scheme, the heat taken up by the circulating gas stream being recuperatively exchanged to a secondary gas stream. This secondary gas would be used as the heating medium in a two-stage entrainment preheater. This appears to have two considerable disadvantages: (a) for use in the entrainment preheater, the secondary gas would have to be heated to about 650°C, thus necessitating the use of austenitic stainless steel recuperators, which would be expensive; (b) the residual energy (30 percent of the total recovered in dry cooling) would be available as sensible heat of gas at a relatively low temperature, so that it could be used only as low-grade or hot water, which is certainly not attractive.

A modification of this approach would use the same basic combination of DCC and preheating, but with part of the circulating gas used directly in the preheater. The other part would be used in a waste-heat boiler, smaller than that of the normal DCC module. As compared with the previous scheme, the costly heat exchanger would be avoided and the energy not required for preheating would be available as high-grade steam. However, the overall design would be expensive and would involve a costly recirculating-gas cooler (a refrigeration unit), this being required for the removal of the moisture, picked up in the preheater, before return of the gas to the dry-cooling chamber. It may be noted that this arrangement would make it possible to cool the coke to about 100°C, instead of the usual 200°C.

The third approach would effect the integration by using the energy recovered in DCC to generate steam in the normal manner and then using this to preheat the coal in a fluidized-bed unit with submerged heating surfaces. Originally, it was intended that a little of the circulating gas would be used directly for the fluidization of the bed; in the latest proposal, an independent stream of inert gas (e.g. nitrogen or products of combustion of blast-furnace gas) is suggested for this purpose. It is clear that very little dust would leave the fluidization bed and the retention time of the coal could be extended as much as necessary. There would be very little danger of overheating or oxidation of the small coal particles.

This scheme appears to be, not only the best of the three approaches, but a practical and economic way of combining the beneficial effects of charge preheating and DCC. It is simple in design and if, for any reason, the DCC installation was out of action, the coal could be preheated by using any other available steam source. Fluidization beds take up little space; a height of about 3.5m is envisaged. Very exact temperature control is possible and all the coal particles could be preheated to the same degree. The system would be safe; the explosion danger in preheating would be eliminated and thus the necessity for the complicated, costly and unsatisfactory arrangements associated with entrainment preheaters would be obviated. In addition, the corrosion problems would undoubtedly be less severe than in an entrainment system, where costly materials and desulphurization of the fuel gas are essential. Further, interesting and potentially valuable claims have been made in regard to techniques for partial desulphurization of the coal while being preheated in the fluidization bed.

Overall, the advantages of integrating DCC and charge preheating may be identified as follows: (a) the realisation of the full economic potential of DCC would depend on the attainment of a level of utilisation of steam for power/ process purposes of only about 30 percent; (b) the efficiency of utilisation of nearly three-quarters of the recovered waste heat would be much higher (at 90 or more percent) than if it were used as steam for generating electricity (25-30 percent). Given the practical and economic advantages of the final scheme outlined above, there would seem to be a strong case for considering the integration of DCC and charge preheating through the use of a steam-energised fluidized-bed preheater.

2.5 CONCLUSIONS

The energy essentially available for recovery in coke ovens is that provided by the underfiring and is principally the stack losses and sensible heat of the coke. The freshly discharged coke from the oven contains approximately 50% of underfiring energy. The exhaust gases from heating flues are rejected to the stack at less than 300°C and can only be considered as a low grade heat source. The gas evolved in the carbonisation process requires wet quenching to remove condensible constituents prior to cleaning and recovery as fuel gas and therefore does not provide a readily accessible source of waste heat.

Coal preheating and formed coke enable a wide range of coals to be used for coke making whilst retaining the coke quality. Performance data on coal preheating suggests that an energy ... of 0.10-0.33 GJ/tonne of coke is achieved.

The dry coke cooling (DCC) offers the most significant opportunities for energy conservation in the industry with an energy saving potential of 10 million GJ per year (360,000 tce/year). Recent developments in DCC and its growing application in USSR and Japan and now in Europe are fully discussed in the chapter.

All major BSC coking plants are considered for DCC application. It is concluded that under 'Greenfield' conditions, the case for the adoption of DCC remained strong. In circumstances of retrofit, the plant layout and energy balance at the particular works will have considerable influence on investment returns. The capital costs of DCC are relatively high but returns of up to 15% on capital invested are possible. Subsidies may be obtainable in the form of Development Grants, in certain areas, under the government schemes to provide energy conservation in industry. At worst, DCC may be regarded as an important anti-pollution measure which will at least pay for itself during the life time of an installation. Ravenscraig is possibly the best site for the first installation.

It is recognised that a reduction of coke rate of the order of 2 percent, such as has been claimed in the USSR, if supported by data from Japan would have a profound effect upon the economics, possibly halving the payback period from a typical eight years to four years.

Process combination of DCC with charge preheating offers:

(a) an increase in energy efficiency of the coking system to 50%

(b) an increase in the coke oven gas supply by around 20% compared with the use of wet coal

(c) an increase in the flexibility of the coking system in relation to coal as charge material and a greater independence from coal

and (d) a contribution towards reducing environmental pollution with the coking process by reducing the required number of coke ovens.

An economic evaluation in graphic form is presented in Fig.2.11. The realisation of the full economic potential of DCC depends on the attainment of a level of utilisation of steam for power/process purposes. Given the practical and economic advantages, there seems a strong case for considering the integration of DCC with charge preheating.

No.	No. of Chambers	Year put into operation	Constructed by	Place of Construction	Notes
1	2	1965	Giprokoks	USSR	
2	1	1966	Giprokoks	USSR	
3	1	1976	Giprokoks	USSR	
4	4	1968	Giprokoks	USSR	
5	6	1969	Giprokoks	USSR	
6	13	1970	Giprokoks	USSR	
7	6	1971	Giprokoks	USSR	
8	6	1972	Giprokoks	USSR	
9	-	1973	Giprokoks	USSR	
10	2	1974	Giprokoks	USSR	
11	11	1975	Giprokoks	USSR	
12	8	1976	Giprokoks	USSR	
13	5	1977	Giprokoks	USSR	
14	6	1978	Giprokoks	USSR	
15	4	1979	Giprokoks	USSR	
16	1	1976	Nippon Steel Corporation	Japan	Under Licence
17	3	1976	IHI	Japan	Under Licence
18	5	1976	Nippon Kokan KH	Japan	Under Licence Licence (Others under construction)
19	5	1976	-	Romania	Under Licence
20	4	-	-	Turkey	Planned
21	4	-	-	Pakistan	Planned
22	4	-	-	Nigeria	Planned

TABLE 2.1

DCC INSTALLATIONS

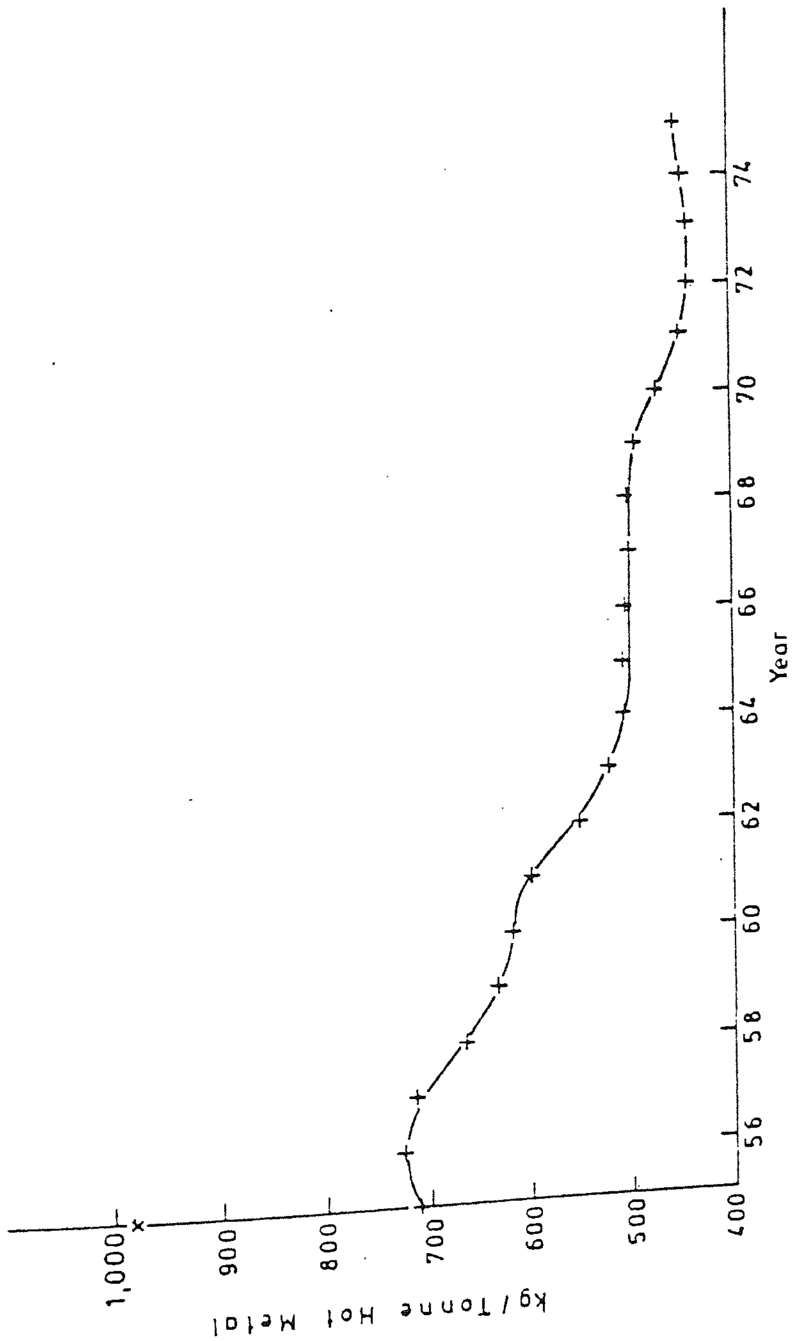


FIGURE 2.1

COKE RATE IN KG PER TONNE HOT METAL

Aston University

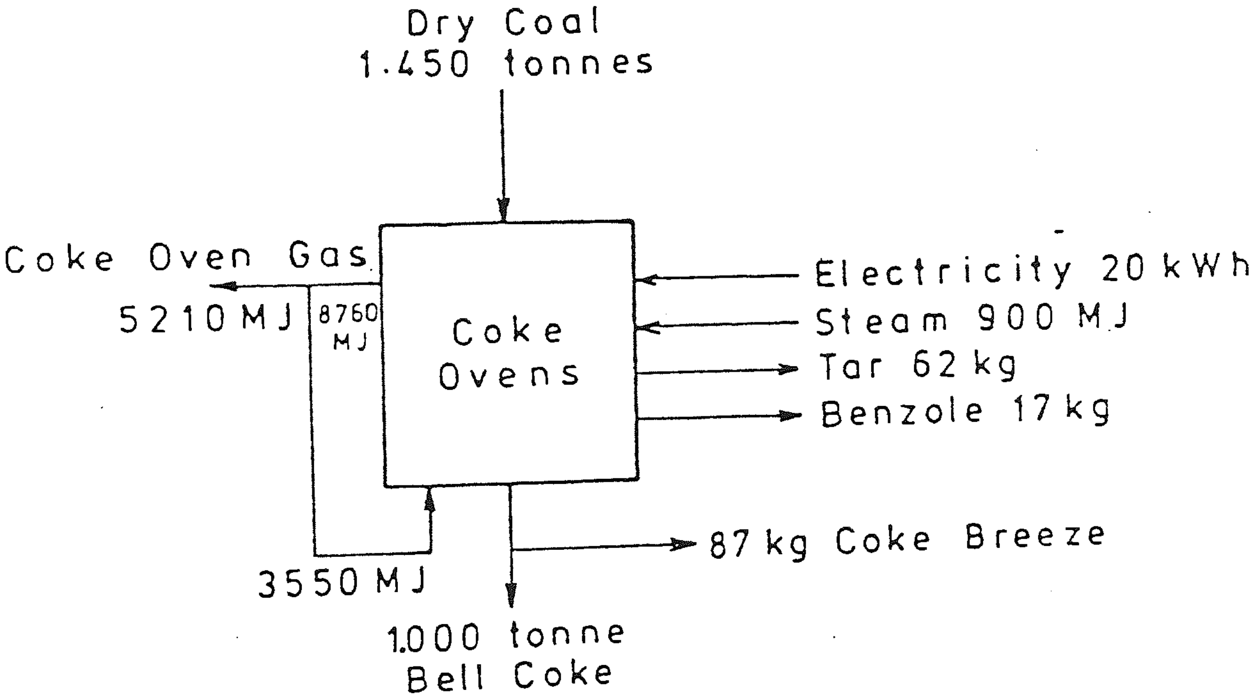
Illustration removed for copyright restrictions

Aston University

Illustration removed for copyright restrictions

CROSS SECTION THROUGH A TYPICAL COKE OVEN

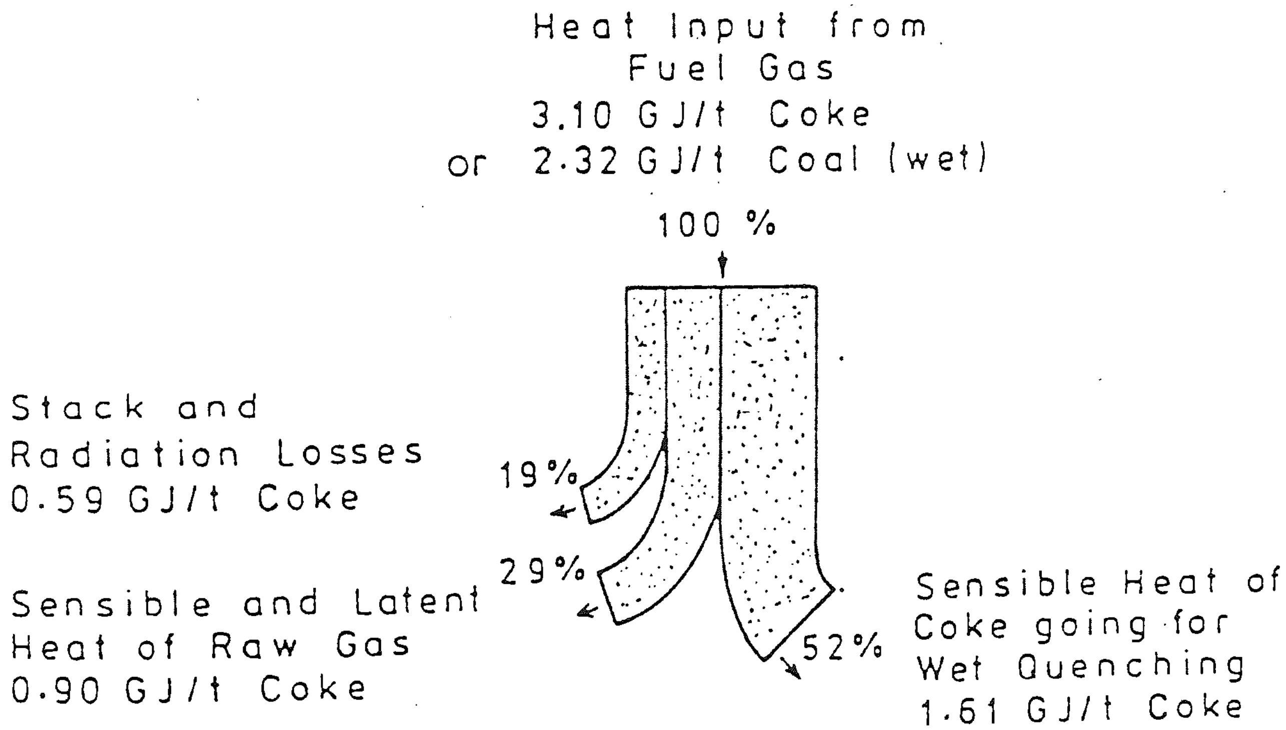
FIGURE 2.2



Bell Coke Yield 0.69 t/t Coal(dry)
Underfiring Rate 2.45 GJ/t Coal (dry)
2.26 GJ/t Coal (8% moisture)

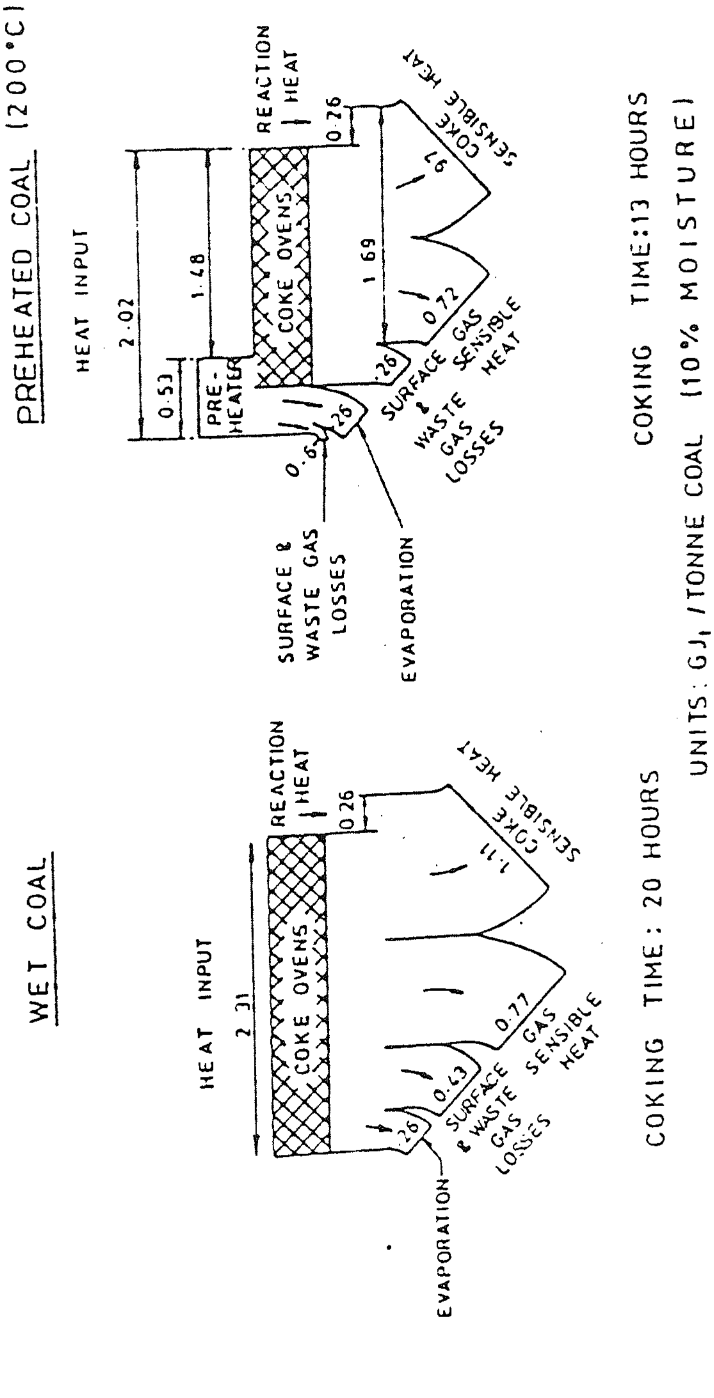
TYPICAL MATERIAL AND ENERGY FLOWS IN A COKE OVEN

FIGURE 2.3



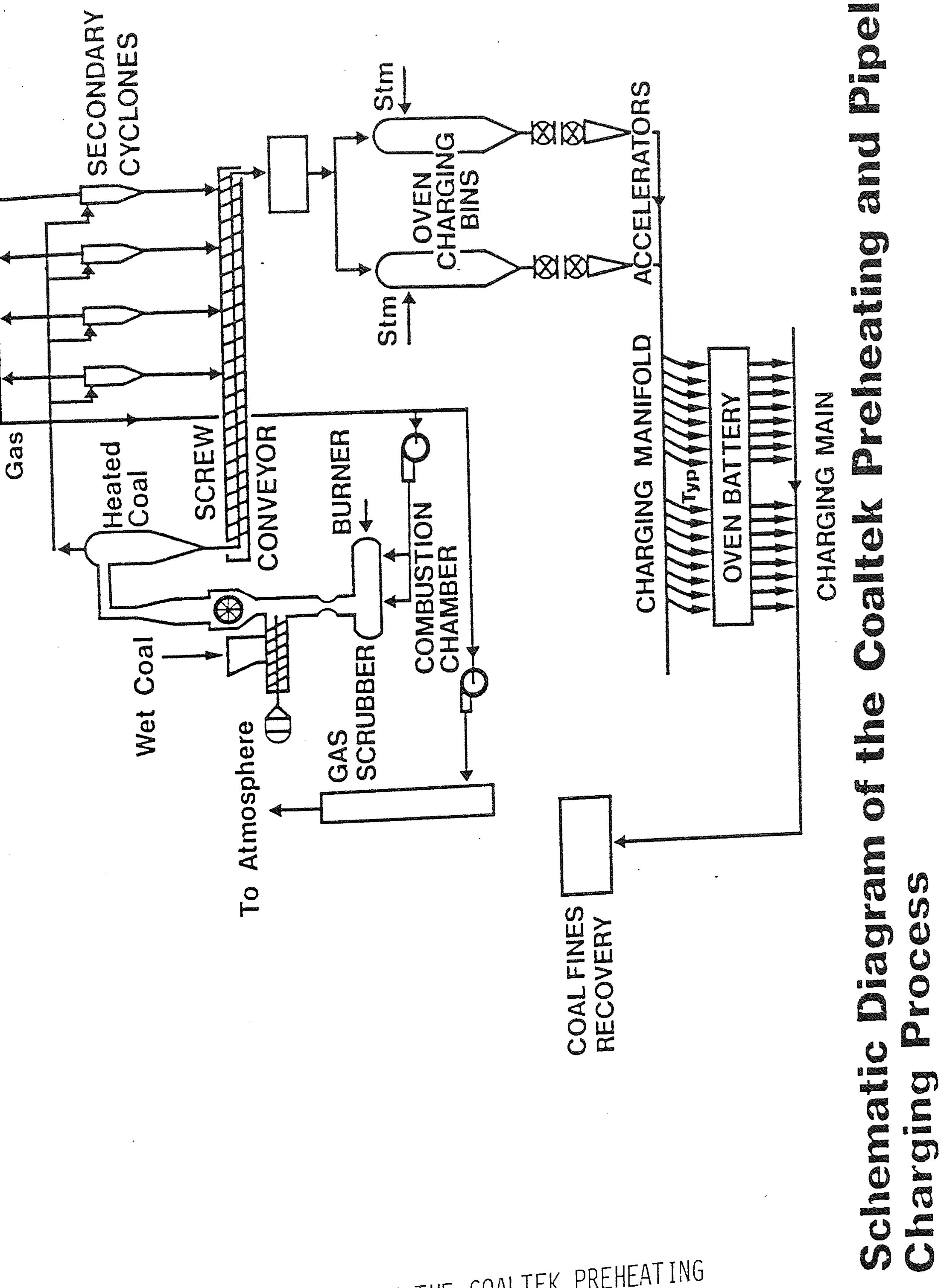
TYPICAL HEAT LOSSES ON A
COKE OVEN BATTERY

FIGURE 2.4



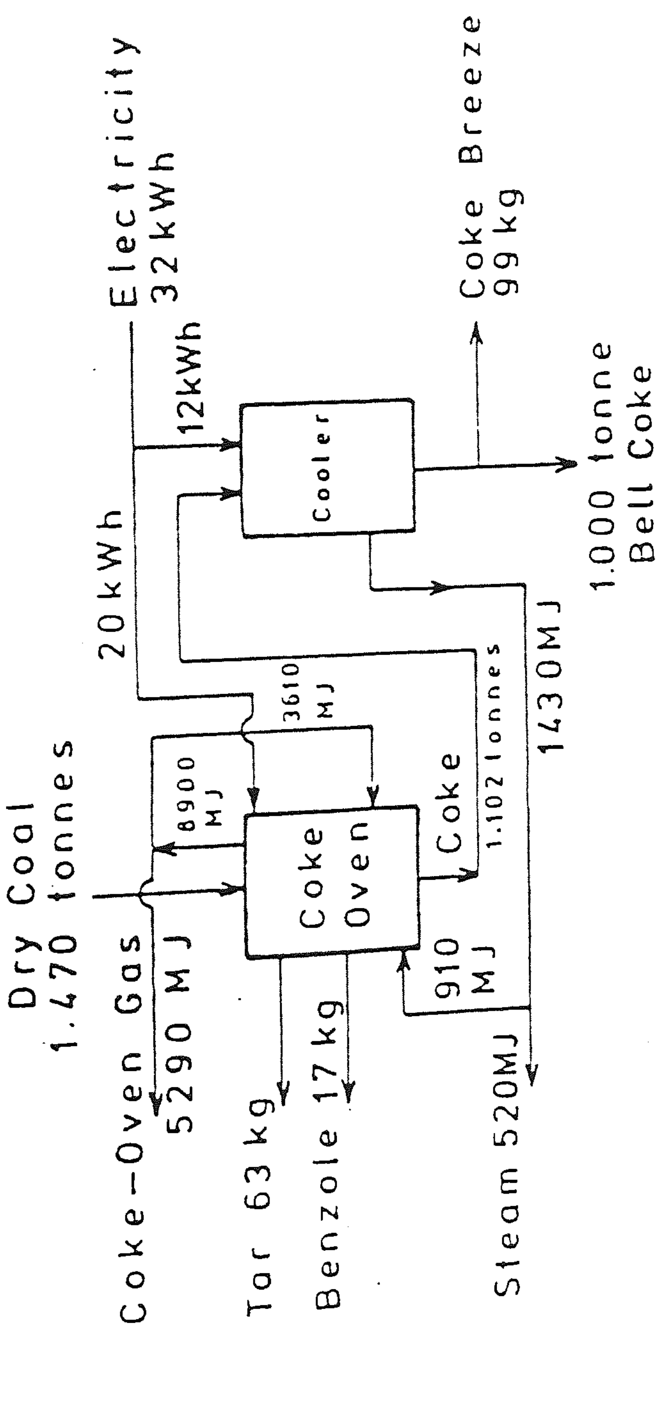
COMPARATIVE HEAT BALANCES FOR CONVENTIONAL AND PREHEATED COKE

FIGURE 2.5



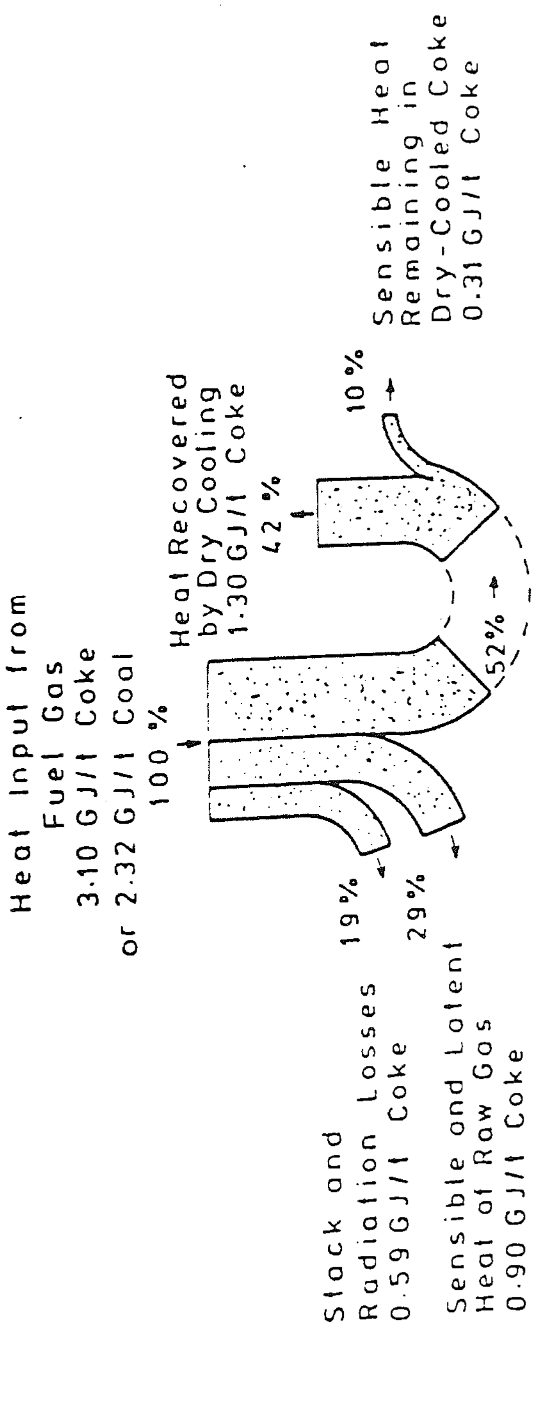
SCHEMATIC DIAGRAM OF THE COALTEK PREHEATING AND PIPELINE CHARGING PROCESS

FIGURE 2.6



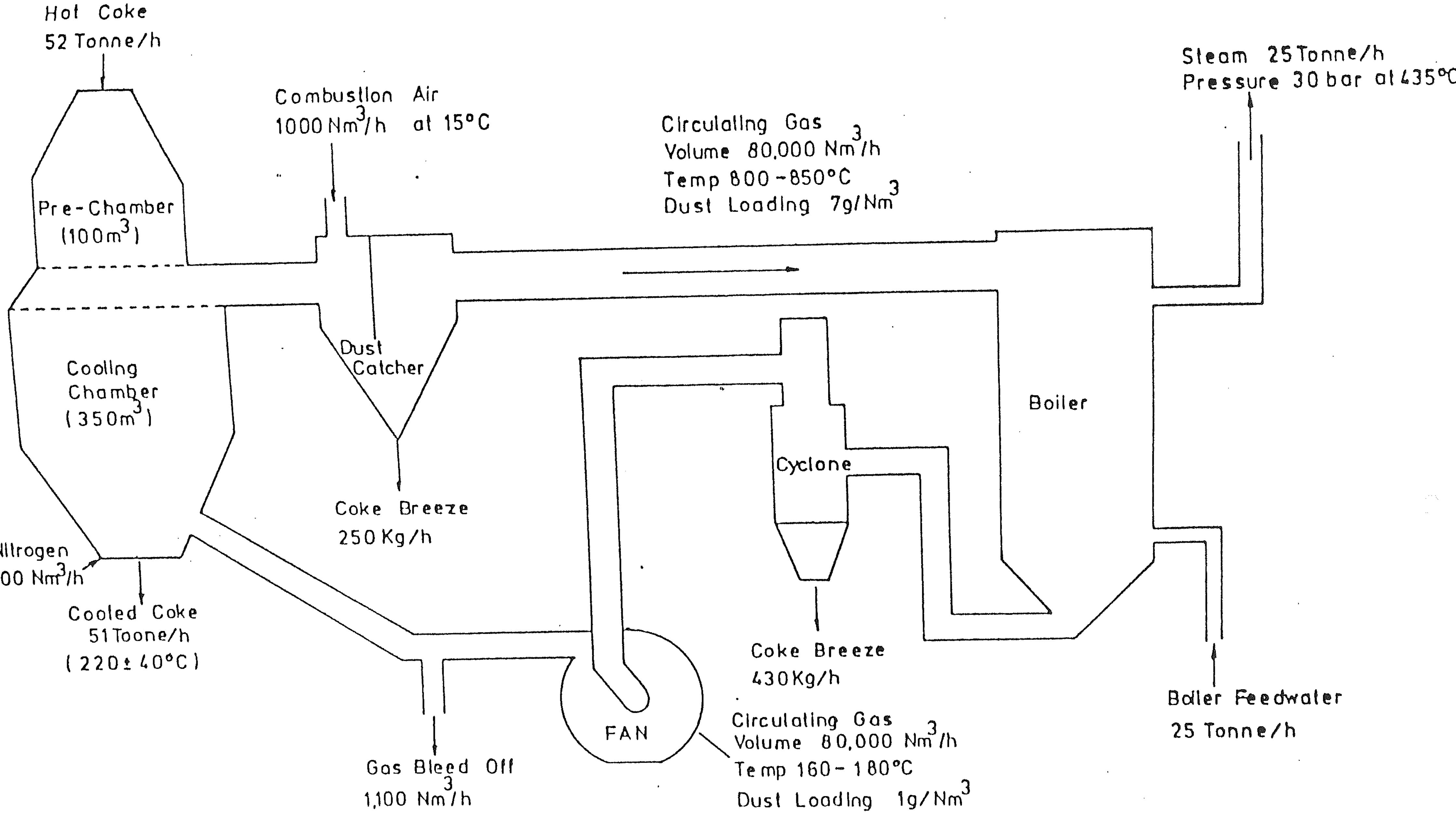
MATERIALS AND ENERGY FLOWS FOR DRY COKE COOLING PRACTICE

FIGURE 2.7



ENERGY UTILISATION IN COKE
MAKING USING D C C

FIGURE 2.8



Hot Coke
52 Tonne/h
Combustion Air
1000 Nm3/h at 15°C
Circulating Gas
Volume 80,000 Nm3/h
Temp 800 - 850°C
Dust Loading 7g/Nm3
Steam 25 Tonne/h
Pressure 30 bar at 435°C
Pre-Chamber
(100m3)
Cooling
Chamber
(350m3)
Dust
Catcher
Boiler
Cyclone
Coke Breeze
250 Kg/h
Cooled Coke
51 Toone/h
(220 ± 40°C)
Coke Breeze
430 Kg/h
Boiler Feedwater
25 Tonne/h
FAN
Circulating Gas
Volume 80,000 Nm3/h
Temp 160 - 180°C
Dust Loading 1g/Nm3
Gas Bleed Off
1,100 Nm3/h

BASIS 52 Tonne/h input to Dry Cooling Plant (Units GJ/h)

Datum 0°C

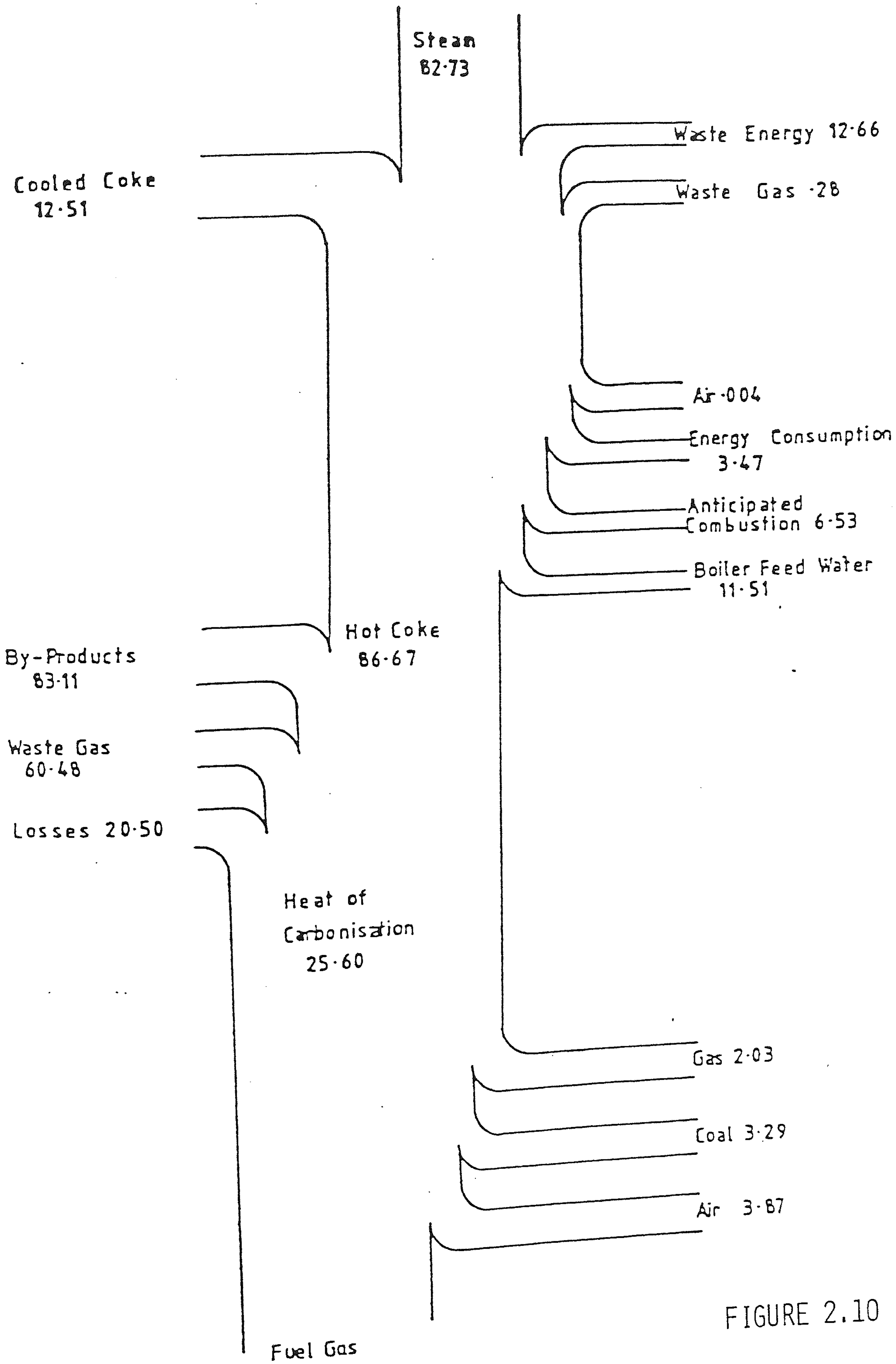
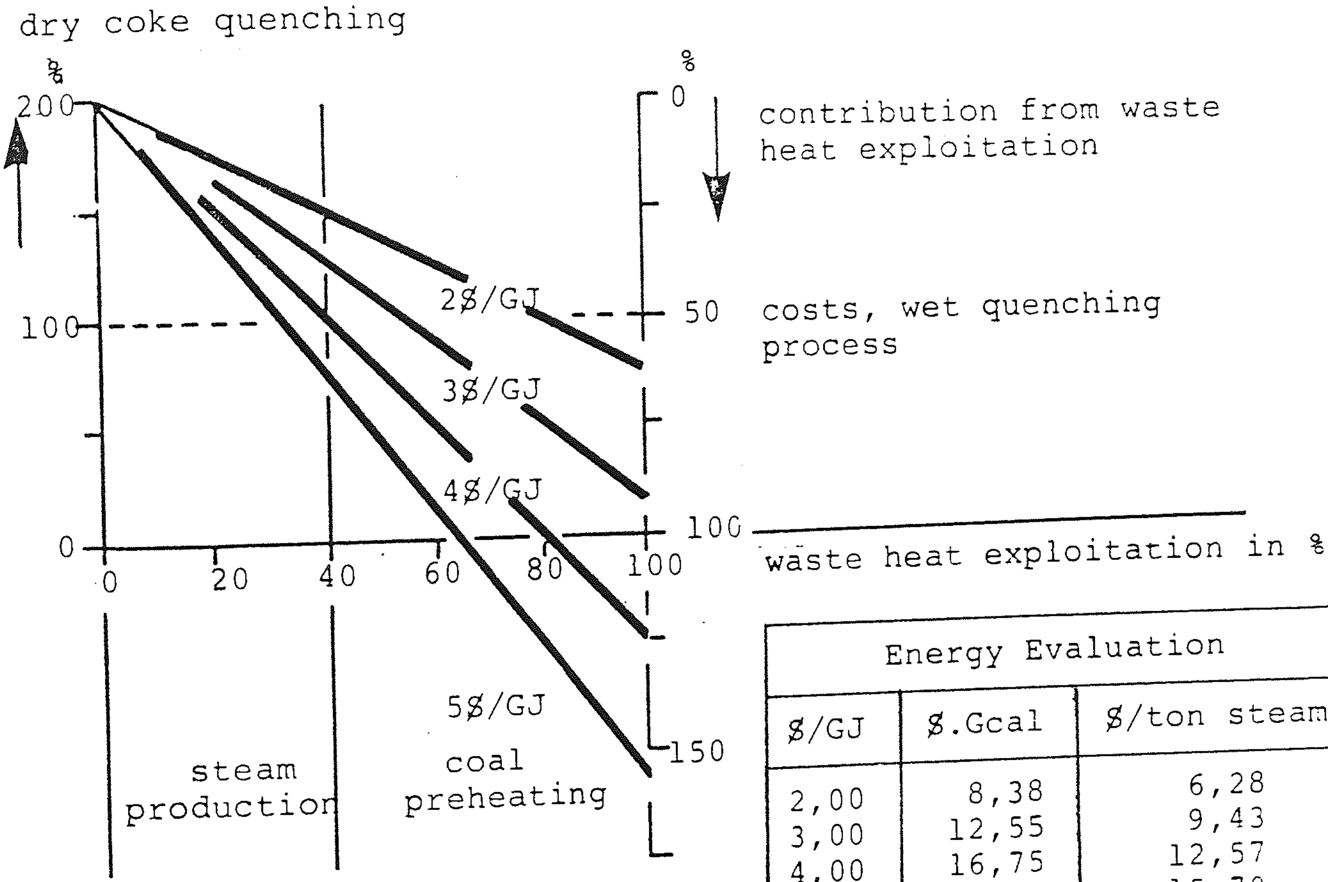


FIGURE 2.10



Energy Evaluation		
$/GJ	$.Gcal	$/ton steam
2,00	8,38	6,28
3,00	12,55	9,43
4,00	16,75	12,57
5,00	20,95	15,70

COMPARISON WET QUENCHING/DRY QUENCHING
INCLUDING ENVIRONMENTAL PROTECTION

FIGURE 2.11

CHAPTER 3

ENERGY SAVINGS: SINTER PLANT

3.1 INTRODUCTION

The use of high quality burden material is an essential pre-requisite for an economic operation of a blast furnace. Since sinter constitutes a major portion of the burden, special importance is attributed to its quality and low cost production with an eye on high plant availability.

The main process parameters in sintering are bed height, return fine rates and suction.[117] The parameters required to achieve maximum specific output are very different from those which are needed to obtain a sinter with maximum strength. In the modern development of sintering techniques, production targets are generally set with a view to achieving a maximum sinter strength with a minimum of coke rate. An improved permeability is thus used not to raise the production rate but to reduce the suction and increase bed height. Under these conditions a higher strength sinter with lower coke rate is produced.

With an escalation in energy costs, since early 1970's special attention is currently being devoted to reduce the specific energy consumption and thus production costs of a sinter plant. In the modern plants this is being achieved through:[118-119]

i) improvement of ignition process i.e. use of newly designed ignition furnaces.

ii) heat treatment of sinter

iii) utilisation of process waste heat.

3.2 HEAT BALANCE OF SINTERING PROCESS

The sintering process is initiated by the coke breeze or CO gas on the sinter mixture surface. The ignition is considered concluded if the ignition process is maintained by the oxidation heat of coke alone without further external heat supply.[120] A typical material flow in a sintering plant is presented in Fig.3.1. The major portion of heat in a sintering process lies in the coke combustion. The volume of regenerative heat in the overall heat balance of the sintering process is almost 50 percent. A simplified method for calculating the regenerative heat in the combustion zone of the bed being sintered is proposed by Klochko.[121]

In the sintering bed air reaches the combustion zone in a preheated condition because of the continual movement of the combustion zone in the direction from which air is drawn. The regenerative heat of air and mix is not from an external source but is a part of the heat of finished sinter and of the gases leaving the combustion zone. The proposed simplified method is based on the relations governing the heating of air passing through a fixed bed of porous material at the same temperature throughout the heat exchange region. As it moves through the pores of the material the air is heated and its temperature t_a rises from the initial t_a' (at the inlet to the bed) to the final t_a'' (at the outlet from the bed). By taking the distance Δx from the top of the bed a region of infinitely small thickness dx with an increase in dt_a in the air temperature it is possible to compile a heat balance for the region for a time interval ($dt = dx/U_b$) for a column of material of cross-section A.

According to the laws of convective heat exchange, the quantity of heat dq received by air from the region of bed can be given by

$$dq = K_v * A * dx * (ts' - ta) * dt \qquad 3.1$$

Alternatively the heat taken by air, in terms of increase in temperature can be expressed by the term

$$dq = A * U_o * dt * ca * dta \qquad 3.2$$

Simplifying 1 and 2

$$\frac{Kv}{U_o * ca} (ts' - ta) * dx = dta \qquad 3.3$$

where ta = ta' @ $\Delta x = o$ giving

$$ta = ta' + (ts' - ta')\left[1 - e^{-\frac{K_v \Delta x}{caUo}}\right] \qquad 3.4$$

If x is taken to be the whole bed thickness then from equation (4) air temperature at the outlet from the bed ta = ta" can be calculated.

In the sintering process where the heat front is moving with respect to combustion zone there is a counter current of air characterised by water equivalent of the stream of air.

$$Wa = FU_o ca \quad \& \quad Ws = F\ Xs\ cs\ w$$

and their ratio

$$m = \frac{Wa}{Ws} = \frac{U_o\ ca}{Xs\ cs\ w} \qquad 3.5$$

For $U = U_o / f$ 3.6

$$mo = ca * f/cs * Xs$$

where m can vary from o to infinity.

If the air velocity greatly exceeds the heat front velocity (i.e. $U_o >> w$) then the value of m rises to infinity. For most sintering mixes m is o.3 to 1.5

Between two boundary conditions say X & Y, equation (3.4) can be simplified to give a general solution for heat-regeneration.

$$ta'' = ta' + \left[ts'(1 - \beta) - ta'\right] X/a \quad \left(1 - e^{-bk\Delta x}\right) \quad \text{—3.7}$$

where $a = m - mo$ & $b = Y\, K_v / caU_o$

Thus the average temperature of air entering the combustion zone ta''_{avg} (when the zone moves from position $\Delta x_1 \rightarrow \Delta x_2$) can be calculated

$$ta''_{avg} = \frac{1}{\Delta x_2 - \Delta x_1} \int_{\Delta x_1}^{\Delta x_2} ta'' \; * \; d\Delta x \quad \text{—— 3.8}$$

integrating

$$ta''_{avg} = ta' + \left[ts' \; (1 - \beta) - ta'\right] \frac{X}{a} \left[1 + \frac{e^{-bk\Delta x_2} - e^{-bk\Delta x_1}}{bk(\Delta x_2 - \Delta x_1)}\right] \text{—3.9}$$

Knowing the quantity of air delivered to bed ($AU_o\Delta x/w$) and the quantity of air remaining in the open channels of the material $Af\Delta x$ the increase in heat content of air entering the combustion zone (heat regenerated by air) can be found by the expression

$$Qa_{reg.} = (AU_o\Delta x/w - A * f\Delta x)(ta''_{avg} - ta')\; ca \quad \text{—— 3.10}$$

Substituting 3.5, 3.6 and 3.9 in equation 3.10 Total quantity of heat regenerated by air for the whole cross-section $o - \Delta x$ can be expressed by

$$Qa_{reg.} = A*cs*\gamma s\left[ts'(1-\beta)-ta'\right] - M*\left\{\Delta x + \frac{e^{-bk\Delta x}-1}{bk}\right. \quad \text{—— 3.11}$$

and air regeneration heat between the zone $\Delta x_2 - \Delta x_1$ is

$$Qa_{reg} = Qa_{reg2} - Qa_{reg1} \quad \text{—— 3.12}$$

or

$$Qa_{reg.} = A*cs*Xs*M\left[ts'(1-\beta)-ta'\right](\Delta x_2-\Delta x_1)-(1/bk)\left(e^{-bk\Delta x_1}-e^{-bk\Delta x}\right\} \quad \text{—— 3.13}$$

and average temperature of sinter coke

$$ts_{avg} = ta' + \left\{ts'(1-\beta)-ta'\right\}\left\{1-X\left(1-\frac{e^{-k\Delta x}-1)}{bk\Delta x}\right\}\right. \quad \text{—— 3.14}$$

Note: In calculations 3.7-3.14, the quantity Δx is the thickness of finished sinter product and not the layer of initial mix.

From the knowledge of heat regeneration, heat balance and thus specific fuel consumption for a sinter quality can be established and hence targets for lowering the fuel consumption established.

A typical heat balance for a BSC Sintering plant is shown in Fig.3.2. The majority of heat for sintering comes from the coke breeze in the raw material blend, but 7.5 per cent of heat input is required to ignite the fuel in the blend using coke-oven gas.

3.3 ENERGY SAVINGS

Through improved furnace design and operating procedures effective utilisation of energy can be improved and this is summarised in a tabular form in Tables 3.1 and 3.2. The work is discussed under the headings of (a) no investment needed and (b) capital expenditure required.

A more detailed assessment of energy conservation through improved furnace design, heat treatment and waste heat recuperation is discussed below:

3.3.1 Improved Ignition Process

In the modern sinter plants the mixing and combustion of gases is accomplished before it enters the furnace. The waste gases produced are passed on to the sinter bed surface without the flame impinging the bed surface. In this way the original flame radiation ignition is replaced by convection heat transfer. The waste gases are already mixed with the necessary excess air before they enter the furnace. In conjunction with the furnace pressure control, a uniform temperature and oxygen distribution over the sinter bed surface is achieved and the air leakage is avoided. Thus the amount of waste gas produced and sucked off remain in equilibrium.

The positive advantages of these new ignition furnaces can be outlined as follows:

1. The ignition conditions adapted ensure that a sinter with good mechanical characteristics is produced thus reducing the rate of return fines.
2. The lower rate of return fines linked with improved fuel utilisation in the upper sinter mix layer provides a reduced coke consumption.
3. By adjusting the furnace atmosphere with sufficient oxygen enrichment, reducing conditions in the upper sinter mix are avoided, which offers a decrease in FeO content and hence a higher sinter strength.

3.3.2 Heat Treatment of Sinter

During the sintering of iron ores part of the solid fuel can be replaced by hot air or hot flue gases. This method of hot air sintering/mixed firing is termed as heat treatment of sinter. Some years ago heat treatment of sinter was introduced in Japan to improve the quality of sinter in the upper layer of bed[122]. Since then, the method has come to the fore-front as it offers a re-utilisation of waste gas and replacement of coke with a cheaper gaseous fuel. Extensive investigations[123-124] have demonstrated that the use of recycled hot gas/fuel gas results in a reduction of solid fuel consumption due to greater amount of heat input to sinter process in a gaseous phase. However, the heat treatment period is extended which offers an improved sinter quality.

3.3.3 Utilisation of Process Waste Heat

Currently some 33% of the heat needed for the production of sinter is discharged into the atmosphere in the form of exhaust air from sinter coolers (Fig.3.2). Recovery and utilisation of sensible heat of the exhaust air from the coolers offer a potential energy conservation scheme in the sintering process.

The idea of recycling of waste gas and exhaust air is not new in burden preparation and is an established feature in the pelletising process. However, waste heat recovery from sinter plants has been slow to find acceptance due to the difference in the agglomerating mechanism between the sintering and pelletising process.

In a sinter cooler one revolution of the trough represents a full cycle of operation from ore feed to discharge. The volume of cooling air in most cases is uniformly distributed throughout the cooling area, a major factor that makes exhaust air temperature high at the feed end and low at the discharge end. Additionally, the air temperature is influenced by the production rate of sinter i.e. heat input to the cooler. Heat recovery in the form of air preheat for ignition burners from the hottest part of cooler offer substantial energy savings.

Tanaka[125] in his work reviewing the plants built in 1976 in Japan shows that 24% savings in CO gas consumption or a reduction of 1.9 Nm^3 ($CV=4500KW/Nm^3$) of CO gas requirement per tonne of sinter can be achieved through waste heat recovery from sinter cooler. This offers an improved production capacity of 1.8 per cent.

Two schematic schemes for recylcing waste heat are shown in Fig. 3.3.

3.4. CASE STUDY: HEAT RECUPERATION AT REDCAR SINTER PLANT

3.4.1.. Objectives

The main objective of the scheme was to utilise the heat of exhaust air from the sinter cooler to preheat combustion air, thus reducing the ignition fuel requirements.

3.4.2 Scheme

In order to assess the potential heat recovery available, hot air temperature measurements around the cooler were taken and are presented in Fig.3.4

As the temperature of the preheated air affects both the coke-oven gas savings and the oxygen content of the flue gases, it is normal to utilise hot air from that part of the cooler that gives the highest temperature. At Redcar Sinter Plant approximately the first 10 per cent of the use of cooler was suggested.

With an average hot air temperature of 300°C and an average ignition temperature of 1100°C, it can be seen from Figure 3.5 that the expected savings in CO gas consumption will be around 22 per cent on the standard consumption of 145 MJ/t using recovered air from the cooler. The air/fuel ratio to the furnace can be increased with a constant furnace temperature.

This results in an increased oxygen content (11% to 13.5%) Fig. 3.6 of the flue gases, and hence increased supply of oxygen to the bed surface. Thus, it may be expected that some savings in solid fuel may be possible by more efficient combustion of the coke in the upper layer of sinter beds.

The system devised for Redcar Sinter Plant is shown in Figure 3.7 It was hoped that the expense of including a multicyclone de-duster will be avoided as the dust in the air around the cooler, and thus to the burner, will be less than 110mg/Nm^3. From the cooler the ductwork will include a tempering air inlet to avoid both overheating of the hot air transfer fan and rapid changes in air temperature and consequent furnace control. The fan itself is designed to operate at 350°C- 400°C and air volume flow is controlled via fan louvres to maintain a slight positive pressure inside the furnace. The air mains are to be lagged and the cold air fan installation to be retained in the event of maintenance requirements.

3.4.3 Financial Assessment and Justification

The justification of the scheme is based on net savings in operating cost resulting from a lower coke-oven gas consumption. Evaluation of the saving was made in terms of energy cost at the price of fuel oil, i.e. reduced coke-oven gas consumption at the sinter plant enables other plants to increase their usage of gas with corresponding savings in fuel oil.

CAPITAL COST: (Based on March 1981 quotations)

		£
	Civil Engineering work	6,000
	Buildings and Structures	43,000
	Electrical	34,000
	Piped Services	49,000
	Process Control	23,000
	Development Charges	50,000
	Mechanical Equipment	289,000
	Sub-total	494,000
	Spares	38,000
	Total Captial Cost	532,000

OPERATING COSTS	Additional costs for different designed new fan running cost	4,000 p.a
MAINTENANCE COSTS		3,000 p.a.
	TOTAL RUNNING COSTS	7,000 p.a

ENERGY SAVINGS:

Std. sinter output = 76,000 t/week

Energy consumed = 145 MJ/t sinter

Energy savings (@ 22% level) = 32 MJ/t sinter

= £338,000 p.a.

(based on 52 weeks basis and fuel prices of £2.67 per GJ - 1981)

Net Annual Savings = £338,000 - 7,000

= £331,000 p.a.

Pay back period is less than 2 years.

This scheme was accepted by the works and commissioned in February 1983. During 1982 the actual consumption of coke oven gas gradually reduced to a lowest level of 110 MJ/t just prior to commissioning the hot air system.

Since commissioning, the system has shown an average reduction of 38 MJ/t against the 1982 average of 121 MJ/t to 83 MJ/t, a reduction of 32 per cent. The air temperatures have been higher than forecast and the control temperature has been lifted from 300°C to 350°C.

If present plant output levels are maintained at 1983/84 fuel prices the savings will increase to £450,000 per year.

3.5 CONCLUSIONS

It can be concluded that through improved furnace design and operating conditions substantial energy savings can be achieved in a sinter plant. Waste heat recovery from a sinter cooler possibly offers the best potential in this field. Even a partial recovery of waste heat from sinter cooler at Redcar Steelworks offers energy savings of up to 32% with a pay back period of less than two years, at a minimum of capital cost. Energy recovery from total waste gases and exhaust air can offer savings of upto 8 per cent in the solid fuel bill.

METHODS TO IMPROVE FUEL EFFICIENCY ON A SINTER PLANT WITHOUT CAPITAL EXPENDITURE

Methods to Improve fuel Efficiency	Fuel Saving
1. Operate at deeper bed depths.	Gas and coke at Sinter Plant/ lower FeO sinter of higher reducibility will lower Blast Furnace coke rate and improve CO.
2. Reduce sinter slag volume	Improve furnace coke rate.
3. Reduction of limestone/ dolomite from blend and maximise use of other fuel bearing materials (e.g. fine iron, flue dust, mill scale).	Lower Sinter Plant coke rate.
4. Coke breeze sizing 2 - 3mm	To minimise use of heat inefficient fines.
5. Detailed examination of ore selection and burdening policy.	Avoiding use of limonitic ore or very fine ores which require high pelletising moisture saves coke breeze, whilst magnetite ores may improve fuel utilisation. Use of high sinter proportions to reduce furnace coke rate.
6. Use a consistent blend with good mixing.	Lower coke breeze rate.
7. High plant availability	Stable operation allows higher coke/gas efficiency.
8. Reduce ignition furnace pressure and minimise damper opening, inject oxygen, burner optimisation	Gas savings.
9. Raising sinter temperature from 60^{o} to $80^{o}C$.	Increase furnace top temperature and reduce coke rate.
10. Other ways of increasing sinter quality will lower fuel consumption.	Fuel improvements, both at Sinter Plant and furnace.

TABLE 3.1

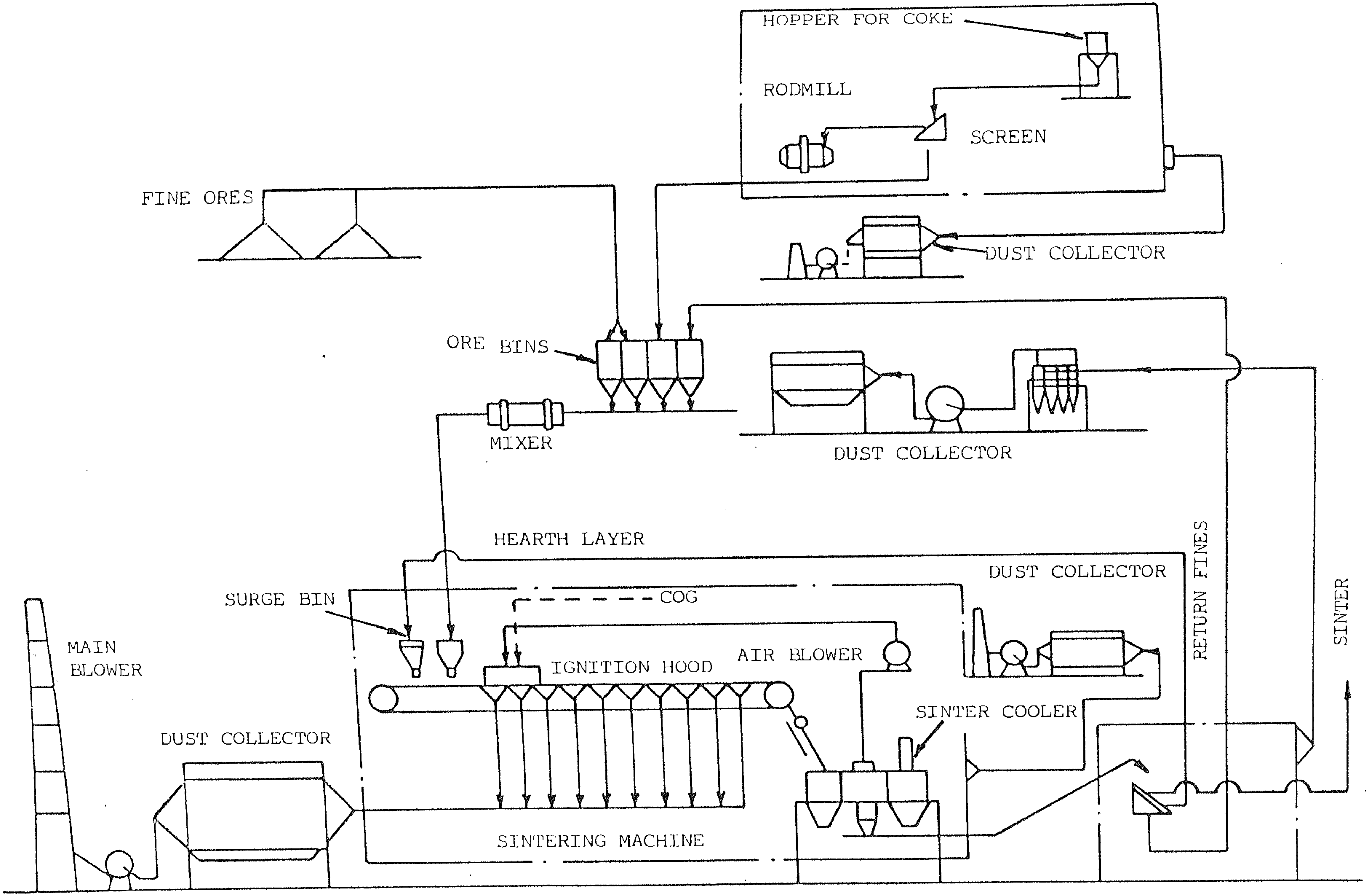
SINTER PLANT:
ENERGY SAVINGS WITHOUT CAPITAL EXPENDITURE

METHODS TO IMPROVE FUEL EFFICIENCY ON A SINTER PLANT WITH CAPITAL EXPENDITURE

Methods to Improve Fuel Efficiency	Fuel Savings
1. Lagging of waste gas main	Optimise waste gas temperature.
2. Coke segregation on strand	Coke Breeze
3. Extended ignition hood	Reduces thermal shock from furnace, improves sinter yield, decreases coke rate.
4. Use of hot air to preheat ignition combustion air.	Saves gas.
5. Hot air to preheat sinter mix	Decreases coke breeze.
6. Hot air for steam/ electricity generation.	Reduces power.
7. Additional lagging of ignition furnace.	Lower gas usage.

TABLE 3.2

SINTER PLANT:
ENERGY SAVINGS WITH CAPITAL EXPENDITURE

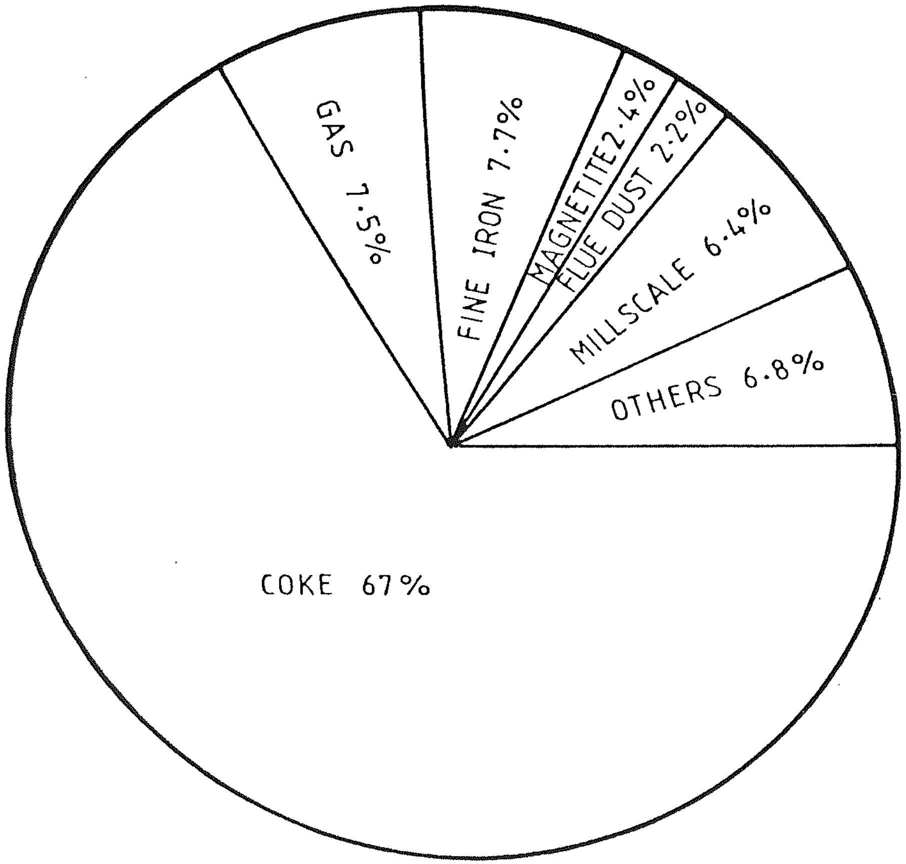


MATERIAL FLOW – SINTER PLANT

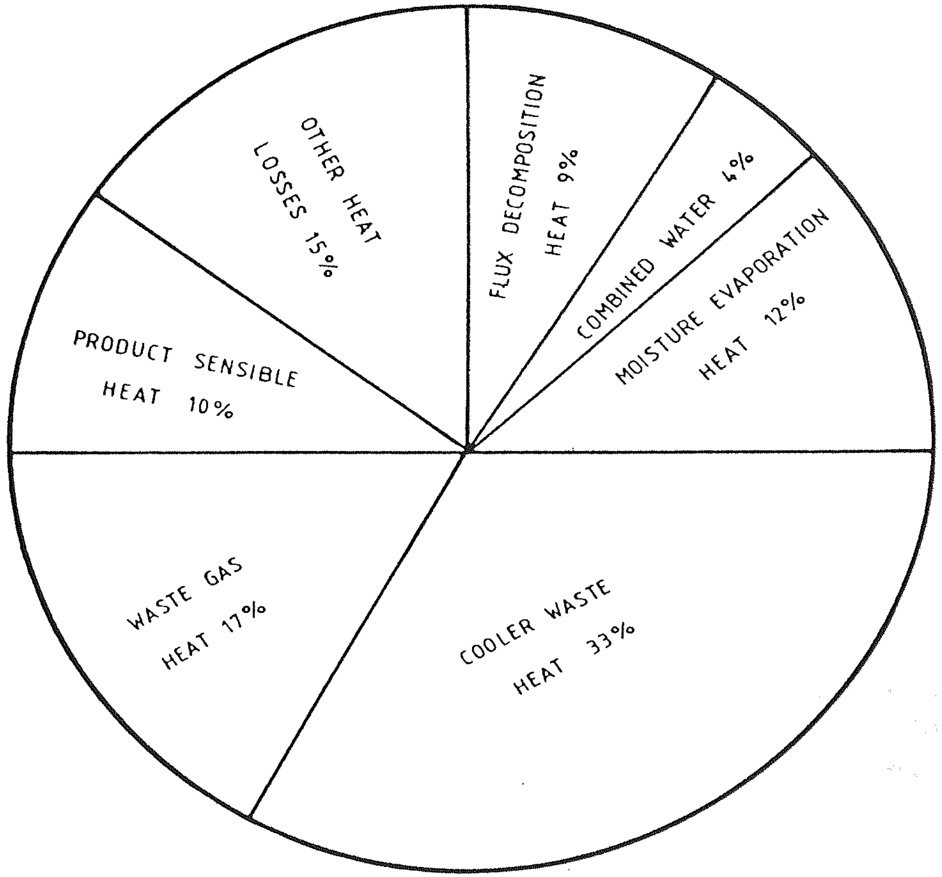
FIGURE 3.1

FIGURE 3.2

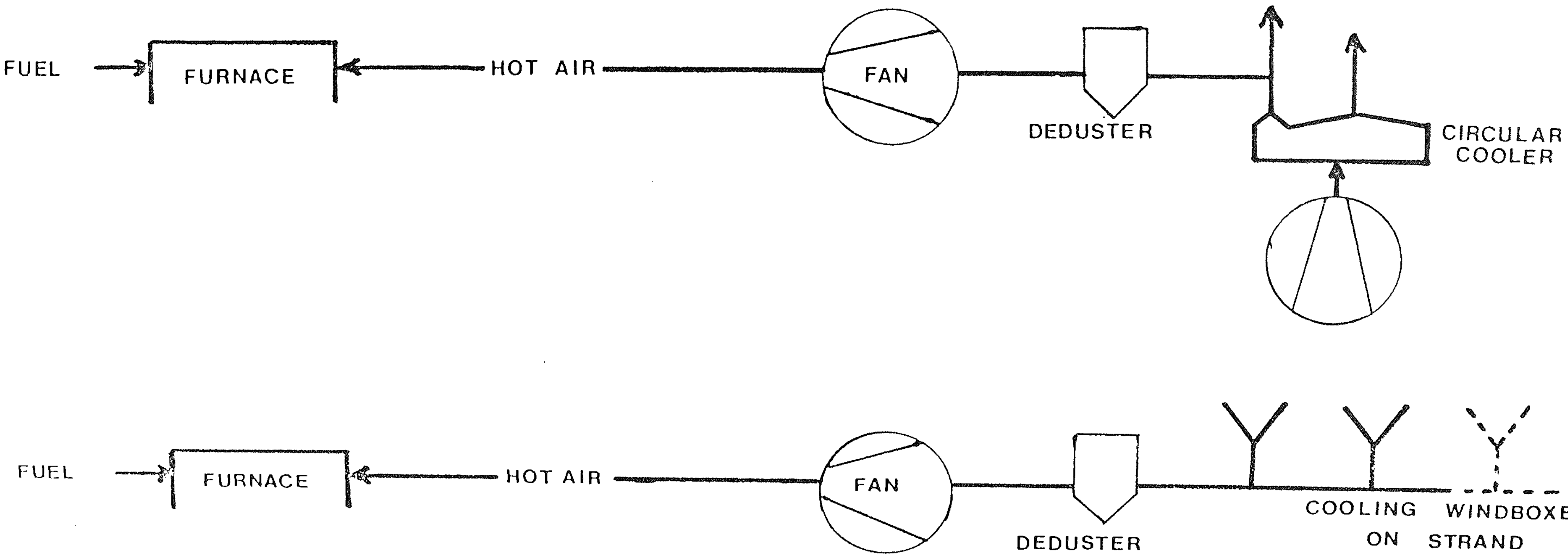
IN A SINTERING PLANT

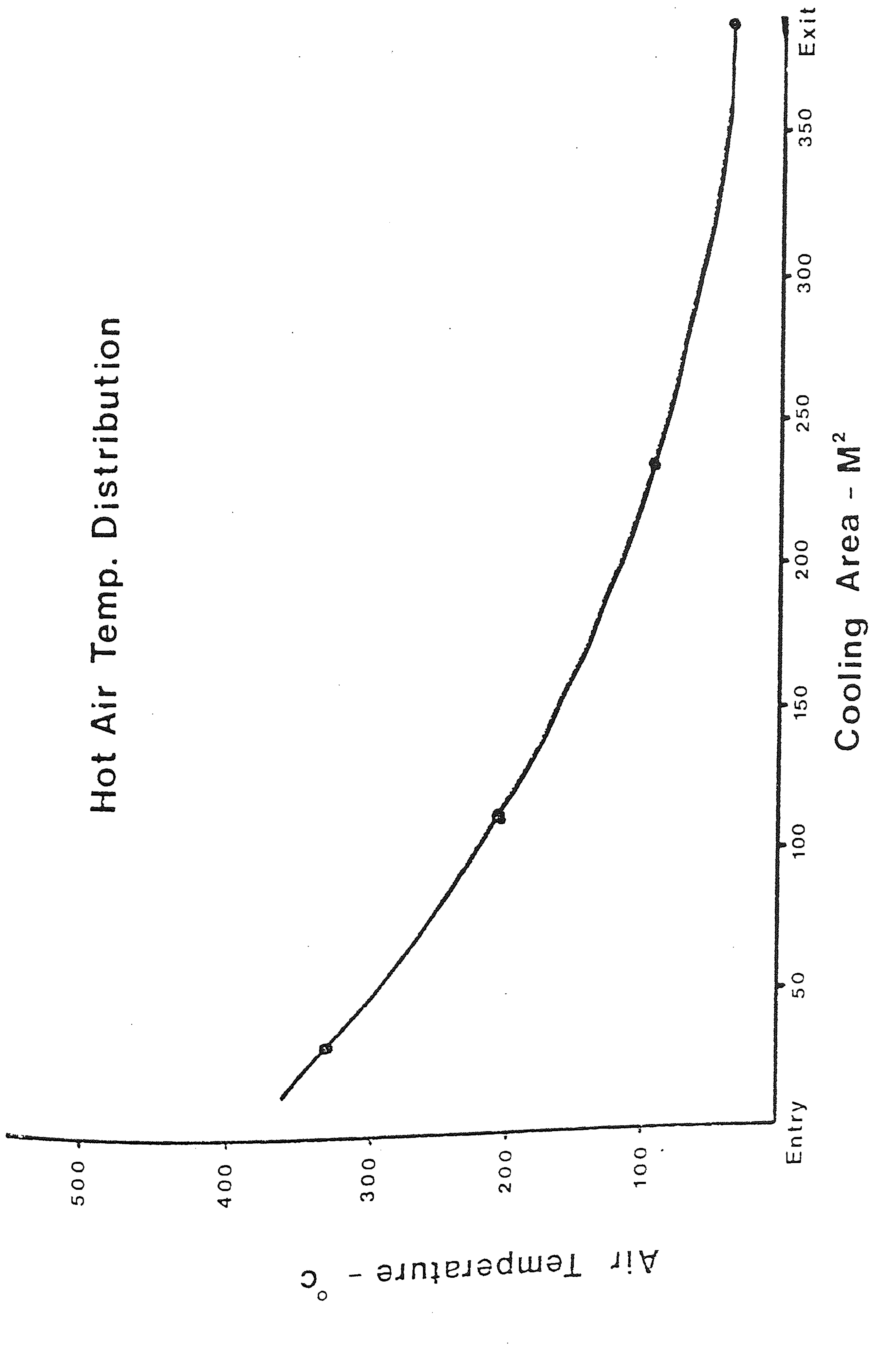


HEAT INPUT



HEAT OUTPUT (Approx.)


FUEL
FURNACE
HOT AIR
FAN
DEDUSTER
CIRCULAR COOLER
FUEL
FURNACE
HOT AIR
FAN
DEDUSTER
COOLING WINDBOXE
ON STRAND



HOT AIR TEMPERATURE DISTRIBUTION ON REDCAR SINTER PLANT

FIGURE 3.4

Fuel Saving by Air Preheating

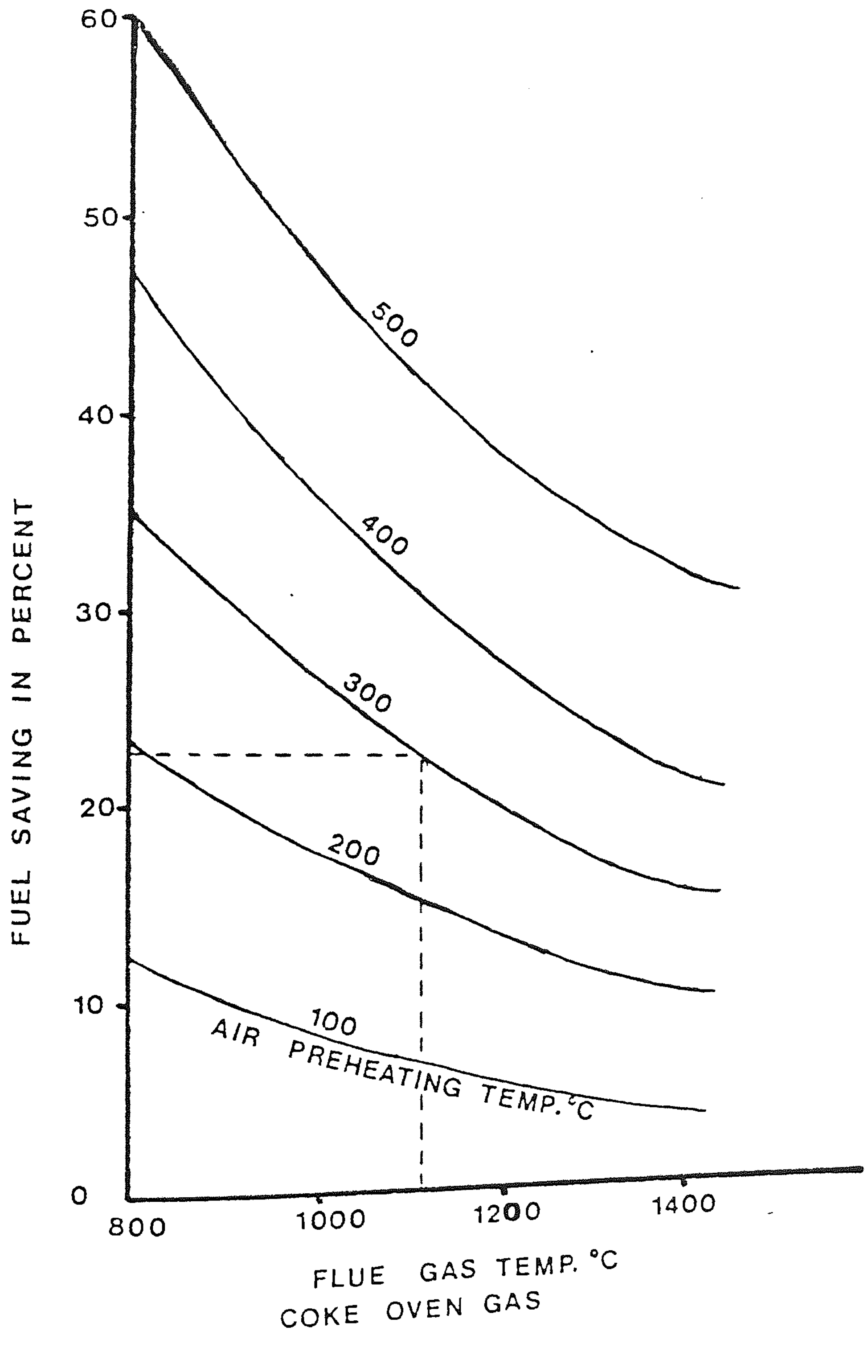
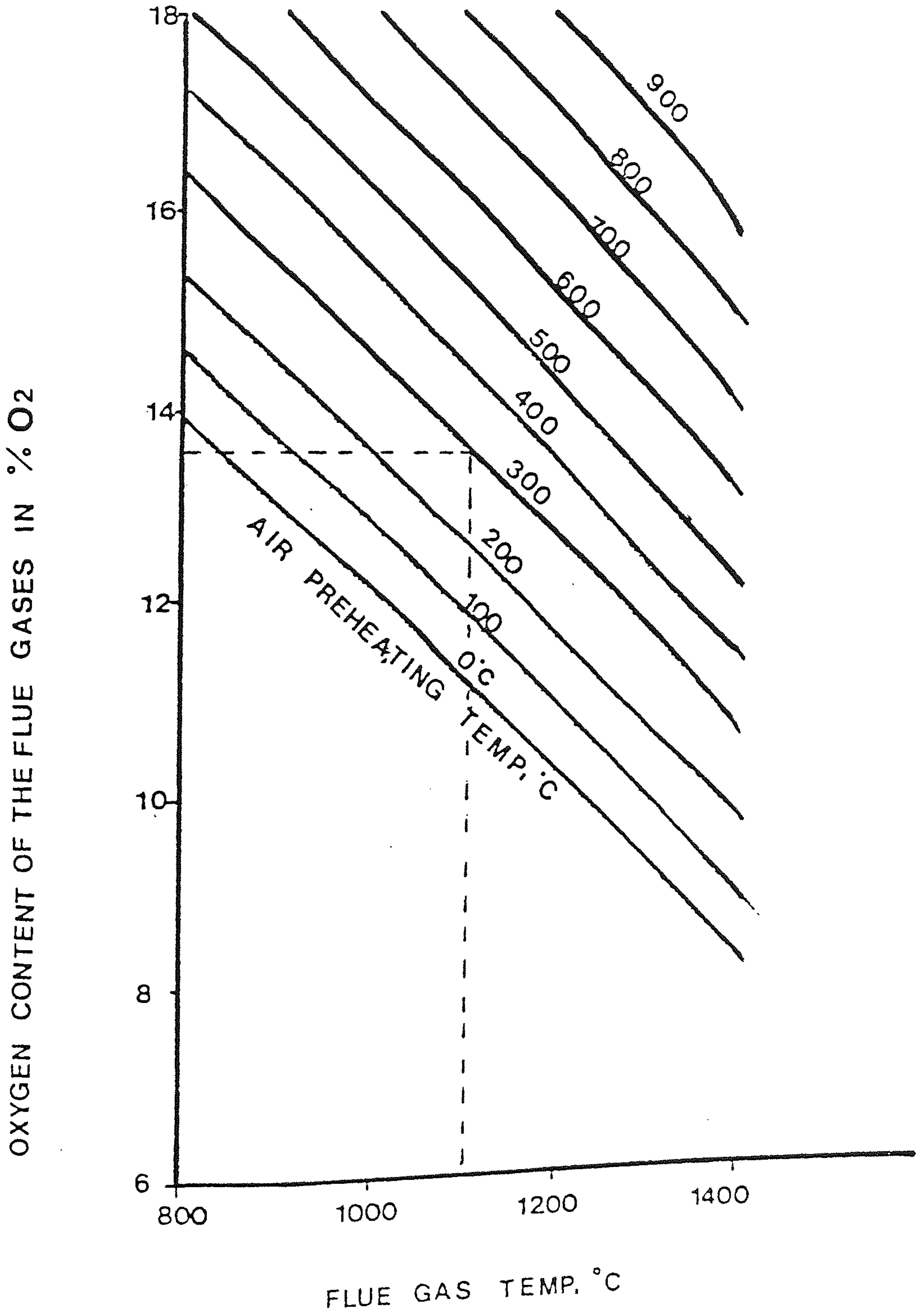


FIGURE 3.5

Oxygen content of the flue gases upon air preheating



COKE OVEN GAS

FIGURE 3.6

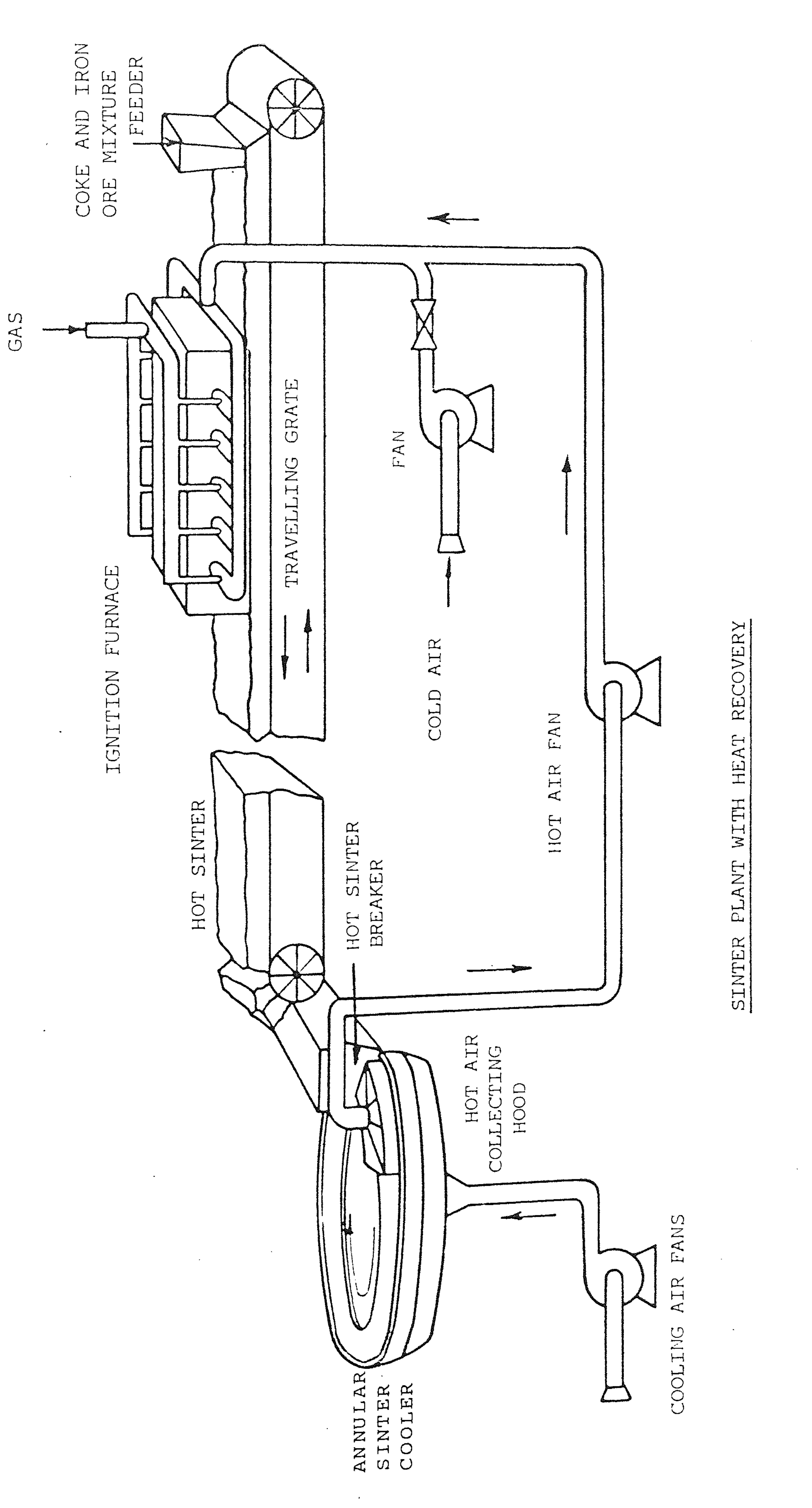


FIGURE 3.7

HEAT RECOVERY AT REDCAR SINTER PLANT

CHAPTER 4

ENERGY SAVINGS: IRONMAKING

4.1 INTRODUCTION

Blast furnace design is becoming more and more of an exact science. As the old blast furnaces become obsolete, new larger blast furnaces designed to operate at high pressure are replacing them. In the U.K. new furnaces are being designed to operate at a top pressure of 2.5 bar gauge.

In high top pressure blast furnace operation the extra compression of gases entering the furnace increase their density and mass flow rate and intensifies the physical and chemcial processes in which the gases interact with the charge materials. This allows more coke and air/oxygen to be fed to the furnace per unit time which raises the output of pig iron. It is also recognised that high pressure blast furnace operation reduces dust emission from the furnace. However, high top pressure is expensive due to the costs of compressing the blast delivered. The blowers which supply the blast handle large volumes of air at a relatively low delivery pressure and are not generally suitable for interstage cooling. The obvious approach still open is to recover some of the work expended on the compression of the blast air by means of expansion gas turbines downstream of the furnace. BSC is currently investing heavily in the new generation of blast furnace, designed to operate at high pressure. Both No.3 blast furnace at Llanwern and a 10,000t/day designed No.1 furnace at Redcar are in operation. The current state of technology existing for the recovery of energy from the top pressure of the blast furnace gases using expansion turbines is outlined in this chapter and assess its potential application in BSC works.

4.2 BACKGROUND - DEVELOPMENT OF EXPANSION TURBINES

The idea of using an expansion turbine is not new and has been investigated on a theoretical level since 1958 and even experimentally since 1962 by the two Belgian iron works, Cockerill-Ougree and Esperance-Longdoz.[126]

4.2.1 Metallurgical Advantages of High Top Pressure

From many practical observations it is well established that application of top pressure to a blast furnace gives rise to the possibility of an increased production rate. Two main factors account for this phenomenon. they are:

i) Chemical: Increasing blast furnace top pressure increases the gas residence time in the furnace which has a favourable effect on the reduction kinetics, with the possibility of decreased coke rates, which might enable increased production rates to be achieved

ii) Physical: Increasing blast furnace top pressure increases the gas density which enables a greater mass flow rate through the furnace at the same gas velocity. Since the production rate is proportional to gas mass flow rate, higher production rates are possible.

Since the effect of top pressure on chemical kinetics is not easy to quantify, only the physical factor is considered. From this viewpoint the blast furnace can be considered to have two distinct zones ie. a fixed grannular bed with associated gas flow representing the stack and a fixed bed with conterflow of liquids and gases representing the bosh.

With regard to the fixed granular bed [127-129] Eigan generates an equation of the form:

$$P_1^2 - P_2^2 = K * U_g^n \qquad \text{--- 4.1}$$

where n is 1.7 to 2.0 depending upon the Reynolds Number for the process.

For a blast furnace the gas velocity can be defined in terms of the production rate and the specific gas requirement per tonne of product.

Assuming constant bed characteristics, constant pressure drop through the bed, and constant product gas requirement then increasing the top pressure by 0.1 bar increases the product rate by between 3 to 4%.[129]

With regard to an irrigated bed with counterflow of liquids and gases, the following relationships have been defined[130-133]

$$\text{Fluid ratio} = \frac{QL}{G}\sqrt{\frac{P_g}{P_L}} \qquad \text{--- 4.2}$$

$$\text{Flooding Factor} = \frac{U_g^2 * S * \rho_g * \mu_L^{0.2}}{g * e^3 * \rho_L} \qquad \text{--- 4.3}$$

For smooth blast furnace operation Segawa[131] concludes that $(\text{Flooding Factor})^2 * \text{Fluid Ratio} \lesssim 10^{-3}$. At this limit and assuming constant bed and liquid physical properties the following relationship can be derived:-

$$\text{Production Rate} = \text{Constant} * G^{0.2} * \text{Pressure}^{0.8} / \rho_g^{0.5}$$

4.2.2 Top Pressure - Energy Recovery

Increased pressure increases the dwell time of the gases in the furnace and thus has an effect on the kinetics of reaction in a favourable way. However, the energy consumed by the blower increases very markedly with pressure, Table 4.1.[134] (compression is adiabatic). For high blast pressures it is necessary to ascertain whether multistage compression with intermediate cooling may be better than the single stage adiabatic compression. Even then the blower capital costs will remain high.

Therefore, before any economic feasibility of a high pressure turbine can be discussed the maximum theoretical energy recoverable in such turbines without combustion or cooling of the gases and the effect of gas temperature on power recovery must be considered.

The maximum electricity theoretically recoverable is given by:

$$W_G = VG_* \, Cp_* \, \eta_G \, (T_G - T_E) \qquad \text{--- 4.5}$$

where C_p = Specific heat of top gas = 1337 J/Nm^{3}K

$$T_G - T_E = \eta_{T*}T_G \left[1 - \left(\frac{p_E}{p_G}\right)^{\frac{\gamma-1}{\gamma}} \right] \qquad \text{--- 4.6}$$

where γ = 1.38

A relatively small modern furnace with a hearth diameter of 6m and a charge height of 17.5m above tuyere level giving an effective volume of 630m^3 and a total volume of 730m^3 is assumed. The furnace will produce up to 1650 t iron per day from a burden consisting of 100% sinter with a blast rate of 85,000 Nm^3/h. The top gas make will be

120,000 Nm^3/h at a pressure of up to 3.5 bar. Using the equation and assuming that the gas temperature varies between 150°C and 250°C Table 4.2. shows the calculated amounts of power recoverable assuming a turbine efficiency of 0.8. A theoretical blower consumption is presented in Table 4.3.

Table 4.4. shows the power gain, which is derived from Tables 4.2 and 4.3. Using Tables 4.2 - 4.4, Figures 4.1 and 4.2 are presented showing the maximum theoretical power gain against the top pressure of the furnace and that power recovered increases with an increase in top pressure. The figures also show that with constant pressure ratios of expansion, the power recovered increases in proportion to the absolute temperature of the gas. (Equation 4.6)

Hence the study shows that higher top pressures provide higher output of hot metal from the same blast furnace and higher gas temperatures provide a better recovery from the system.

4.2.3 Design and Operation Limitations for Expansion Turbines

Having established the theoretical basis of an expansion turbine, the design requirements needed for its installation in a steelworks are required. These are:

1. The turbine should cause a minimum of interference with blast furnace operation if any trouble arises with the turbine itself.
2. With changing operating conditions it must still be

possible for the blast furnace operator to select blast rate and top pressure independently of each other in order to obtain optimum furnace operation.

3. The turbine should be simple, requiring a minimum of attendance and maintenance.
4. The turbine should be designed for high efficiency when driven by top gases over the expected ranges of flow rate, temperature and pressure. The gas temperature will lie in the range of 50 - 150°C, depending on the recovery scheme, and the pressure will be no more than 2 - 3 bar gauge.
5. The turbine should be capable of handling wet or dry dust laden gases. It may also be advantageous if the turbine through its centrifuge action can reduce the dust content to not more than 5 mg/Nm^3.
6. The turbine should be free from erosion and corrosion by the gases and carried matter. The design should either be inherently free of deposition of carried matter or be easily cleared of such matter.
7. Although it may be simple to link the turbine to a generator so that it can operate at a constant speed, it is better to link it directly to the blower to achieve better efficiency but this is only practicable in certain cases.

4.2.4 The Development of Expansion Turbines

After numerous experiments, theoretical investigations and detailed tests which, in the Soviet Union started in the early fifties and in Belgium were commenced around 1958, two alternative schemes emerged for incorporating an expansion turbine in blast furnace plant:

1. The turbine can be run on hot gas which has undergone only a coarse dry cleaning process (Fig.4.3)
   Unfortunately this type of turbine would be most exposed to erosion and/or clogging of the flow channels. But if some way is found to overcome these problems, this type has distinct advantages - mainly a relatively high power output per unit of gas throughput.

2. The turbine can operate on gas that has been subjected to wet scrubbing, but then needs either heating or a deliberate rise in the initial gas temperature to prevent condensation and deposition. The heating may be accomplished by partial combustion of the gas ahead of the turbine. (Fig.4.4)
   However, reheating the gas requires extra energy and thus reduces the overall efficiency of the process.
   The Soviet Union have developed their GUBT turbine on this principle.

Over the past few years, SOFRAIR of France developed a variation of the first of these schemes. The turbine in this case is interposed between the primary and secondary scrubbing and handles all the blast furnace gas generated in the plant. The first such experimental single stage turbine tried at the Hayange plant, yielded a mere 90 kW for a throughput of 10,000 Nm^3/hour gas flow. Tests at the Longdoz plant, run on dry and coarsely cleaned blast furnace gas, extracted 280 kW for the same throughput. Although both the turbines were centripetal, the difference between their outputs is largely explained by the fact that the turbine at Hayange was a single stage type whilst the turbine at Longdoz was a more efficient two stage unit.

SOFRAIR calculations show [135] that a two stage centripetal turbine placed between the primary and secondary scrubbing stages should yield 30 kW of electricity from a gas throughput of 1000 Nm^3/h with an inlet temperature of 80°C, as against the 33 kW available from a similar turbine receiving the same amount of dry hot gas at 150°C. It is worth noting that SOFRAIR, who are part of the NEV group, previously experimented with a turbine running on dry coarsely cleaned gas. The wet design was adopted because in most of the modern furnaces, top gas temperatures could be as low and even lower than 150°C.

Results from these pilot plant studies showed that the turbine suffered little or no abrasion, but clogging of the flow channels with deposits was a problem.

However, their latest plant designs show that this problem has now been overcome by the injection of water at various points in the vicinity of the impeller.

The position of the dirt collectors is determined from the formula [136]

$$df = \frac{Vr^2 * L * Ds * Kr}{Vt^2 * \pi * 4} \qquad 4.7$$

where df is the limiting dimension of residual particles which in turn decides the limiting size of particles removed.

In a saturated blast furnace gas application the volute is designed to remove particles greater than 3 - 4 μm size from the inlet gas. Test results from Mizushima [137] have indicated better than design performance i.e. with inlet dust loading of 15 mg/ m^3 the turbine outlet measurements have been 1 - 2 mg/ m^3 against a design value of 5 mg/ m^3.

An example of the second of the two schemes is the GUBT turbine. Here, blast furnace gas is heated to 120 - 140^{o}C in a gas heater before entering the turbine. The heating is effected by means of burning a portion of the gas in the gas heater burning zone and subsequently mixing the products with the balance gas. The relatively high temperature of gas thus prevents clogging of the turbine with wet ash.

Following successful applications of the two turbine systems, Japan for the past few years has been developing axial wet turbines in order to achieve maximum power recovery but still retain the advantages of a wet system.

4.3 EXPANSION TURBINES PRESENTLY AVAILABLE

The thesis considers in detail the design features of each type of the turbines presently commercially available

4.3.1 SOFRAIR Expansion Turbine

The principal features of the current SOFRAIR turbine are:

1. Radial fluid inflow within a colute casing which promotes spearation of particles and water droplets suspended in the gas.
2. Swan neck collectors for the removal of material impinging on the volute casing.
3. Volute profile designed to act as a vaneless nozzle to minimise impeller abrasion.

4. Water injection employed for continuous or intermittent flushing of impeller casing.

5. Dynamic pressure water seals to prevent any gas leakage

A radial flow turbine can be run either wet or dry. The main advantages of the wet type are:

(i) The gas is not burned thus the heating value of the gas remains unaltered.

(ii) Existing gas cleaning equipment can be used without modification.

(iii) Without gas combustion, control is simpler.

(iv) The permissible quantity of dust and moisture is greater than in the dry type.

(v) Changes in gas temperature and heating values only affect the turbine output, not its functioning.

(vi) Maintenance and adjustment are simple.

To assess the design and performance of a SOFRAIR turbine the example of Mizushima No.2 is discussed, where first hand information was available.[138]

Fig. 4.5 shows a general view of a SOFRAIR turbine as installed at Mizushima steel plant. In order to ensure that the turbine does not affect the performance and operation of the blast furnace the turbine is located beyond the venturi scrubber (Fig. 4.6) Since the furnace top pressure greatly influences the blast furnace operation, the conventional septum valve control is also used. The turbine receives 75% constant gas flow while the other 25%

necessary for top pressure control flows through the septum valve.

Fig. 4.7 shows a pictorial view of the turbine rotor while Fig. 4.8 shows a sectional drawing of the assembled turbine. The gas flows from the circumference of the impeller towards the rotor axis before being discharged to the first exhaust chamber. The gas then passes through the intermediate duct which connects the first stage exhaust chamber to the second stage impeller, thence flowing out from the second stage exhaust chamber.

The rotating parts and surrounding area are designed to provide clearances as large as possible (Fig.4.9). Vaneless nozzles prevent clogging and wear from the dust.

Mizushima claim that in order to conserve energy, water is injected intermittently to remove accumulated dust without any adverse affect on the turbine operation. The water injection system is shown in Fig. 4.10.

The gas flow in the casing forms a free vortex and goes into an impeller. Dust and moisture in the gas collect on the inner walls of the casing, being separated from the gas by centrifugal force and are removed by the swan neck utilising centrifugal force and gravity. The dust and moisture thus pass through a seal tank, settling tank and then are discharged to the outside.

The turbine shaft sealing is achieved by a disc type seal and carbon packing. The disc is fixed to the turbine shaft and when it rotates, water is sealed between the disc and the casing, its pressure opposing the gas pressure. The seal is maintained as long as the turbine shaft rotates and the sealing water is continuously supplied. As soon as the turbine stops rotating, the seal is lost.

The electric control system is generally fitted with safety devices to eliminate any abnormality in the power system and to proctect the turbine/ generator. Since the paramount objective is the production of hot metal, the turbine/ generator is of secondary importance. Thus the safety devices have to be highly reliable in order not to adversely influence the blast furnace operation.

It is general to provide two safety systems i.e. the emergency trip device and alarm system. When the emergency trip is activated the emergency stop valve closes, stopping the turbine. If the emergency stop valve is fully closed, an electrical signal opens the circuit breaker and the turbine is released from parallel operation.

As the trip device operates, an alarm rings both in the electrical room and blast furnace control centre. The operation of the safety devices is completed within a few seconds.

Fig. 4.11 shows a typical safety block diagram.

A. SOFRAIR POWER RECOVERY

The SOFRAIR turbine can be designed to operate at:

(a) Constant Power Recovery (Fig.4.12) where blast furnace gas passes through two stage gas cleaning and then partly through a septum valve and partly through a recovery turbine arranged in parallel. Normally the septum valve is almost closed and most of the gas passes through the recovery turbine. Sudden shut-down of the turbo-alternator requires a valve sequence which results in a momentary increase of about 5% of the furnace top pressure until the control system opens the septum valve to get back to the desired value of furnace top pressure.

This is the most common mode designed and has been used at Mizushima steel works.

(b) Maximum Power Recovery using septum valve

In this the whole gas flow is normally passed through the turbine. The parallel septum valve leg is used only when power is not being generated. (Fig.4.13).

This system recognises that in order to recover the maximim power generation, a variable recovery rate dipping to a minimum every ten minutes or so must be accepted. In the Japanese steel industry where the power stations are partly or wholly owned by steelworks, the power recovered from the system is fed back to the grid and purchased from the grid as required with no financial penalty. Therefore the dip in power recovery has little effect on the total grid production and because of the power structure it encourages the Japanese steel industry to recover maximum power wherever possible.

(c) Power Recovery with variable scrubber pressure differential

This is the type of system most likely to be applicable to the BSC blast furnaces where Bischoff twin stage scrubbers are used without septum valves.(Fig.4.14).

BISCHOFF - SOFRAIR SYSTEM

Both the high top pressure designed blast furnaces in the BSC (i.e. Llanwern No.3 and Redcar No.1) have a Bischoff gas cleaning system as opposed to venturi scrubber linked with a septum valve.

A visit to Ruhrort Steel Plant and discussions with Bischoff [139] who are currently marketing the SOFRAIR turbine in Europe, showed that the application of 'SOFRAIR' turbine, where a Bischoff cleaning system exists, can be carried out quite successfully.

In the Bischoff gas cleaning system, a dust catcher is provided to collect the coarse particulates by a dry mechanical method. Wet mechanical separation of dust particles takes place in primary and secondary scrubbers both of which are located in a single space-saving vertical housing. Results show that a pressure drop of only 5 mbar occurs in the Bischoff primary scrubber thus giving little wastage in the pressure differential required for power generation through the expansion turbine.

In the Bischoff-Sofrair system the adjustable annular gap scrubber controls the furnace top pressure and, when the turbine is not operating serves as a low noise throttling element. The plant is therefore arranged from the outset as if a turbine did not exist. The advantage of such a system is that even when the turbine is operating, the annular gap scrubber continues to control the furnace top pressure regardless of what happens downstream, the turbine runs without being adjusted and without any control function. Therefore if the turbine breaks down blast furnace operation remains unaffected.

The turbine inlet control flap is only used for start up speed control. When the synchronous speed is reached, the generator can be connected to the electrical network which then keeps the turbine speed constant. After the start up period the inlet control flap remains fully open.

Different operating conditions in terms of a smaller throughput volume which do not match the turbine capacity result in a change of furnace top pressure which is, however, immediately compensated by the annular gap scrubber until the pressure at the turbine inlet matches the turbine characteristics again. If an operational state deviating from the normal blast furnace operation arises which lies below the turbine characteristics with too low a pressure and too great a gas volume then a bypass control is activated. If the turbine output has become too small for a certain period to produce electrical energy the turbine is automatically switched off by a disturbance program. This is necessary after a period of 200 seconds beacuse otherwise the heat generated in the turbine becomes inadmissibly high.

For safety reasons there is also a quick-activating valve located at the turbine inlet which cuts off the gas flow to the turbine within 2 seconds. At the same time the bypass valve is opened. This immediately prevents overspeeding in the event of a sudden disturbance.

A list of SOFRAIR turbines currently operating worldwide is presented below.

SOFRAIR TURBINE INSTALLATIONS

Client	Plant Location	Furnace	Recovery Unit	Service Date
1. Kawasaki Steel Corporation	Mizushima	B.F. No.2 Hearth diameter 11.5cm Volume 2857	8000 KW	August 1974
2. Nippon Kokan	Fukuyama	B.F. No.4 Hearth diameter 13.8m Volume 4197 cu.m.	9500 KW	February 1976
3. Kawasaki Steel Corporation	Mizushima	B.F.No.4 Hearth diameter 14.4m Volume 4323 cu.m.	13000 KW	October 1976
4. Nisshin Steel Company	Kure	B.F. No. 4 Hearth Diameter 8.9m Volume N.A.	9500 KW	December 1976
5. Societe Usinor (Now closed with plant closure)	Thionville	B.F. No.1 Hearth diameter 8m Volume N.A.	4000 KW	December 1976
6. Kawasaki Steel Corporation	Mizushima	B.F. No.3 Hearth diameter 12.4m Volume 3363 cu.m.	9500 KW	December 1978
7. August Thyssen-Hutte	Ruhrort	B.F. No.6 Hearth diameter 11m Volume 2226 cu.m.	7500 KW	October 1978
8. Kobe Steel	Kakogawa	B.F. NO.3 Hearth diameter 14.m Volume 4300 cu. m	13500 KW	April 1978
9. Sumitomo Metal Industries	Kashima	B.F. No.5 Hearth diameter 14.m Volume N.A.	12500 KW	April 1976
10. Ka asaki Steel Corporation	Mizushima	B.F. No.1 Hearth diameter 11.m	7500 KW	December 1978
11 Pohang Iron & Steel	Pohang	B.F.No. 3 Hearth diameter 13.2m Volume 3850 cu.m	11700 KW	June 1978
12 Pohang Iron & Steel	Pohang	B.F. No. 4 Hearth diameter 14.m	11700 KW	December 1980
13 Inland Steel	Harbour Works	B.F. No. 7	15000 KW	December 1980

4.3.2 GUBT Expansion Turbine

The basic principle of energy recovery in this sytem is similar to that used in the SOFRAIR expansion turbine, i.e. the high pressure gases from the top of the blast furnace are passed through a turbine, converting potential energy in gases to mechanical energy which is either utilised directly by linking the turbine to the blower or used for power generation by coupling the turbine to a generator.

However, the GUBT turbine is a dry type axial turbine which requires dry gases with a dust content of less than 10 mg/Nm^3 (the Kawasaki-GUBT installation operates with a dust content of 5 mg/Nm^3). Since, after scrubbing, the gases are wet and cool, a heater is installed in the line where up to 5% of the total blast furnace gas is burned and then mixed with the rest of the gases providing a dry gas at 120 - 140^{0}C. To avoid any explosion hazards (due to oxygen presence) good instrumentation is needed. To achieve the dust content level of 5 mg/Nm^3, the gases have to be passed not only through a scrubber but also through a precipitator system.

Fig. 4.15 shows a line diagram of the GUBT turbine system where gases after cleaning are passed through the heater before passing though the turbine. Use of dry, clean gases minimises the problems of corrosion and erosion. But compared with the SOFRAIR system it does need extra hardware in the form of cleaning and heating equipment thus making it more complicated. Because of higher gas temperatures at the inlet a better thermal recovery may be achieved.

Fig. 4.16 shows the actual construction of a GUBT turbine. It is a two stage axial flow turbine with axial entry and exhaust at 55° to the turbine axis. The turbine casing is made of carbon steel. The rotor is a rigid type with a critical speed of 4200 rpm and is made of chromium stainless steel. The through hole in the turbine shaft serves as a suction channel for the two fans pushing air/ nitrogen to the end of the seal chamber. The nozzle rings of both stages are cased in one unit. The first stage nozzles have adjustable blades.

The gas turbines are designed for a maximum power generation of 12 MW. Any higher ratings are provided by more than one turbine.

Knowing the gas flow rate and top pressure of the gas the recovery can be calculated from Fig.4.17(140) assuming an inlet gas temperature of 120°C and a back pressure of 1.15 bar.

A turbine linked directly to the blower can provide up to 30% of the total energy required by the blower.

As with the SOFRAIR turbine system it is recognised that fail-safe devices which would trip in case of turbine/ generator failure are essential so that in no way is the performance of the blast furnace affected.

The operational experience of GUBT turbines at Cherepovets Metalworks, USSR(141) show that the two turbines installed have operated quite satisfactorily for the past ten years. They accepted that considerable problems were experienced during the initial commissioning trials due to the build up of dust deposits on the guide blades reducing the gas passage by as much as 70% at one time. In addition high abrasion of blades due to hot dry gases was also experienced. However, these problems have now been overcome by operating the turbine at a dust loading of less than 10 mg/Nm^3. The turbines now operate trouble free and are only checked every eighteen months.

Kawasaki steelworks (138) is currently operating both the SOFRAIR and GUBT turbine systems. After the initial successful application of a SOFRAIR turbine, Kawasaki in an exchange deal with the USSR installed the GUBT turbine.

Although satisfied with its performance, their major criticism of the GUBT turbine is that the system is over engineered and requires as much as 5% of the total BF gas for pre-heating. In addition the turbine size is limited (12MW) and has not been developed to permit installation on bigger furnaces as a single turbine system. Their recent test trials, over a period of a few months, showed that the turbine can be operated quite satisfactorily even without heating the gases i.e. operates with wet gases.

A list of GUBT Turbines Installations is given below.

No. off	Location	Design Rating	Year
2	Cherepovets - USSR	12 MW	1968/69
1	Krworojskiy - USSR	8 MW	1971
3	Kawasaki - Japan	12 MW	1975/76

4.3.3 Wet-Axial Turbine

The main difficulty in using a wet axial flow turbine lies in that the sticky dust contained in the wet blast furnace gas easily causes passage obstruction.

However, with their operational experience of both SOFRAIR (wet, Radial) and GUBT (Dry axial) turbines, the Japanese industry with its need to recover maximum energy has now developed a wet-axial turbine. The development has been concurrently carried out by MITSUI, KAWASAKI and HITACHI.

The basic features of all these systems are similar. The inlet casing is designed to have a cyclone effect, and large dust particles are thrown away from the turbine by centrifugal force. There are several water jet nozzles in the inlet casing of the turbine and on the periphery of the rotor disc. Any dust deposited on the turbine blades can be continuously washed away by these internal water injections. Therefore, similar to SOFRAIR, the turbine acts as a dust removal system.

Fig. 4.18 shows a three stage MITSUI turbine. The casing is horizontally split and is made of steel and cast iron. It is claimed[142] that the smooth inside wall of the casing prevents the deposit of any dust. The turbine is equipped with a water injection cleaning system. Dust containing moisture condensed in the turbine is discharged to the outside through a circumferential slip on the inside of the casing. The bearing pedestal is supported by the turbine casing and can be overhauled separately. Labrinth nitrogen and oil film seals are employed at the same time. The nitrogen gas acts as a buffer and prevents clogging and sulphur oxide contamination.

Fig. 4.19 shows that using variable angle nozzles a wet/axial turbine can be operated at low pressures for power recovery.[143] Power recovery using such a multistage turbine, is presented in Table 4.5.

The turbine is axial and thus compact when compared with the SOFRAIR turbine and eliminates any need for preheating the gas as in the GUBT turbine and thus should be a little cheaper than the other two systems.[144]

The manufacturers also claim that the system in addition to being able to operate at lower pressures offers a 10% increase in power recovery when compared to SOFRAIR systems for 'total energy recovery state'.

However as most of the installations are currently being commissioned, actual data on the turbine performance is not yet available.

A list of Wet-axial Top Pressure Turbine installations is given below:

No. Off	Client	Plant Location	Turbine Type	Recovery Rating	Service Date
3	Nippon Steel Corp.	B.F. No.1 Sakai	Mitsui TRT	14,000 kW	1980
		B.F. No.4 Yawata	Mitsui TRT	12,500 kW	1980
		B.F. No.1 Nagoya	Mitsui TRT	16,000 kW	1980
1	Kawasaki	B.F. No.5 Chiba	Kawasaki	9,200 kW	1979
3	Sumitomo	Wakayama	Hitachi	7,000 kW	1979/80
1	Kobel Steel	B.F. No.3 Kobe	Hitachi	7,000 KW	1981
1	Hoogovens	B.F. No.7 Ijmniden	TRT	14,000 KW	1981

4.3.4 Comparative Assessment

Due to insufficient data available on the wet-axial turbine at present, a comparison between GUBT and SOFRAIR turbines is carried out.

	SOFRAIR	GUBT
Mode:	Radial Turbine	Axial Turbine
Type:	Wet type turbine	Dry type turbine
Acceptable Dust loading:	15 mg/Nm^3	5 mg/Nm^3 - secondary cleaning essential
Dust cleaning:	System reduces the outlet dust loading to 5 mg/Nm^3 thus acting as a cleaning system.	Does not clean the gas further.
Prevention of Clogging:	Prevention of dust in the turbine impeller because of radial flow.	Clogging could be expected but dust content at inlet is low.
Prevention of Abrasion:	Low turbine peripheral speed and relative low gas speed are used to reduce abrasion of the impeller blades.	
Gas Temp:	No gas heating needed for prevention of dust and mist.	Gas heater needed for prevention of dust and mist.
Combustion:	Simple control system	Sophisticated instruments are needed, especially to guard against explosion risks in B.F. gas system.
Noise:	Reduces noise level	Reduces noise level
Economic Evaluation:	True economic information readily available.	Economic assessment difficult to carry out to Western standards.

4.4. CASE STUDY: APPLICATION OF TOP PRESSURE TURBINE IN BSC

Wide industrial experience is available both on SOFRAIR and GUBT turbine operation. Therefore transfer of this technology can be carried out on any high top pressure blast furnace without much problem.

Unfortunately study[145] shows that only two new generation furnaces such as Llanwern No.3 and Redcar No.1 are designed to operate at high pressures. The other BSC furnaces, designed to operate at modest pressures of 1.0 to 1.5 bar, are Port Talbot Nos. 4 and 5 and Ravenscraig Nos. 1 and 3 but none of these are operating at the designed pressure.

Ravenscraig No.1 or No. 3 with its Paul Wurth system could be used to demonstrate a pilot plant scheme but, in general, it should be accepted that the application of such a turbine can only be successfully carried out either at Llanwern No.3 or Redcar No.1 blast furnace.

Ever since commissioning, Llanwern No.3 furnace has had leakage problems even when operating below design pressure[146-147]. Until all leakage problems are resolved and the furnace is able to operate at its designed pressure of 2.5 bar, application of a top pressure turbine cannot be initiated at Llanwern No.3 furnace. Thus, Redcar No. 1 blast furnace where the commissioning has been quite successfully completed, offers the most suitable furnace within the BSC for a top pressure turbine installation at this time.

A detailed economic assessment for the application of a top pressure turbine on Redcar No.1 furnace is presented below. Corresponding assessments for Llanwern No.3 and a pilot plant are presented in Table. 4.6

4.4.1 Economic Assessment of Redcar No.1

Note:

1. All quotations for capital costs have been updated to 1980 using BSC indices.
2. Fuel costs are based on average BSC prices,

i.e. Electricity	=	£0.027 per kWh.
B.F. Gas	=	£2.47 per GJ
Compressed Air	=	£0.004 per Nm^3

Blast Furnace Data - Redcar No.1

Assumptions:

Hearth Diameter	=	14 m
Top gas volume	=	620,000 Nm^3/h
Top gas/blast ratio	=	1.4
Top pressure	=	2.5 bar
Mains pressure	=	0.1 bar
Gas Conditions at Turbine Inlet		
Pressure	=	2.25 bar
Temperature	=	$53^{o}C$
Dust Loading	=	10 mg/Nm^3
	=	1.384 for saturated gas
Cp	=	1337 J/Nm^3 K (dry)
Turbine Polytropic Efficiency η_p	=	0.70
Electrical Efficiency η_e	=	0.97
Mechanical Losses, Lm	=	40 kW
Constant Power Recovery	=	0.75 x Max Power recovered.

(where 75% gas passes through turbine)

$$\text{Maximum Power Recovered} = \eta_e \cdot Cp.\ V.T_1 \left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] - Lm$$

where V = Volume Flow Rate

$$\frac{k-1}{k} = \eta_p . \frac{\gamma-1}{\gamma}$$

$$= \frac{0.7 \quad (1.384-1)}{1.384}$$

$$= 0.1942$$

where γ = 1.384

Cost Appraisal - SOFRAIR Turbine

Maximum Power Recovered	=	13.795 MW
Constant Power Recovered	=	10 MW

This agrees with a Chemico estiamte [148] that a maximum power recovery of 14 MW can be achieved on a 14 m hearth diameter, 10,000 tpd blast furnace using a single turbine.

Capital Cost of SOFRAIR Turbine	=	£3.0 million
Cost of Foundation of Buildings and Services (estimated @ 15% of capital cost)	=	£0.45 million
∴ Total Capital Cost	=	£3.45 million
Labour cost: based on 1 man/shift*	=	£32,000 p.a.
Maintenance: based on 3% per annum of installed capital cost	=	£104,000 p.a.
Supplies of Services: estimated	=	£20,000 p.a.
Total Running Cost	=	£156,000 p.a.

* Mitzushima have shown [137-138] that the expansion turbine can be operated without any additional manpower requirements.

(a) Finacial benefits from maximum electricity generation (14 MW)

$= 14,000 \times 8250 \times 0.027 = £3.12$ million per annum

$\therefore$ Net Benefit = £2.96 million per annum.

(b) Financial benefits from Constant Power Recovery (10 MW)

$= 10,000 \times 8250 \times 0.027 = £2.23$ million per annum

$\therefore$ Net Benefit = £2.07 million per annum.

Investment appraisal indicates that the installation of a SOFRAIR turbine providing constant power recovery on a 14 m hearth diameter, 10,000 tpd Redcar No.1 blast furnace would yield a net return of £2.07 million per annum of total investment cost of £3.45 million with a pay back period of 1.67 years. For maximum power recovery the net return can be increased to £2.96 million per annum giving a pay back period of 1.17 years.

Cost Appraisal - GUBT Turbine

It is assumed that all parameters remain the same as for a SOFRAIR turbine except that the gas temperature is $140^{o}C$ and the dust content is 5 mg/Nm^3.

Maximum Power Recoverable = 17.5 MW

$\therefore$ Constant Power Recovery = 13.0 MW

Energomachexport [149] advise that two 9 MW GUBT units should be installed on a 14 m hearth diameter 10,000 tpd blast furnace and the plant availability should be 8250 hours per annum at full power output (94.2% utilisation factor).

Capital Cost of GUBT Turbine 25*	=	£2.56 million
(Assuming exchange rate of US $2.3 per £1)		
Cost of Foundations, Buildings and Services	=	£0.38 million
(Energomachexport suggest 15% of capital cost)		
Total Capital Cost	=	£2.94 million

Energomachexport advise that 3.6 per cent of the blast furnace gas would be consumed in the preheater although Japanese experience shows that it could be between 4 - 5 per cent.

Assuming Calorific Value of Gas	=	3.516 MJ/Nm^3 = (840 kCal/Nm^3)
Gas Burnt	=	620,000 x 0.036 x 8250
	=	184.140 million Nm^3/year
Financial Penalty	=	184.140 x 3.5169 x 0.00247
	=	£1,599,600 per annum

Energomachexport advise that a compressed air flow of 12,000 Nm^3/h would be required for each of the turbines.

Air Requirement	=	2 x 12,200 x 8250
	=	201.3 x 10^6 Nm^3 per annum
Air Cost	=	£201.3 x 10^6 x .004
	=	£0.80 million per annum

Labour Cost: 5 men per annum @ £8000/man	=	£40,000 per annum
Supplies & Services: Estimated	=	£20,000 per annum

* This price seems low and realistically should be very similar to SOFRAIR costs.

Maintenance: based on 3 per cent per annum of

Capital cost = £88,000 per annum.

Total Running cost = £2.25 million per annum

(a) Financial Benefit providing maximum power recovery

= 17500 x 8250 x 0.027

= £3.90 million per annum

Net Benefit per annum = £1.35 million per annum

(b) Financial Benefit of GUBT Turbine, providing Constant

Power Recovery = 13.0 x 8250 x 0.027

= £2.90 million per annum

Running Cost = $0.75 \times 1.6 \times 10^6 + 0.8 \times 10^6$
$+ 20{,}000 + 88{,}000$
= £2.15 million p.a.

Net Benefit per annum = £0.75 million per annum

Thus investment appraisal indicates that the installation of 2 x 9 MW GUBT turbines on a 14 m hearth diameter 10,000 tpd blast furnace would yield, for maximum power recovery, a net return of £1.35 million per year on a capital investment of £2.94 million. This provides a pay back period of 2.2 years.

However with a constant power recovery, a net return of £0.7 million for year is achieved with a pay back period of about four years.

In all these economic appraisals, price of electricity plays a very important role.

Assuming an electricity price of £0.02/kwh as per Teesside Division, with all other parameters remaining the same, a comparative appraisal is illustrated in Table. 4.6

It can be seen that savings can vary by as much as 25% with this change in electricity price.

From the above results it can be concluded that the financial returns on the application of SOFRAIR turbine in Redcar No.1 blast furnace are very attractive where at best the scheme can offer a pay back period of just over one year.

Even under the worst market conditions where a plant utilisation of 60% is achieved and a constant power recovery is maintained, installation of a SOFRAIR turbine at Redcar No.1 furnace can offer a payback period of three to four years.

Table 4.7 shows a comparative economic assessment for the installation of top pressure turbine on Redcar No.1, Llanwern No.3 and a pilot plant.

4.4.2 Funding the Scheme

It is recognised that under the present B.S.C. financial restrains where £3.45 million investment scheme, though offering good return, may be difficult to launch directly. In such cases, outside funding is available if so required.

Newton Chambers, the U.K. Licencee of the SOFRAIR turbine are prepared to invest the total capital costs of the project and lease the scheme to B.S.C. The leasing arrangements show [150] that if the start date of the contract were December 1980, NCE would complete contract in September 1982 and B.S.C. would commence lease in October 1982 at an annual rate of £567,732 for ten years, giving a total cost of £5.68 million for a plant at installed value of £3.4 million. Over this ten years period the credit from energy recovered would have been of the order of £20. million. With an anticipated increase in energy price, the savings achieved could be much larger.

Although the delivery period from ordering to commissioning could take as long as two years, the actual connecting of the turbine to the blast furnace mains with mains modifications can be carried out within a few days.

4.5 CONCLUSIONS

It can be concluded that modern blast furnaces are being designed and operated with top pressures in excess of 2 bar gauge. The reduction of this top pressure through the septum valve is wasteful of energy, the expenditure of which is accompanied by high noise levels which may themselves cause severe environmental problems. By allowing the gases to expand through an energy recovery turbine, power recovery of up to 30% of the total blower input together with noise reduction can be achieved.

The technology for the power recovery, though novel for B.S.C., is both proven and available. The economic case for the installation of a top pressure turbine is very much dependent on parameters such as top pressure, top gas rate, clean gas temperature, local cost of electricity etc.

Of the two main turbines with well proven records the SOFRAIR turbine is recommended in preference to the GUBT turbine based both on technical and economic merit. Technical/Operating data for the new generation of wet axial turbines is not available at the present time but assessment of of manufacturers data seem to suggest that this may offer the best results, when commercially proven.

An economic assessment shows that the application of the SOFRAIR turbine can offer a maximum power recovery of 14 MW and a constant power recovery of 10 MW on Redcar No.1 furnace, offering a net return of capital investment of £3.45 million within a payback period of two years, making it a highly attractive scheme.

The successful operation of the turbine is assured by the manufacturers and to back up their claims, under the present B.S.C. financial restraints, are prepared to fund the scheme on a leasing basis over a ten year period for a leasing premium of £0.57 million against a potential energy savings of £2.0 million per year.

ADDITIONAL BLOWER ENERGY REQUIREMENT WITH INCREASING BLAST COMPRESSION

Compression to - atm	Factor for Increased Energy Requirement for Initial Pressures - atms		
	2	3	4
2	1.0	-	-
3	1.72	1.0	-
4	2.33	1.32	1.0
5	2.80	1.59	1.20
6	3.20	1.82	1.38
7	3.58	2.00	1.54

ADDITIONAL BLOWER ENERGY REQUIREMENT
WITH INCREASING BLAST COMPRESSION

TABLE 4.1

GAS TEMPERATURE AFTER EXPANSION (DRY GAS) AND RECOVERABLE POWER (EXPANDED GAS FLOW RATE = 120,000 Nm^3/h)

Turbine Pressure Expansion Ratio	Top Gas Temp - °C			
	150°C		250°C	
	Temp. - °C	Power - kw	Temp - °C	Power - kw
1.2	133	750	229	930
1.4	120	1360	213	1680
1.6	109	1860	199	2300
1.8	99	2290	187	2840
2.0	91	2670	177	3300
2.5	75	3420	157	4230
3.0	62	4010	141	4950
3.5	51	4480	128	5540

EFFECT OF GAS TEMPERATURE ON RECOVERABLE POWER

TABLE 4.2

POWER REQUIREMENT FOR BLOWER (85,000 Nm^3 blast/h)

Pressure in Front of Turbine - atm.	Blower Pressure - atm.	Blower Power (W_B) - kW
1.0	2.20	3190
1.2	2.28	3350
1.4	2.38	3550
1.6	2.51	3800
1.8	2.64	4030
2.0	2.78	4280
2.5	3.16	4910
3.0	3.57	5540
3.5	4.00	6130

POWER REQUIREMENTS FOR BLOWER

TABLE 4.3

POWER GAIN $W_G - W_B$ kW

Pressure in Front of Turbine (atm)	Blower Power (W_B) KW	Gas Temperature 150°C		250°C	
		WG KW	$W_G - W_B$ KW	WG KW	$W_G - W_B$ KW
1.2	3350	750	-2600	930	-2420
1.4	3550	1360	-2190	1680	-1870
1.6	3800	1860	-1940	2300	-1500
1.8	4030	2290	-1740	2840	-1190
2.0	4280	2670	-1610	3340	- 940
2.5	4910	3420	-1490	4230	- 680
3.0	5540	4080	-1539	4950	- 590
3.5	6130	4480	-1650	5540	- 590

TABLE 4.4

POWER GAIN V/S TOP PRESSURE

PERFORMANCE OF HITACHI WET/AXIAL TURBINE

	Wakayama Steel Works		Kohura Steel Works
	No.5 BF	No. 4 BF	No. 2 BF
Gas Cleaning System	Baumco Venturi scrubber	Single Stage Venturi scrubber	Ring Slit Washer
Rated Dry Gas Flow (Nm^3/h)	345×10^3	330×10^3	250×10^3
Top Pressure (Atg)	1.5	1.5	1.4
Turbine Inlet Pressure (Atg)	1.25	1.35	1.2
Turbine inlet Gas Temp. °C.	42	45	46
Exhaust Pressure (mmAq)	500	1400	800
Rated Output (KW)	6,050	5,650	4,800
Capacity (KW)	7,000	7,000	6,000
Operation	Remote auto start and stop function with manual control Unmanned supervision during operation		
Turbine Type	Axial flow, raction stage, 3 stages with washing water injection system.		
	Double Flow		Single flow

TABLE 4.5

EFFECT OF ELECTRICITY PRICE ON PAYBACK PERIOD

NET SAVINGS/ANNUM = £million

	SOFRAIR Elect. Cost.		GUBT Elect. Cost.	
	£0.027/ KWL	£0.02/ KWL	£0.027/ KWL	£0.02/ KWL
Max. Power Recovery	2.96	2.16	1.35	0.4
Payback Period Years	1.17	1.60	2.20	7.4
Constant Power Recovery	2,07	1.50	0.75	Nil
Payback Period Years	1.67	2.31	3.95	-
Plant Utilisation 60% & Constant Power recovery	1.18	0.84	-	-
Payback period Years	2.66	4.11		

EFFECT OF ELECTRICITY PRICE ON PAYBACK PERIOD

TABLE 4.6

ECONOMIC ASSESSMENT

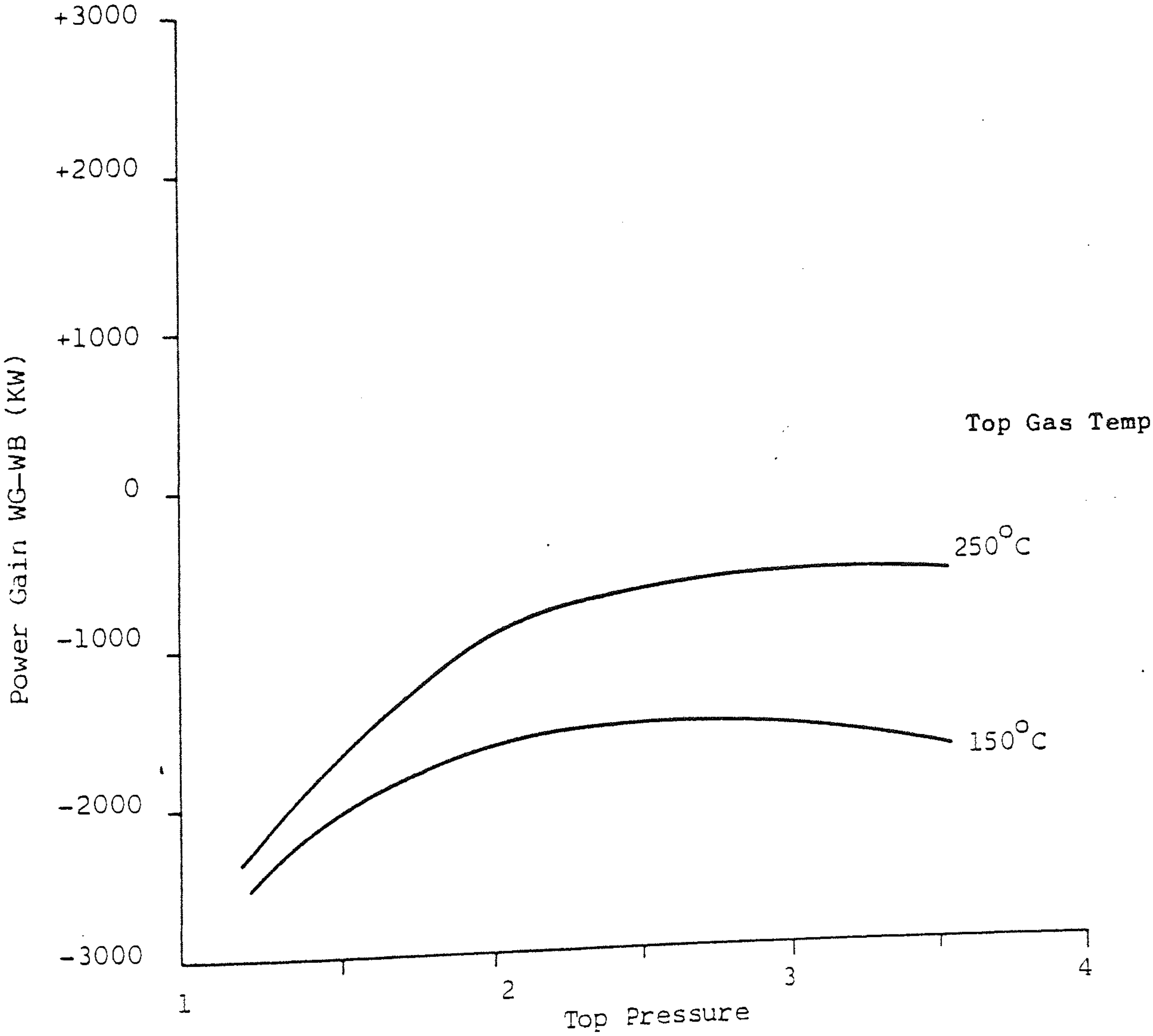
(Assume Electricity Price @ £0.027/Kwh)

	REDCAR NO.1		LLANWERN NO.3		PILOT PLANT
Hearth Dia - m	14.0		11.2		8.0
Top Gas Vol. - 10^3 Nm^3/h	620		459		170
Top Gas. Press - Kg/cm^2	2.5		.5		1.5
Turbine Inlet Press. - Kg/cm^2	2.25		.26		1.26
	53		53		53
	SOFRAIR	GUBT	SOFRAIR	GUBT	SOFRAIR
Max.Pwr. Recovered (MPR)-MW	14.0	17.5	7.0	8.84	2.58
Constnt. Pwr Recvd (CPR)-MW	10.0	13.0	5.0	6.63	2.0
Capital Cost -£x10^6	3.45	2.94	2.43	1.58	1.06
Running Cost (MPR) -£x10^6	.156	2.50	.122	1.66	.072
(CPR) -£x10^3	.156	2.16	.122	1.37	.072
Savings (MPR) -£x10^6	2.96	1.35	1.44	0.31	-
(CPR) -£x10^6	2.07	0.75	0.99	0.11	.)37

TABLE 4.7

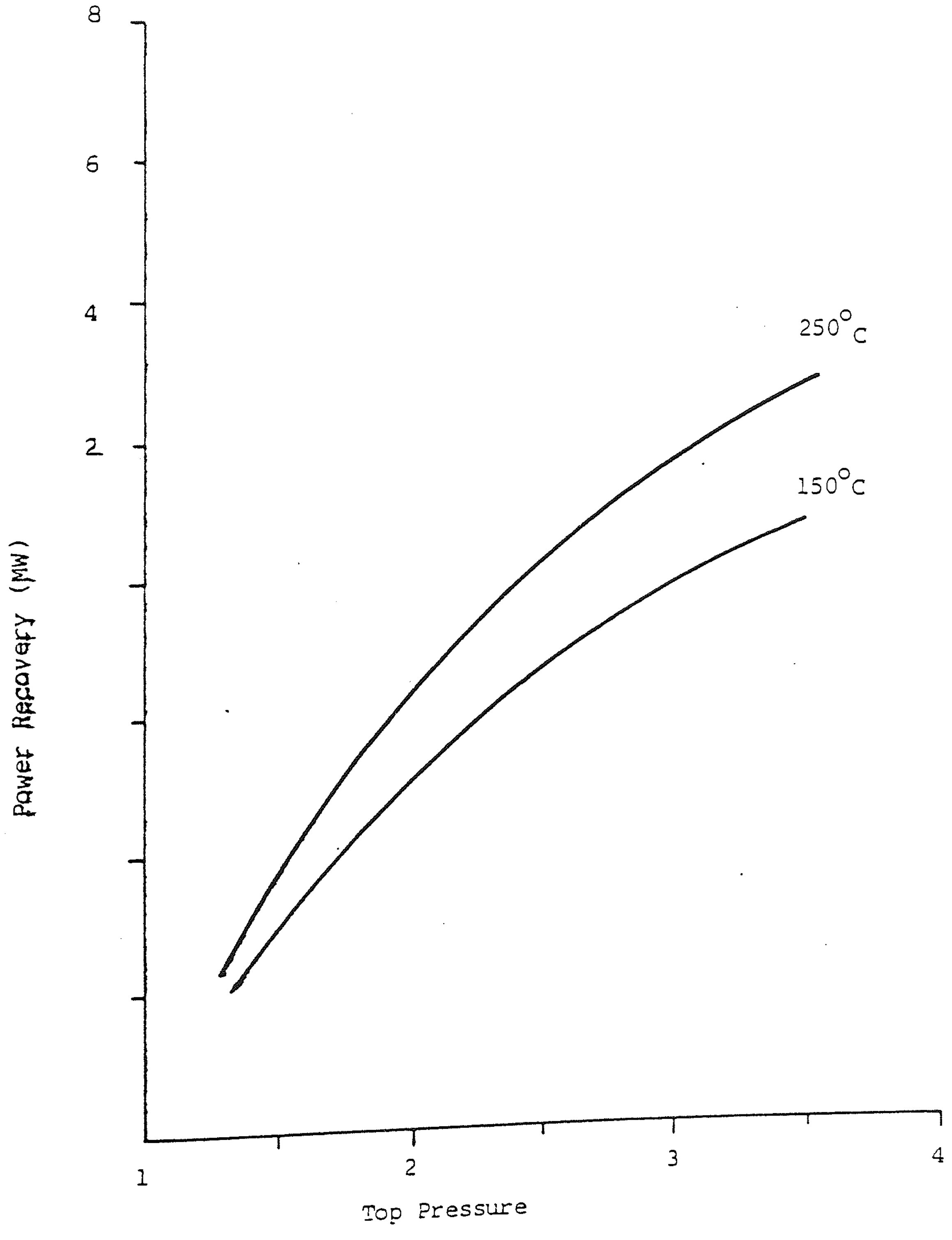
COMPARATIVE ECONOMIC ASSESSMENT

Gas flowrate @ 120,000 Nm^3/h



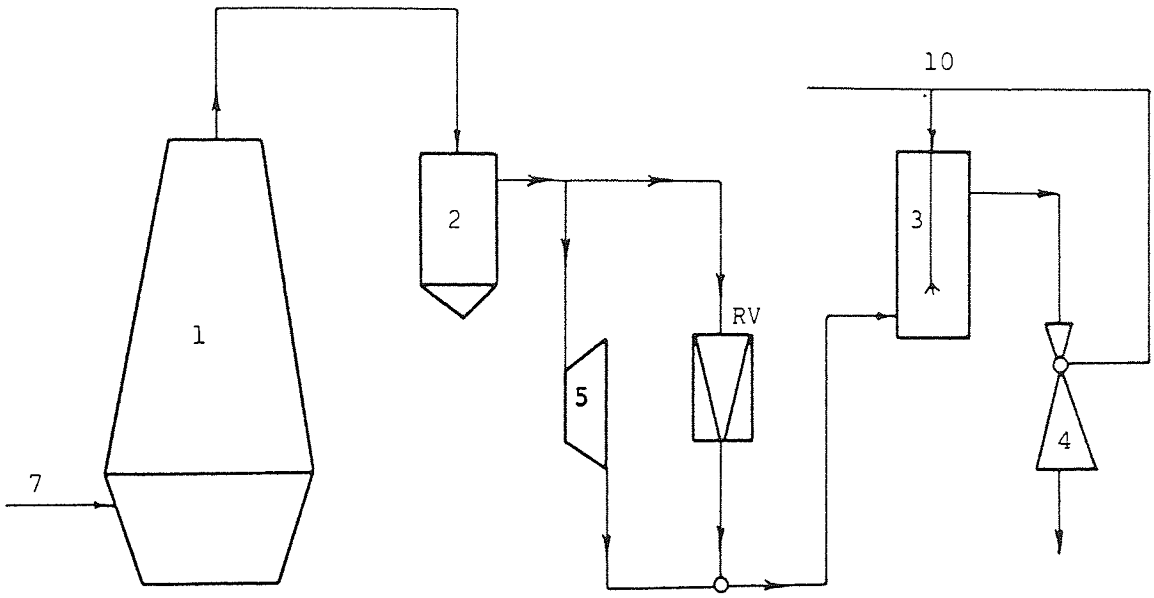
POWER GAIN AS A FUNCTION OF TOP PRESSURE

FIGURE 4.1



RELATION OF TOP PRESSURE AND POWER RECOVERY

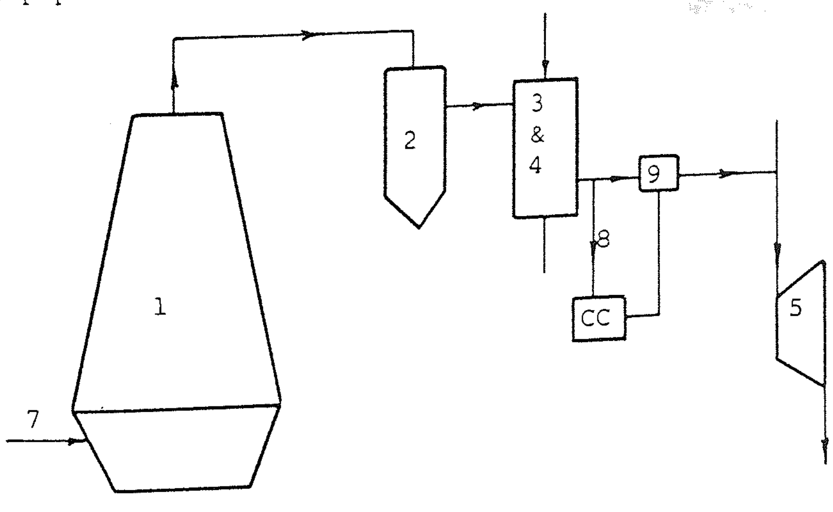
FIGURE 4.2



EXPANSION TURBINE - TYPE A

1 Blast Furnace
2 Dust catcher
3 Scrubber
4 Venturi scrubber
5 Expansion turbine
6 Clean gas
7 Hot blast
8 Gas bled off for partial combustion
9 Mixing chamber
10 Water for gas scrubber
RV Top pressure control facilities

FIGURE 4.3



EXPANSION TURBINE - TYPE B

FIGURE 4.4

PICTORIAL VIEW OF SOFRAIR TURBINE

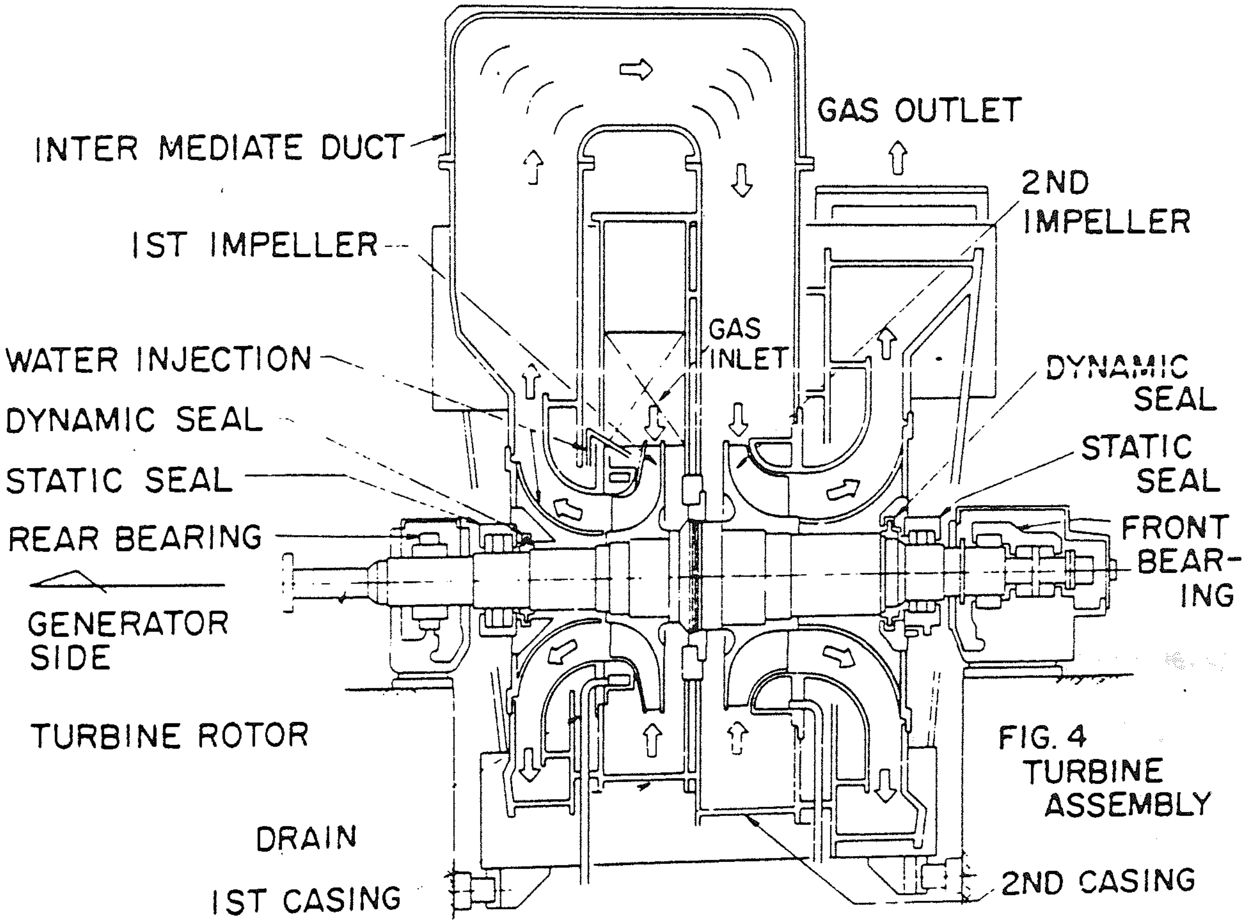
PICTORIAL VIEW OF SOFRAIR TURBINE

FIGURE 4.5

PICTORIAL VIEW OF TURBINE ROTOR

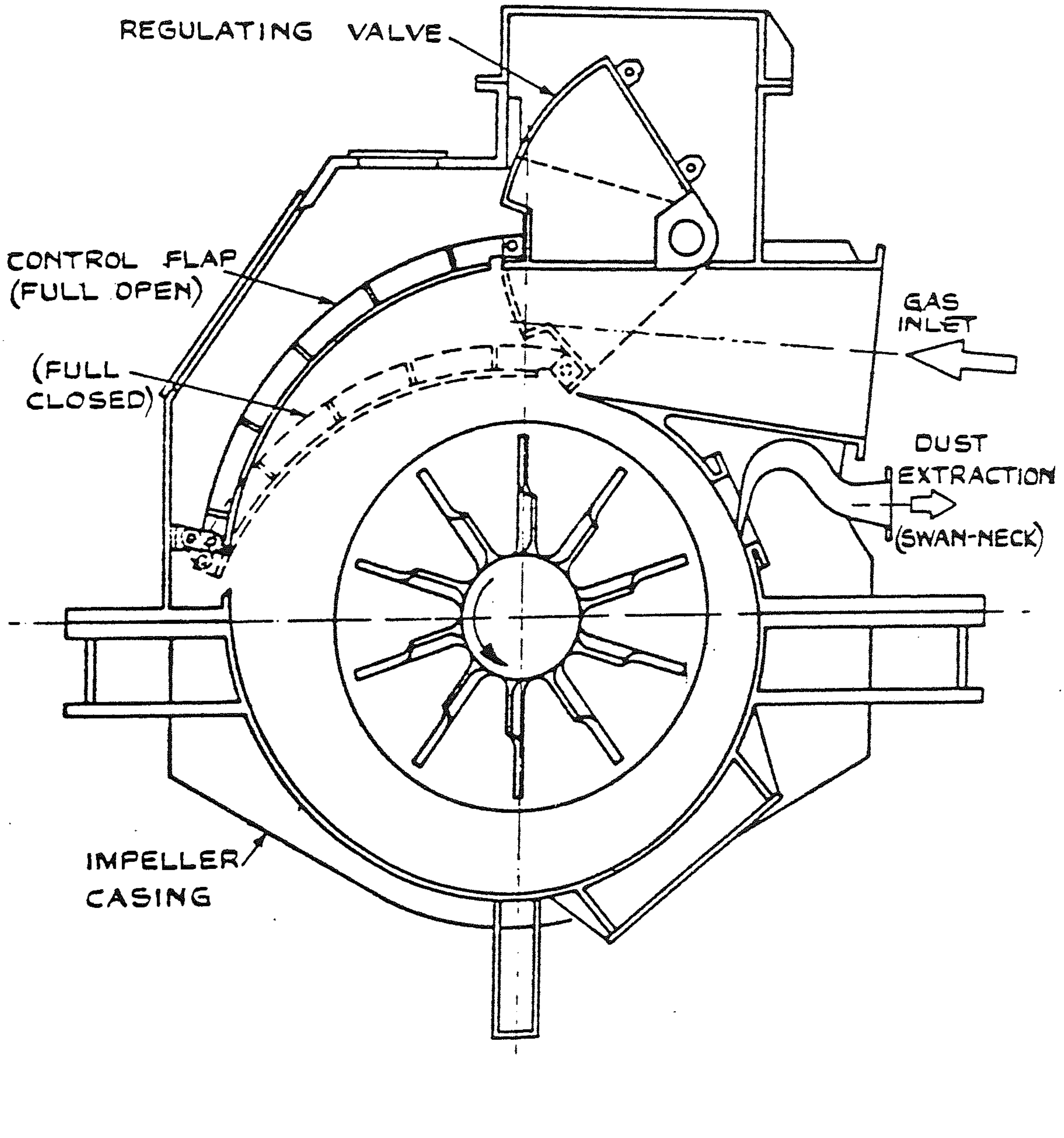
PICTORIAL VIEW OF TURBINE ROTOR

FIGURE 4.7



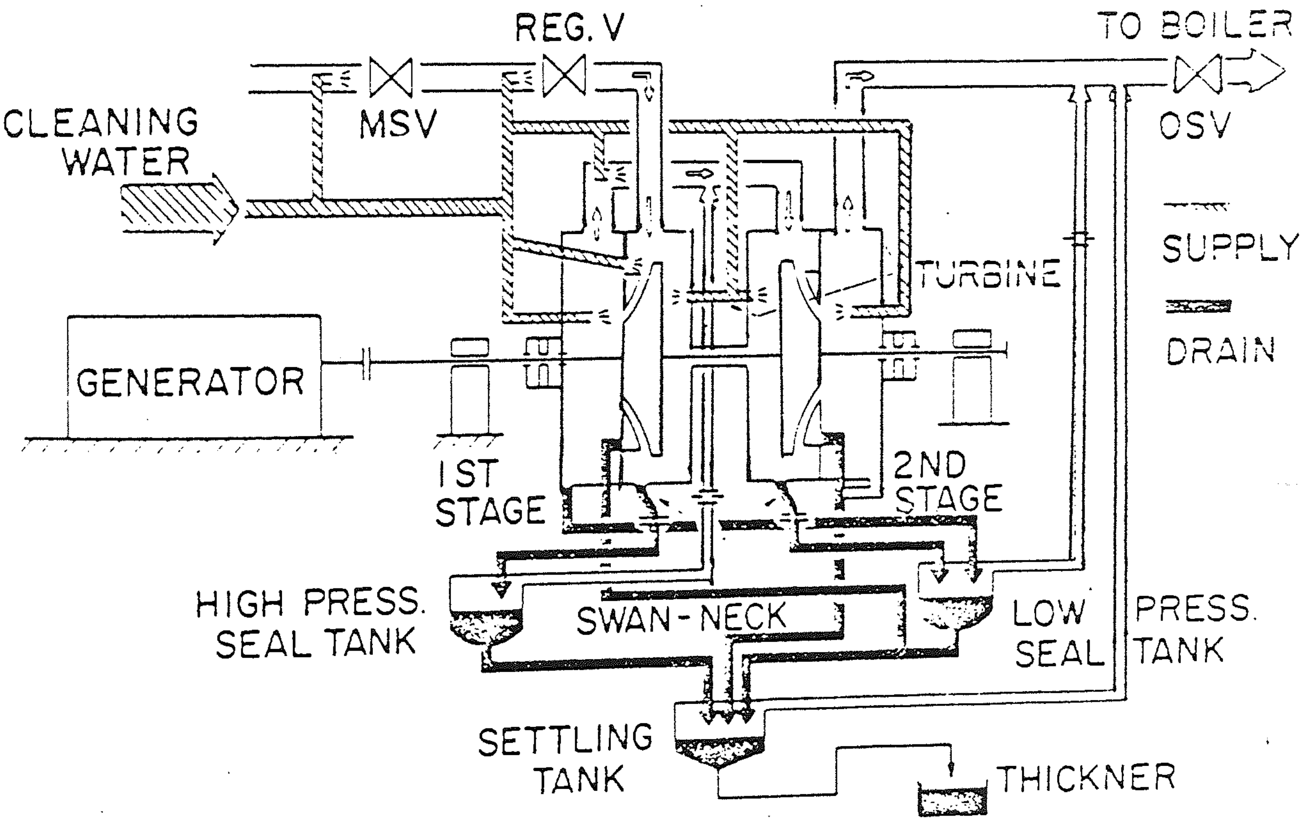
SECTION THROUGH ASSEMBLED TURBINE

FIGURE 4.8



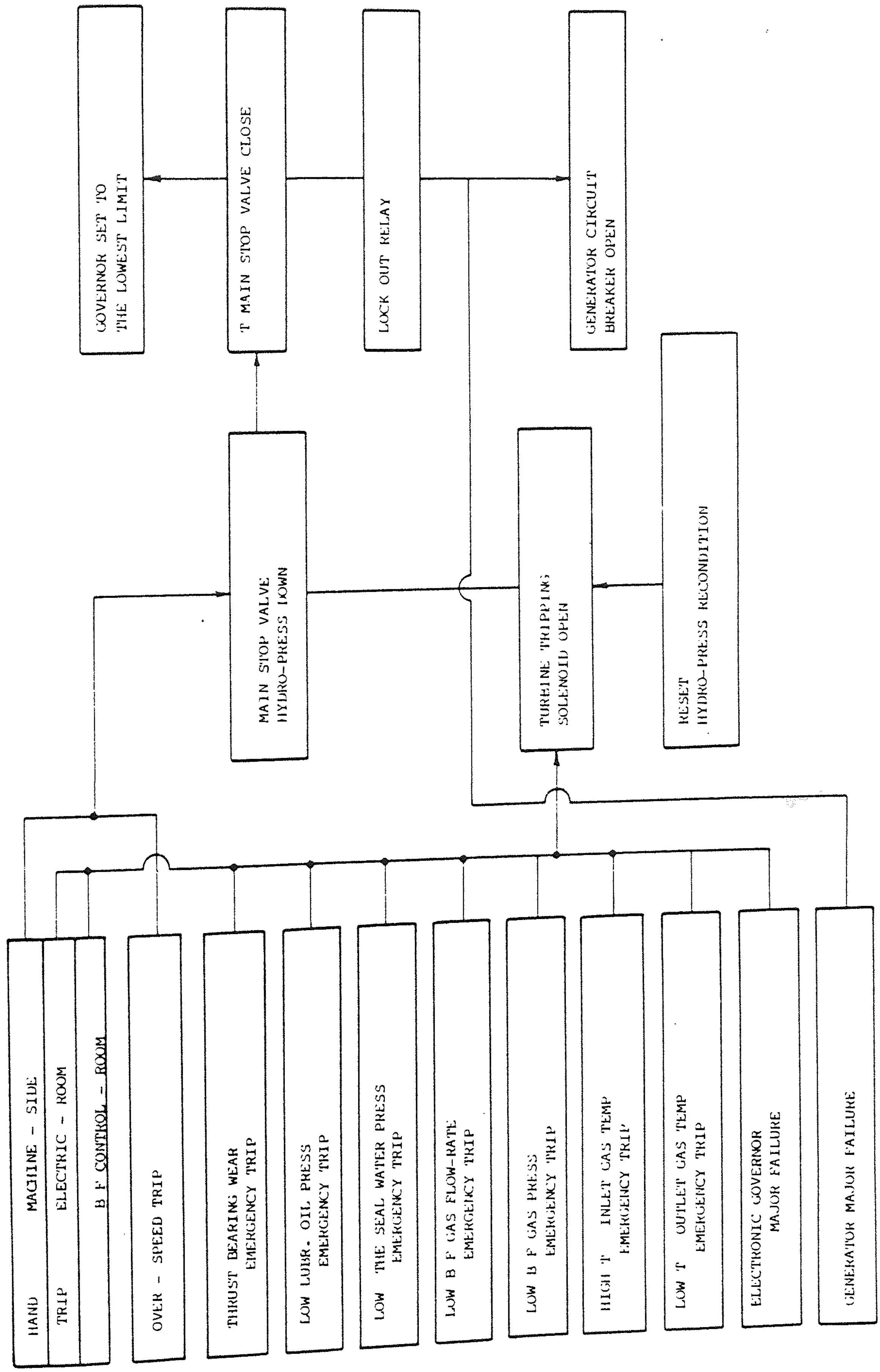
Cross-section through a Sofrair-Bischoff centripetal turbine

FIGURE 4.9



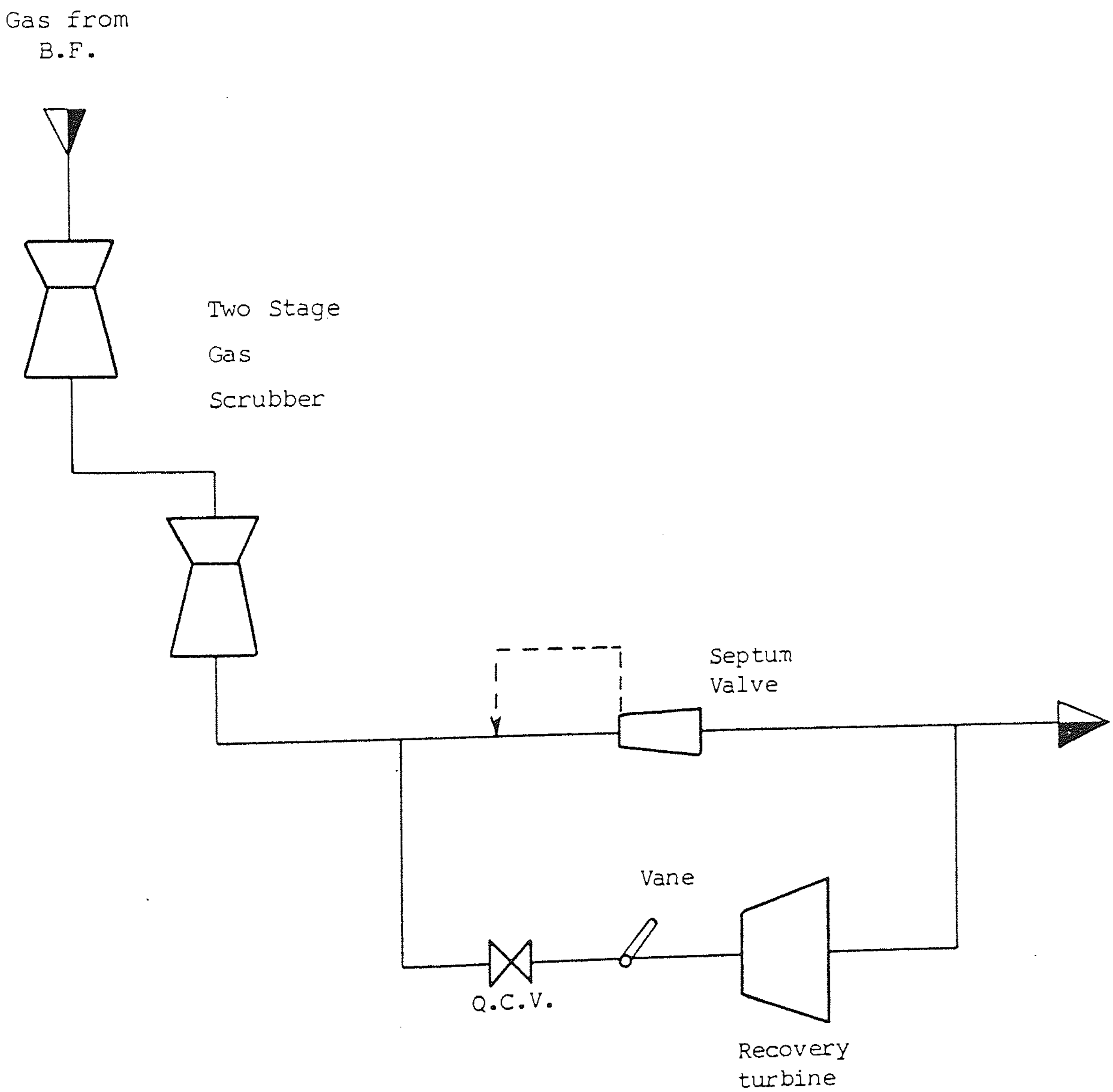
WATER INJECTION SYSTEM

FIGURE 4.10



SAFETY SYSTEM: BLOCK DIAGRAM

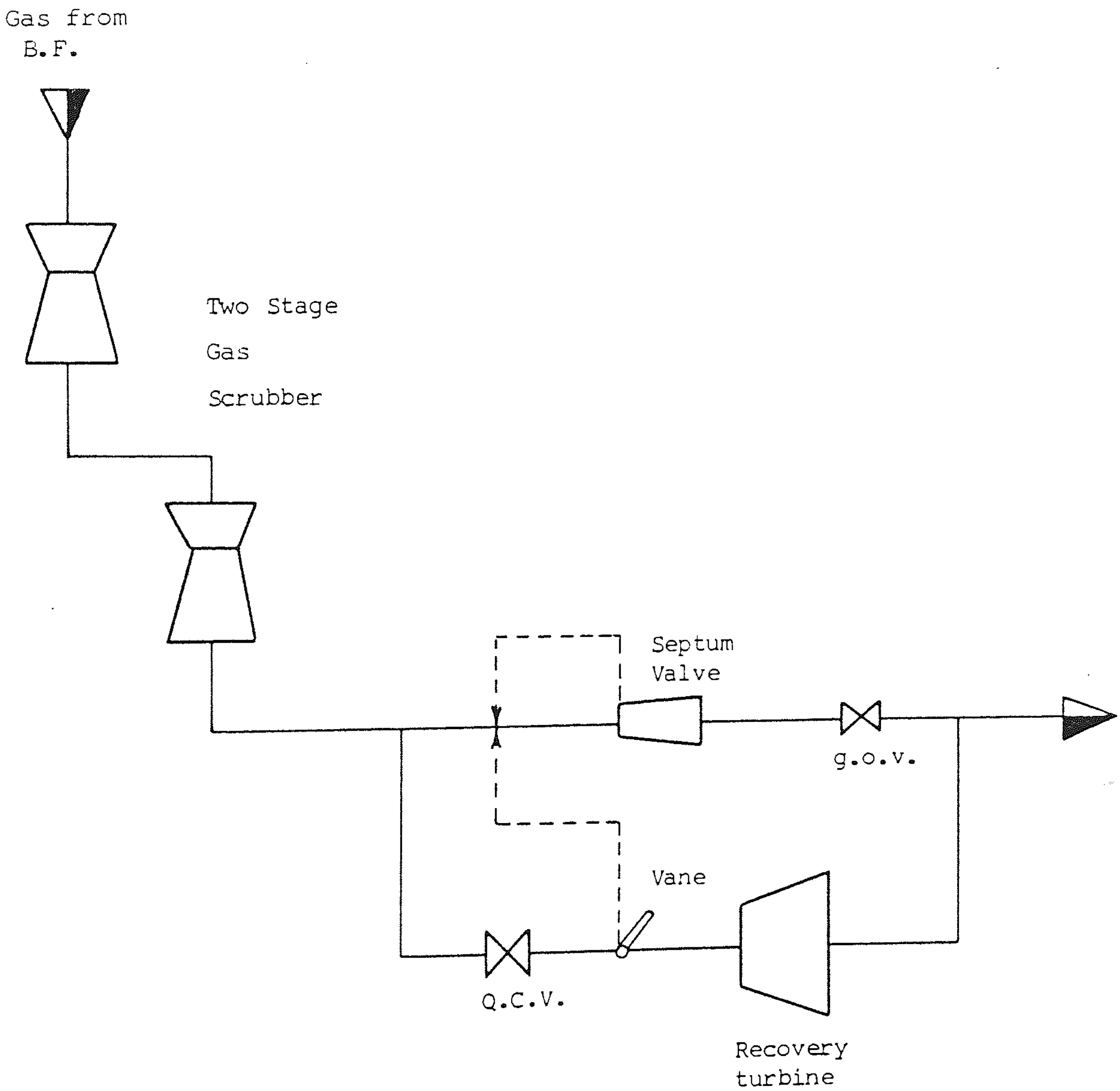
FIGURE 4.11



Q.V.C. - Quick Acting Valve (Closing)
Butterfly)

CASE 1: CONSTANT POWER RECOVERY

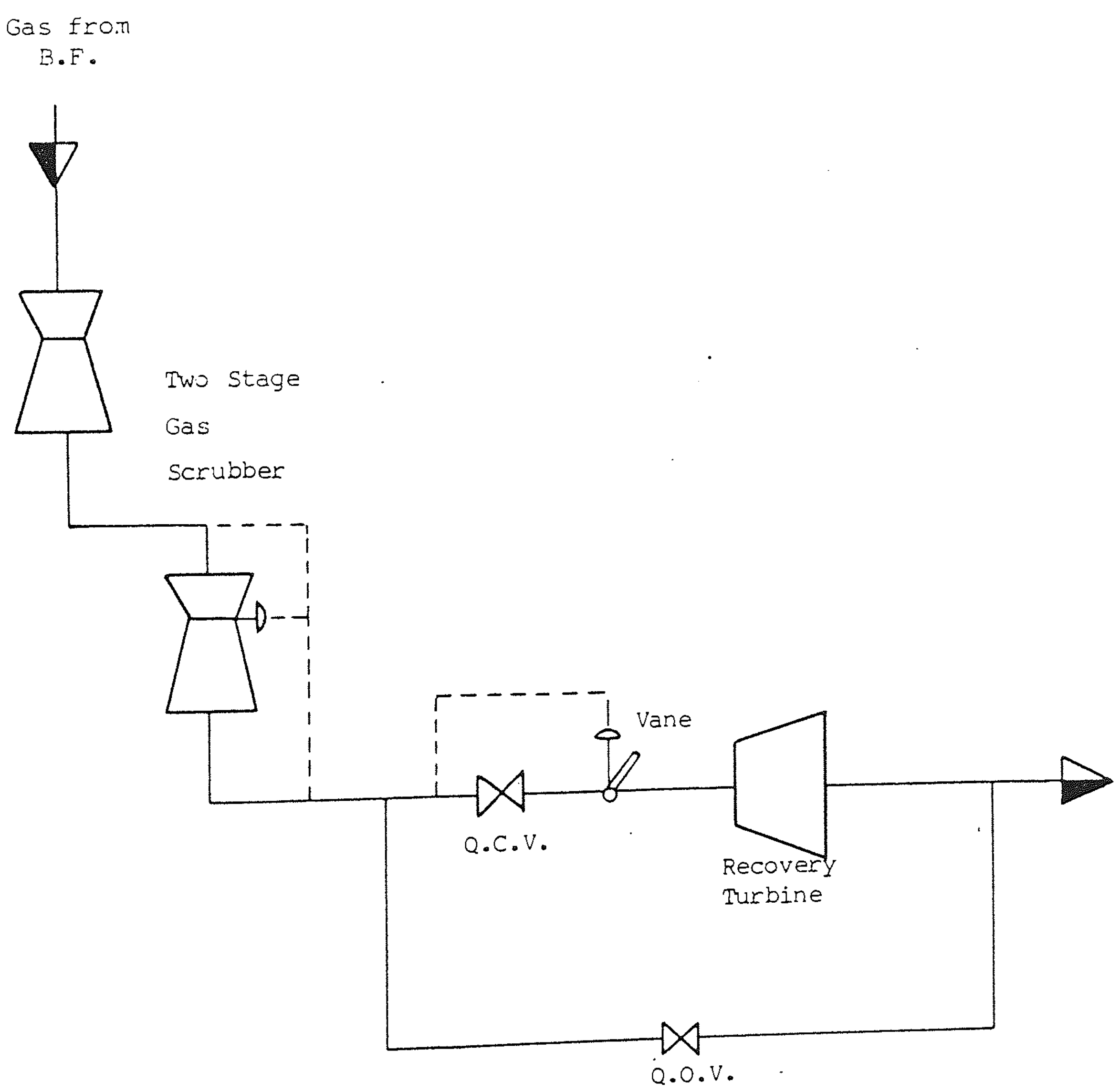
FIGURE 4.12



Q.C.V. - Quick Acting Valve (Closing)
(Butterfly)

CASE 2: MAXIMUM POWER RECOVERY

FIGURE 4.13

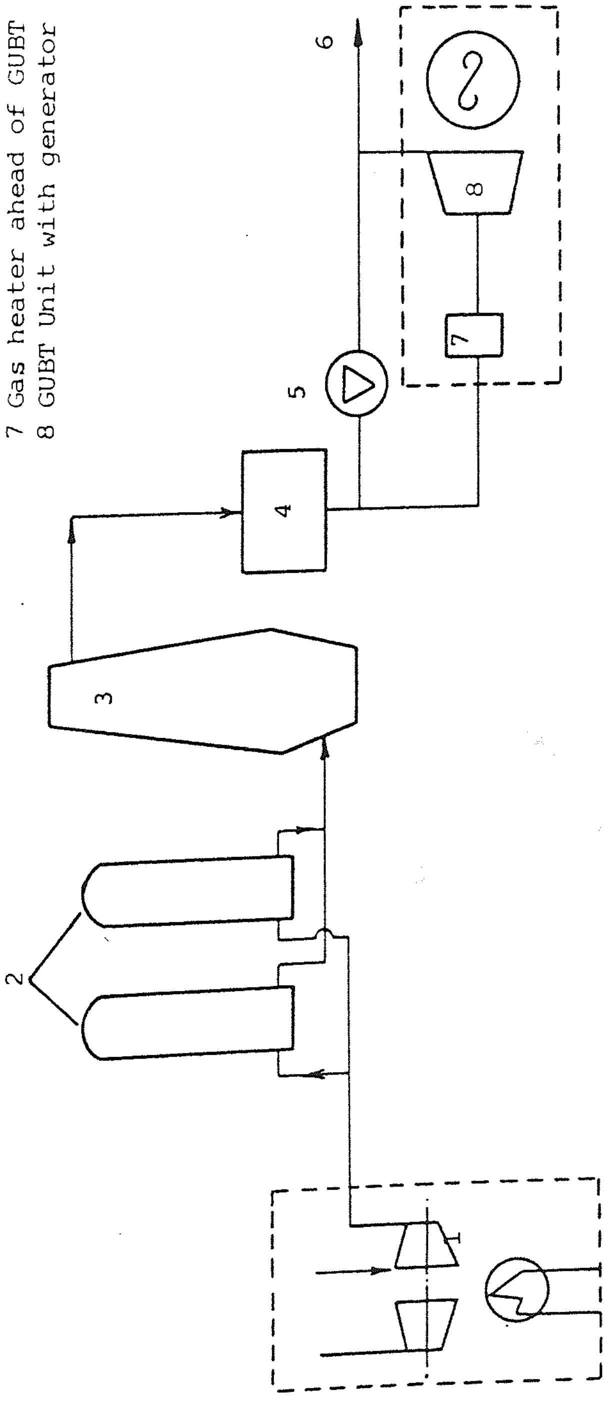


Q.C.V. - Quick Acting Valve (Closing)
Q.O.V. - Quick Acting Valve (Opening)

CASE 3: MAXIMUM POWER RECOVERY WITH VARIABLE PRESSURE

FIGURE 4.14

1 Blast blowers
2 Blast stoves
3 Blast furnace
4 Gas purification (scrubber followed by precipitator)
5 Septum valve
6 Blast furnace gas of low pressure to consumers
7 Gas heater ahead of GUBT
8 GUBT Unit with generator



LINE DIAGRAM OF 'GUBT' TURBINE

FIGURE 4.15

Aston University

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Aston University

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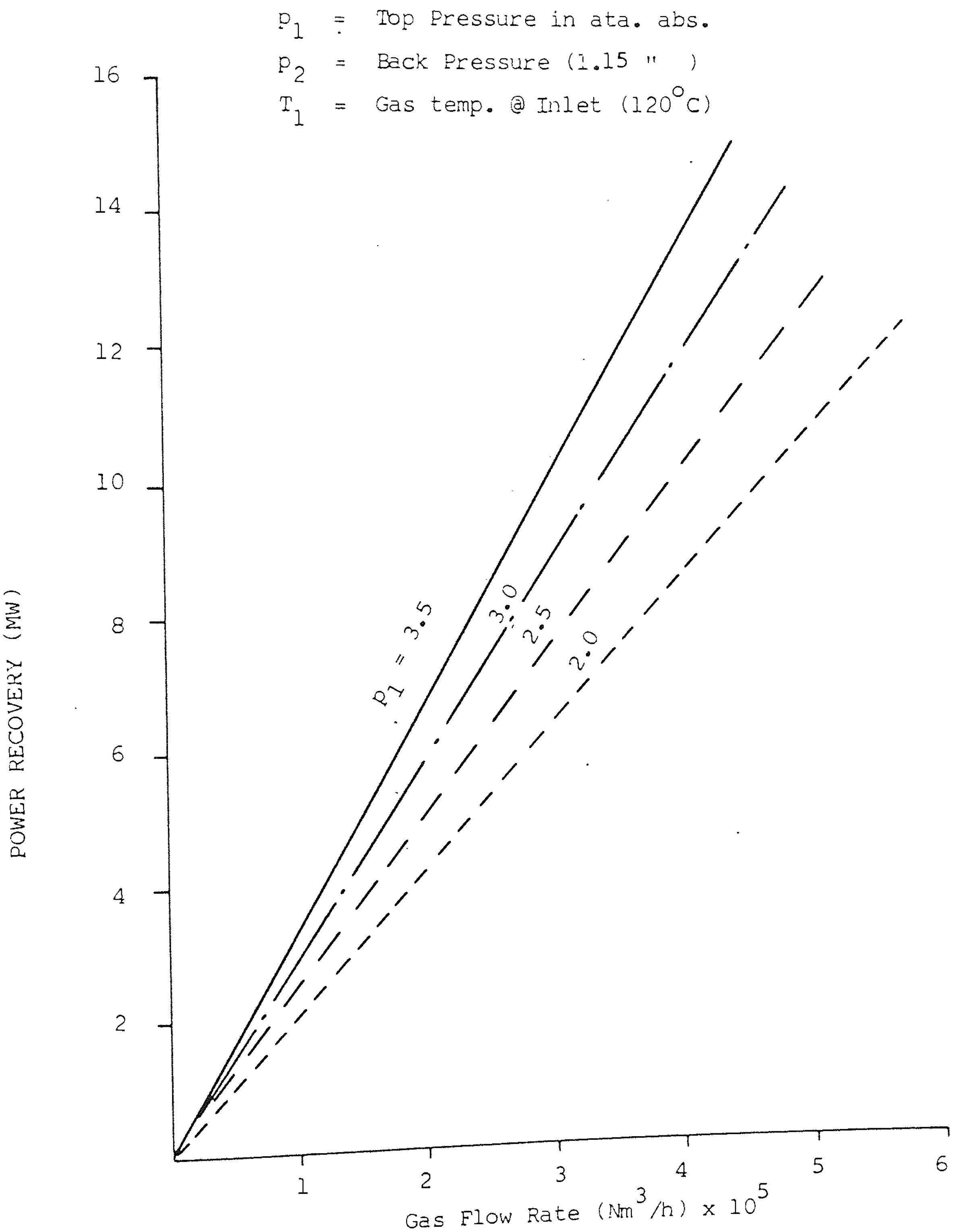
PICTORIAL VIEW OF GUBT TURBINE

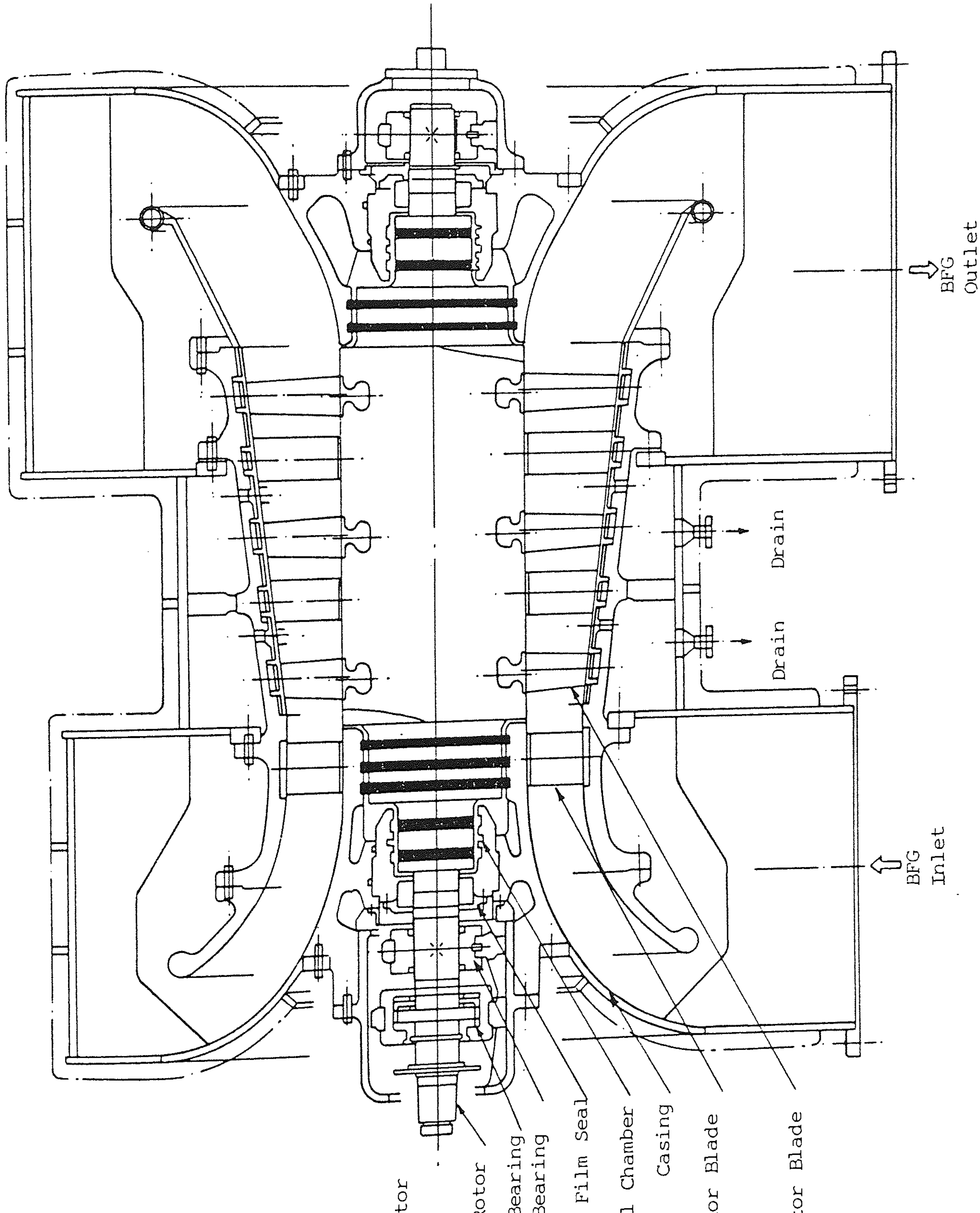
PICTORIAL VIEW OF GUBT TURBINE

FIGURE 4.16



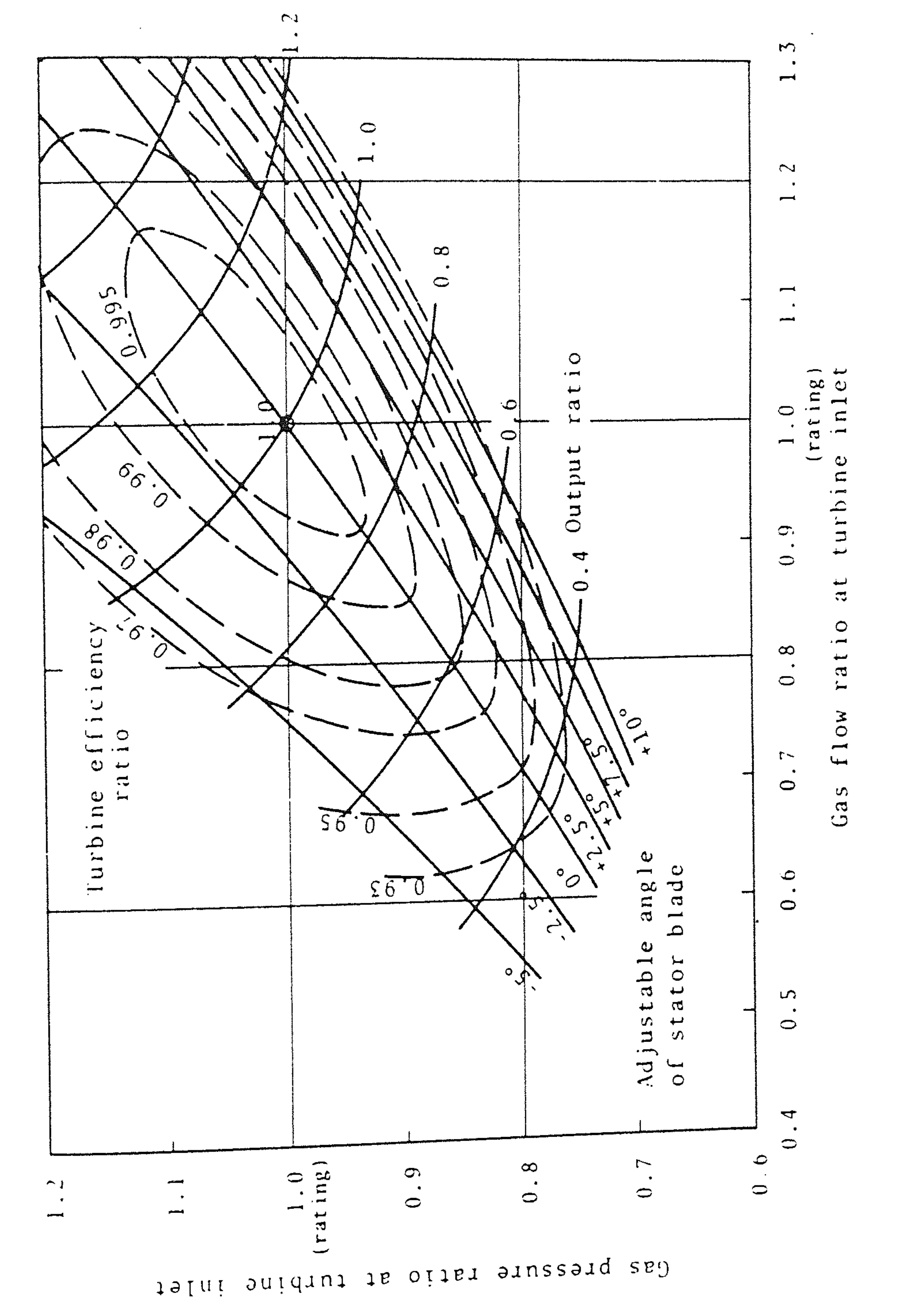
'GUBT' TURBINE POWER RECOVERY

FIGURE 4.17



MITSUI 'TRT' RECOVERY TURBINE - SECTIONAL DRAWING

FIGURE 4.18



OPERATING RANGE WITH VARIABLE NOZZLES

FIGURE 4.19

CHAPTER 5

ENERGY SAVINGS: STEELMAKING

5.1 INTRODUCTION

The choice of steelmaking process in any particular part of the world depends mainly on the compostion and availability of the essential raw materials e.g. iron ore, fuel and scrap[151] The basic Oxygen Steel (BOS) process is the major method of making steel in the UK and accounts for 65 per cent of total crude steel production in the 29 IISI member countries.[152] It consists of blowing a high velocity jet of oxygen onto the 'hot-metal' and scrap held in the vessel. The oxygen thus reacts with the carbon in the hot metal producing carbon monoxide.

Due to the intermittent nature of the BOS process the composition of waste gas varies during the blowing as shown in Fig.5.1. At the beginning of the blow the decarburisation reaction is not established. Thus the waste gas is mainly composed of CO_2, N_2 and O_2, the evolved CO being fully burned. One to two minutes after the start of blow the vessel reaction is underway, CO gas evolution rates are high and suppressed combustion conditions are established.

The rate of reaction and both the energy input and energy released very much depend on the ratio of hot metal to scrap and the method of blowing i.e. blowing rate, lance head position, etc.[153] The reaction is highly exothermic.

The calorific value and sensible heat of waste gases from the Basic Oxygen Steelmaking process represent a major source of potentially recoverable waste energy to the steel industry.

These carbon monoxide rich gases, having a typical calorific value of 8 MJ per m^3 and a BOS vessel exit temperature of 1600^{o}C, contain about 800MJ of energy for each tonne of liquid steel produced.(154) Recovery of this energy can reduce the total energy consumption in finished steel production by up to 3 per cent and, for a 5 million tonnes per annum steelworks, recovery at the rate of 480GJ per hour (133 MW thermal) is theoretically possible.(155)

Energy recovery from BOS gases is widely practised overseas and all major Japanese BOS plants are fitted with recovery systems. In contrast, however, all major BSC BOS plants reject most of the available energy to the environment as flared gas and in cooling water. Some early BSC BOS plants incorporated energy recovery systems in the form of waste heat boilers situated directly above the vessel, but operational difficulties forced the system to be disbanded.

The modern trend in BOS gas energy recovery has moved away from the waste heat boiler concept,(156) i.e. direct recovery of high-grade steam, and has moved towards recovery of cleaned BOS gas as a fuel gas source, together with a reduced make of steam from the sensible heat content of the gases. The widespread Japanese recovery plants generally employ the cleaned fuel gas technique.

In response to the need for enhanced energy conservation, brought about by the recent dramatic upswing of world energy prices, several of the major BSC works are re-considering the BOS waste gas problem and are focussing more attention on the new cleaned gas recovery technique.[157]

This chapter briefly considers the three main types of BOS energy recovery systems in turn and proceeds to develop the fuel gas recovery technique for a BSC installation.

5.2 THERMAL BALANCES IN BOS STEELMAKING

In order to arrive at a total flow of energy into the BOS furnace it is assumed that the hot metal feed carried the energy of transformation consumed in the blast furnace and the sensible heat and solution energy are in proportion. From the earlier BSC work, the energy required to dissociate iron-oxide into metallic iron and oxygen is included in all the scrap addition (self generated or purchased from oxide) and assumed to be 7400 MJ per tonne. In addition an energy input value of 1600 MJ has been included to cover the exothermic release of energy due to chemical (notably oxidation) reactions. The calculations thus show that a total flow of energy into the BOS furnace is of the order of 12,180 MJ per tonne of steel.[158-159]

Energy losses from the BOS plant are shown in Table 5.1. The table shows that as much as 25% of the energy loss is carried in the hot waste gases from the BOS process representing a major source of potentially recoverable energy.

these anticipated losses, predicts a steam make of 200 kg per tonne of liquid steel, at 15 bar gauge. From this figure, it is apparent that the Full Combustion System can offer an annual steam saving worth £8 million for a 5 million tonnes per annum steelworks, assuming a value of £8.00 per tonne for steam at 15 bar gauge.

Whilst the potential steam savings are very attractive theoretically, the Full Combustion System has a number of serious operational drawbacks and, despite its installation on a number of early BSC BOS plants, its operating record is not good and the systen is no longer favoured. The principal disadvantages of the Full Combustion System are as follows:

(a) The position of the waste-heat boiler directly above the vessel, and ahead of the gas cleaning plant, threatens vessel availability in that any boiler failures result in the need to halt steel production to safeguard the gas cleaning plant against non cooled gases. There is an additional serious safety hazard in the advent of water-tube rupture directly above the vessel.

(b) The intermittent nature of the BOS vessel requires auxiliary firing in the boiler hood to provide a steady steam make and allow for efficient boiler operation. Auxiliary firing requires extra capital expenditure and creates operational problems

(c) A large hood, ducting, and gas cleaning plant, are necessary to handle the high gas volumes resulting from the combustion of the BOS gases in 100 per cent excess air.

5.3.2 Partial Combustion System

In the Partial Combustion System the gases from the BOS vessel are partially burned in a deficiency of air, normally about 30 per cent of the stoichiometric volume. The partial combustion again takes place in a hood situated directly above the BOS vessel, as in the Full Combustion System, the heat produced may be removed by water cooling or by using the hood as the radiation section of a waste-heat boiler. In the case of the Partial Combustion System it is not usual to include a convection section for the waste heat boiler. The cooled burned gases are cleaned and vented to atmosphere.

The Partial Combustion System has the advantage of a considerably smaller hood and ducting compared with the Full Combustion System, because of the large reduction in burned gas volume. In addition, it is possible to use a wet scrubber type of gas cleaning plant rather than the more expensive electrostatic precipitators normally used in the Full Combustion System.

The recoverable heat from the Partial Combustion System is reduced to 225MJ per tonne of liquid steel, by virtue of the fact that the burned gas temperature is only of the order of 875°C. This heat release equates to a maximum theoretical steam generation of 80kg per tonnne liquid steel at 15 bar gauge and, if allowances are made for predictable heat losses, this figure falls to 79kg per tonne of liquid steel which would yield annual saving of £3.16 million for a 5 million tonnes per annum steelworks assuming a value of £8.00 per tonne for steam at 15 bar gauge.

5.3.3 Minimum Combustion System

The Minimum Combustion System is designed to reduce air ingress between the BOS vessel and the extraction hood immediately above the vessel to an absolute minimum, thereby preventing combustion and conserving the calorific value of the carbon monoxide rich BOS gases. The desired effect is obtained by a combination of hood design and careful pressure control, and it is quite practical to reduce air infiltration to below 10 per cent of the stoichiometric volume.[161]

The unburned gases enter the hood at about 875°C and a small degree of steam generation is possible as an alternative to water cooling on an open or closed circuit. The cooled gases are cleaned in a wet scrubber and then pumped into a storage holder ready for use as fuel gas.

The calorific value of recovered BOS gas can be estimated from a knowledge of the Decarbonising Efficiency, (η_c) and the Air Factor, (λ), (Fig. 5.2).

The Air Factor is the ratio of the volume of air ingress into the unburned BOS gases to the stoichiometric volume of air required for complete combustion, and is typically of the order of 0.10 or slightly less.

Given a hot metal ratio of 85% with a 4.4% carbon, air factor of 0.1 and a steel yield of 93%, a potential energy recovery of approxaimtely 710MJ/tonne steel in the form of fuel gas (with typical calorific value of $8MJ/Nm^3$) and steam is possible. However it is recognised that the actual energy recovered is not only dependent on the above factors but also on the operational philosophy in relation to collection times, vessel/skirt gaps, hood pressure control, etc.

Reports from the Japanese steel industry[162] indicate that the value of recovered clean BOS gas which may be expected from the Minimum Combustion System is of the order of 475MJ per tonne of liquid steel produced. In addition, a theoretical recovery of 44 kg of steam at 15 bar gauge is possible in cooling the gases from 875°C above the vessel down to the gas cleaning temperature, which involves removing 125MJ of sensible heat per tonne liquid steel. A practical recovery figure, making allowances for predicatable losses, is likely to be about 40 kg of 15 bar gauge steam per tonne liquid steel.

The total potential savings from implementing the Minimum Combustion System to a 5 million tonnes per annum steelworks is of the order of £6.9 million per annum. The major portion £5.3 million, of this saving is derived from the fuel gas, assumed to be valued at £2.3 per GJ in line with fuel oil pricing; the minor portion, £1.6 million, is derived from steam, assumed to be valued at £8.00 per tonne.

5.3.4 Recovery System Selection

In terms of total potential for recovery of energy from BOS gases, the Full Combustion System offers the best return with 235kg of steam per tonne of liquid steel produced, valued at £8.0 million per annum for a 5 million tonnes per annum steelworks. The Minimum Combustion System offers a slightly reduced return from recovery of fuel gas and steam valued at £6.9 million per annum on a comparable basis. The Partial Combustion System offers a much poorer return from 70kg of steam per tonne of liquid steel produced valued at £2.7 million per annum on a comparable basis.

The O G Minimum Combustion System, however, offers a number of significant advantages over both the other systems in respect of capital cost and operational factors. The capital cost of this system is less than its two rivals by virtue of the reduced gas volumes handled, and the absence of large waste-heat boilers directly above the vessel greatly reduces operational problems and associated safety hazards. The reduced duct size, in addtion to reducing capital costs, facilitates waste gas flow measurement which is important for computer control of the BOS process. The collection of gas represents a convenient, low-cost means of securing a flexible energy supply for the works.

The last point on flexibility of supply stems from the intermittent nature of the BOS process, because it is not practically possible to synchronise the blowing of adjacent BOS vessels to ensure a uniform make of waste gases. It becomes necessary, therefore, to provide some peak smoothing mechanism in the recovered energy system, and it is much simpler to collect and store gas than it is to add auxiliary firing to overhead waste-heat boilers or provide large steam accumulators.[163-164] In addition, fuel gas can be adopted into the works energy system much more easily than steam which is relatively inflexible in its use, whereas gas can be routed to a variety of furnaces, power generating plant and steam boilers or scrap preheating.[165]

Consideration of the above factors has led to the current view that the Minimum Combustion System offers the most desirable recovery alternative.

5.4 CASE STUDY: BOS GAS RECOVERY SCHEME AT LACKENBY STEELWORKS

5.4.1 Introduction:

The major objectives of the scheme are to recover BOS waste gas to supplement gaseous fuel requirements at Cleveland and/or Lackenby steelworks thus reducing production cost per tonne of steel.

The recovery of BOS gas has long been practised in Japan since the first commercial operation of the OG gas collection system at Tobata No.2 BOS plant in March 1962. At the end of 1977, of the sixty five suppressed combustion gas collection units operating in Japan (of which 5 units are IRSID type) sixty are equipped for gas recovery.[166] The recovery process is therefore both long established and widely used.

In order to relate the Japanese experience on to BSC installation it is important to look at their blowing practices and hence compare them to thr BSC operations. The main converter blowing parameters are thus summarised below:

Plant	Av.Heat Size (Tonnes)	Av. Blowing Time (Mins)	Av. O_2 Rate m^3/min	Vol. Inside Lining (m^3)	Specific Vol. (m^3/tonne)
Oita	341	16	915 (1,330 max)	325.8	0.955
Mizushima No.1	200	20	500	176	0.880
Kakagowa	246	17	700	210	0.853
Nagoya No.2	250	17	670	221	0.884
Kimitsu No.1	240	17	600	241	1.004
Ohgishima	269	17.5	700 (1,100 max)	231	0.858
Lackenby	260	17	800	206	0.79

The oxygen input rates and blowing times of the Japanese plants are generally in line with current BSC practice but the specific volume of convector is higher. This is important in that a low specific volume makes blowing control more difficult in terms of:

(a) Slopping - there is less volume to contain the foaming slag developed during the maximum carbon removal period and therefore there is a greater risk of slag and metal ejections from the converter.

(b) Hood Pressure Control - similarly there is less volume to take up the sudden surges in gas extraction from the bath; a high internal volume tends to act as a 'shock absorber' which assists the hood control system.

This would seem to explain to some extent the most obvious differences between the Japanese practice and that currently at some BSC works, in that the Japanese operate with a smaller visible gap between the vessel top and skirt. This is reflected in a maximum CO content in the waste gases of between 50/60%, equivalent to an air factor of 0.2/0.3; the Japanese adhere more strictly to the designed 0.1 air factor with maximum CO levels of about 80/90% and average CO values in the recovered gas of 60/70%.

The flux additions and lance height techniques in operation at the Japanese plants follow established BOS operating practices and there was no indication that this had been modified in any way as a result of the gas collection process. Iron ore coolant is added via a trickle feed system in accordance with normal O.G. operating practice.

5.4.2 O G Gas Recovery System

The arrangement of a typical gas recovery installation is shown in Fig.5.3.(167) Prior to recovery being initiated the waste gas flows through the ID fan and the three way valve and is burnt at the flare stack. When gas recovery conditions are established the water sealed check valve, provided between the gas holder and the three way valve to prevent reverse gas flow from the holder, automatically opens to the gas holder. Gas recovery is terminated automatically, either normally at the end of blow or during blow under emergency conditions whenever any one of the gas recovery inter-locks is not satisfied. On termination the three way valve first changes over to divert gas flow to the stack followed by closure of the water sealed check valve. The purpose of the by-pass valve installed between ID fan and the flare stack is to provide an alternative path for venting gas to atmosphere in the event that the gas flow cannot be switched from recovery to flaring due to a failure of the three way valve. The recovered BOS gas in the gas holder is boosted up to distribution pressure by fans located adjacent to the gas holder, and is utilised either unblended or mixed with BF gas or CO gas.

The condition which is most significant in determining the initiation and termination of recovery is carbon monoxide content of the waste gas. In general a higher setting of CO value for recovery results in the recovered gas being of a higher calorific value, but the total energy recovered per tonne of steel is lower (other factors being constant).

5.4.3 Technical Considerations for Recovery

The main factors determining the rate at which BOS gas can be collected and its calorific energy value are:

Vessel Size:	Fixed for any given plant.
Carbon removed:	Dependent on hot metal carbon and end-of-blow carbon specification.

HM/Scrap Ratio:	High scrap additons reduce recovered energy per cast for a given vessel size.
BOS Operating Practice:	Degree of suppressed combustion mainly governs calorific value and volume of gas collected. The timing of flux control also affects collected gas volumes to some extent.
Collection Efficiency:	Dependent on gas collection threashold levels, equipment reliability and user plant management.

Fig. 5.1 illustrates the typical pattern of CO and CO_2 waste gas emissions during a BOS steelmaking heat. The quantity of CO available for collection is governed initially by the rate of carbon oxidation and secondly by the gap between the movable skirt and vessel mouth which determines the volume of excess air sucked into the hood to burn a proportion of CO to CO_2. Lackenby BOS Plant has been designed and proven to operate with an excess air factor $\lambda = 0.1$.

In order to ascertain the potential for gas recovery at Lackenby, Gragetown Laboratories condicted a series of tests covering a total of 29 heats on "A" and "B" vessels. The results are summarised in Table 5.2. It must be emphasised that energy recovered is greatly influenced by the amount of excess air entrained at the gap between the vessel mouth and hood skirt, the position of which is controlled by the steelmaker. In the absence of any gas recovery plant at Lackenby there is clearly no incentive to operate at the design air factor of $\lambda = 0.1$ and the results in Table 5.2

reflect this situation. Gas analysis records have therefore had to be interpreted to establish the potential gas collection performance at design conditions as indicated in the right hand column.

The above figures have been plotted in Fig. 5.4 to show the dependency of energy recovered on excess air factors. For comparison, the Japanese performance figures referred to earlier have also been included. It should be appreciated that the lower the value of λ the higher becomes the gas calorific value for smaller collected volumes giving the most cost effective returns for a given size of gas holder.

Following discussions with BOS Management, a practical operating standard of $\lambda = 0.15$ was agreed which gives a recovered energy level of 590 MJ/tonne of tapped steel on the basis of Redcar iron and a 75% HM to scrap ratio. The full potential for energy recovery is 625 MJ/tonne at $\lambda = 0.1$ for the same operating conditions.

5.4.4 Description of Proposed Scheme

The scheme for recovering gas from any two of three operating vessels at Lackenby and its storage and associated distribution system is shown in Figure. 5.5.

Considering only one vessel stream, during both the initial and final stages of a blow converter gas will exhaust via a new three-way valve to the existing flare stack. When all conditions for gas recovery are satisfied including gas composition, flowrate, gasholder capacity, valve positions and permissive time within the blow, the water seal check valve will then open automatically followed by changeover of the three-way valve to divert the flow from the flare stack to the gas holder.

If any of the essential gas recovery conditions deviate from the operational requirements during the blowing period, or if an emergency arises the system will automatically change from recovery to flaring for the remainder of the blow. This will be achieved by reversing the operating sequence of the three-way and water seal check valves, the latter preventing gas flow back from the holder. As a safeguard against possible malfunction of the three-way valve, a gas by-pass valve provides an alternative route between the ID fan and flare stack.

Two goggle valves and a U-type water seal complete with dump tank will be installed in each vessel's gas recovery line to facilitate safe access for maintenance to the water seal check valve and other equipment upstream.

The common gas main will be sized to accommodate two vessels blowing simultaneously. Closer to the gas holder a blow off pipe is provided for nitrogen purging when necessary or for release of gas from the holder if access is required. Primary isolation on the inlet side of the holder is provided by a U-type water seal backed by a single goggle valve.

The average dust loading of BOS gas over a collection period has been measured by Grangetown Laboratories at approximately 85 mg/Nm^3. However, long term operation at this level is unacceptable in terms of dust drop out in ducts and burner maintenance at user plants downstream of the gas holder. A single wet precipitator has therefore been included to clean gas to 15 mg/Nm^3. The precipitator can be isolated for maintenance by two goggle valves. A by-pass route has also been provided to ensure continuity of supply albeit for relatively short periods of higher dust loading.

A common duct connects two identical boosters via goggle valves, each booster being rated at 1 200 Nm^3/minute. While one booster will be quite sufficient to supply user requirements on most occasions leaving a live standby, both boosters could run in parallel to increase flow rates to approximately 1 800 Nm^3/minute which roughly matches the input rate to the holder with one vessel blowing.

The gas cleaning and booster plant will be provided with U-type water seals at the inlet and outlet sides as the means of primary isolation. Reverse gas flows will be prevented by the status of goggle valves. Boosted gas will feed into the existing Cleveland/Lackenby gas main and apart from a flowmeter and gas control valve in the Cleveland route, no other user plant modifications are necessary.

5.4.5 Gas Recovery Equipment

A. Mechanical Equipment

i) I.D Fans: New fans will be installed to provide the increased pressure requirement from 65 in WG to 85 in WG for gas recovery. Measurements taken on plant have demonstrated that the present fans are slightly oversized for their duty thereby enabling the existing motors to be retained.

ii) Three-way Valves: The valve comprises two mechanically coupled butterfly valves actuated by a common pneumatic cylinder. The valves are fitted with rubber seals which sre flushed with water to prevent dust build up. Each butterfly is fitted with limit swithces at both the open and closed position, these being used in conjuction with a timer to check the valve has safely operated.

iii) Water Sealed Check Valves: The function of water-sealed check valve is to prevent reverse flow of gas from gas holder into the OG. It provides a back up to the three-way valve as well as providing a rapid means of positive isolation.

The rotary type is recommended in preference to bell type because of its faster response time and lower pressure drop (zero) across the valve.

iv) By-pass Valve: The by-pass valve installed will be of the butterfly type with pneumatic actuation and water washed rubber seats.

v) <u>V-Type Seal Valves</u>: To permit the maintenance of check valves and gas holders, V-type seals are fitted downstream to each check valve and at both input and output of the gas holder. These seals are constantly fed with a small flow of water and continuously overflow when in operation to ensure safe operation.

vi) <u>Gas Holder</u>: A few years ago there were some doubts about the quality and effectiveness of the rubber seal in the Wiggins holder, and wet holders were preferred. More recently, as confidence has grown in the seal, the situation has been reversed and the current preference is for dry seals. A dry holder also eliminates the risk of damage by frost and any control problems caused by pressure changes as various lifts in the wet holder are used.

To ensure adequate mixing of gas to minimise variations in calorific value, separate inlet and outlet connections are provided.

The choice of size is dependent on local conditions and must take account of the intermittent nature of production and the pattern of consumption. There must be sufficient storage to give time for alternative fuels to be introduced in the event of a sudden stop in the BOS gas recovery system.

A 85,000 m^3 Wiggin holder was thus recommended.

vii) Precipitator : One wet electrostatic precipitator of Peabody Holmes manufacture is proposed on the outlet side of the gas holder with provision for a second unit which could be interposed in the bypass ductwork should the need arise. The precipitator will be complete with water flushing and effluent handling systems.

Surveys were carried out by BSC and Lodge Cottrell on the suitability of Hartlepool precipitators due for demolition but costs for dismantling, refurbishment and re-erection were almost identical to the installation of new equipment.

viii) Booster Fans: Two booster fans, one standby, are located after the precipitator(s) each capable of feeding gas at 1200 m^3/hr at the required gas main pressure. The fans are of single inlet, backward bladed, centrifugal type having a running speed of 985 rpm.

B. Electrical Equipment:

Electrical equipment is located in two main centres:

i) The existing west sub-station is to be extended to house motor controls and sequencing equipment associated with safety interlocks and the main gas valves in the ID fan area.

ii) A new gas control room and power house will accommodate the motor and sequence controls for the gas holder, precipitator and gas booster station. The equipment room and transformer unit will be sized to accommodate a second precipitator if required.

In addition, field mounted equipment for lighting, motors, limit switches will be required all conforming to the explosive atmosphere regulations. Apart from these special requirements, all other equipment will be of identical or similar design to that familiar to plant engineering personnel.

C. Maintenance Considerations:

The heavy items of equipment require very little maintenance but it is strongly stressed that in order to maintain satisfactory gas recovery the maintenance of the following units must be of a very high standard.

(i) gas analysers;

(ii) safety limit switches;

(iii) solenoid valves.

5.4.6 Proposed Use of Recovered BOS Gas

At "Teesside Load" producing the design BOS gas collection levels of 590 MJ/tonne steel the average rate of energy production is 232,000 MJ/hour. Table 5.3 presents the works energy balance for producing/consuming plants for different strategic cases ranging from full Cleveland operation (without Clay Lane) through to Lackenby only consumers. At full capacity levels of steel making the average rate of energy production is 343,000 MJ/hour.

From the test data obtained on the Lackenby steelplant, results indicate that the calorific value of gas collected will be in the range of 6 to 9 MJ/Nm^3. Even at 6 MJ/Nm^3 the CV is such that the gas could be used directly for heating steel in either soaking pits or reheating furnaces, for blast furnace stove heating, as a boiler fuel or as an enriching fuel for blast furnace gas. The gas is sulphur free, and the waste gas from combustion of BOS gas is low in water vapour and has a low dew point.

Two particular requirements for BOS gas recovery depending on future operating levels. The first, resulting from high output levels is to meet coke oven gas deficits which would otherwise have to be met by purchased premium fuels such as natural gas, LPG or fuel oil. And the second at lower output levels i.e."Teesside Plant Load", as a replacement for fuel oil, either directly as on Cleveland boilers, or indirectly by releasing coke oven gas on an application to replace fuel oil on another.

It is porposed that recovered BOS gas initially will be utilised to replace oil consumption at the Lackenby No.1 Primary Mill soaking pits and at Cleveland Power Station. In case presented in Table 5.3 the increased BOS gas make of 111,000 MJ/hour over "Teesside Load" can be fully consumed except during weekend operation of the

Lackenby No 1 and 2 Mills, when it will be used for power generation. However, at full capacity levels of operation it is assumed that all plants will be geared to higher throughputs and therefore increased hourly energy demands.

The proposed BOS gas "User" plants at Lackenby and Cleveland are diagramatically shown in Fig. 5.6. Only if the entire Cleveland Works closed down would gas be fed to No.3 Primary Mill which will require some additional money to be spent on modifications to the ductwork, recuperators and instrumentation system.

5.4.7 Economic Assessment

The justification for BOS recovery depends on the reduction in purchased energy costs set against the costs of Installation and Operation.

Japanese experience shows that energy savings of 600 MJ/tonne steel produced is possible. However under the present operating conditions where the total plant production is 3.0 million tonne per annum (Mtpa) against the designed production of 4.5 Mtpa data collected from five heats in the acceptance trials on 'Vessel A' and 'Vessel B' indicate that an average energy recovery of 508 MJ/tonne and 566 MJ/tonne respectively is possible. Since the vessels are to operate on an alternative basis, this shows an average energy recovery of 537 MJ/tonne.

This lower figure is also attributed to larger combustion factor at Lackenby because of a larger gap between the vessel mouth and skirt, which possibly can be rectified.

A. Capital Costs:

Turnkey BOS Gas Recovery Scheme:

Cost included capital, duct-work civil and distribution system (Ref: DAVY ASHMORE)[168]	£11.7 M

B. Operating Costs:

Power cost to provide extra lift & fan to force gas into gas-holder	£ 0.20 M
Power cost for distribution system	£ 0.06 M
Labour costs	£ 0.05 M
Maintenance	£ 0.20 M
Services	£ 0.01 M
Total Operating Costs	£ 0.52 M

C. Fuel Savings:

Assuming BOS gas to replace the purchased fuel oil,

BOS Gas Price = £ 2.3/GJ

Plant Output - Mtpa	3.0		4.5	
Energy Savings/t - MJ/t	537	600	537	600
Total Energy Savings - MGJ	1.61	1.80	2.42	2.70
Cost of Energy Savings - £M	3.70	4.14	5.56	6.21
Net Savings - £M	2.18	3.62	5.02	5.69
DCF Return on investment over 15 years depreciation period - % p.a.	18	29	40	45

Therefore without any modification to the BOS plant (i.e. hood and skirt) and the plant operating at 3.0 Mtpa capacity a net savings in energy recovered from the BOS gas collection can be £2.2 million per year giving a DCF return of 18%. However with the market upturn, the plant operating on the designed capacity of 4.5 Mtpa, energy savings of £5.7 million giving a pay back period of approximately two years can be achieved.

5.5 CONCLUSIONS

From the studies carried out, the following conclusions can be drawn:

1. BOS gas recovery offers a potential energy saving scheme to the steel industry. The system is tested and proven in the Japanese steel industry.

2. Minimum combustion system (OG - system) offers the best possible method of BOS gas recovery. The collection of gas represents a convenient, low cost means of securing a flexible energy supply for the works.

3. Application of OG - Gas Recovery System at Lackenby steelworks, when running at design capacity of 4.5 Mtpa offers a net energy savings of £5.0 million per year at 1980 costs for a capital investment of £11.7 million giving a payback period of just over two years.

   A minor enhancement of the plant reducing the gap between the vessel mouth and skirt can offer an additional energy saving of £0.7 million per year.

4. Under current operating conditions with plant production of £3.0 Mtpa, an energy sagings of £2.2 million per year giving a DCF return of 18% can be achieved.

ENERGY LOSSES FROM THE BOS STEEL PROCESS

(Basis: 1 Tonne of steel)

	MJ	Percentage
Iron losses due to vaporisation and mechanical losses	850	25
Losses associated with evolved BOS gases	840	25
Unaccounted losses (sequestered energy in O_2, mech. energy)	740	22
Radiation and convection losses (estimate)	415	12
Cooling water losses	285	8.5
Enthalpy loss in BOS slag	250	7.5
TOTAL	3380	100.0

ENERGY LOSSES FROM BOS STEEL PROCESS

TABLE 5.1

BOS GAS RECOVERY TEST RESULTS AT LACKENBY STEELWORKS

TEST RESULTS PARAMETER	RANGE	OVERALL RANGE	PERFORMANCE $\lambda = 0.1$
Air Factor	0.09-0.30	0.20	0.1
Carbon Monoxide (%)	42.1%-71.0%	54.0%	69.3 % CO
Calorific Value (MJ/Nm^3)	5.32-7.99	6.82	8.75 MJ/Nm^3
Recovered Volume (Nm^3/tonne steel)	67.4-95.6	81.6	71.4 Nm^3/tonne
Recovered Energy (MJ/tonne)	417-696	557	625 MJ/tonne

BOS GAS RECOVERY: TEST RESULTS AT LACKENBY

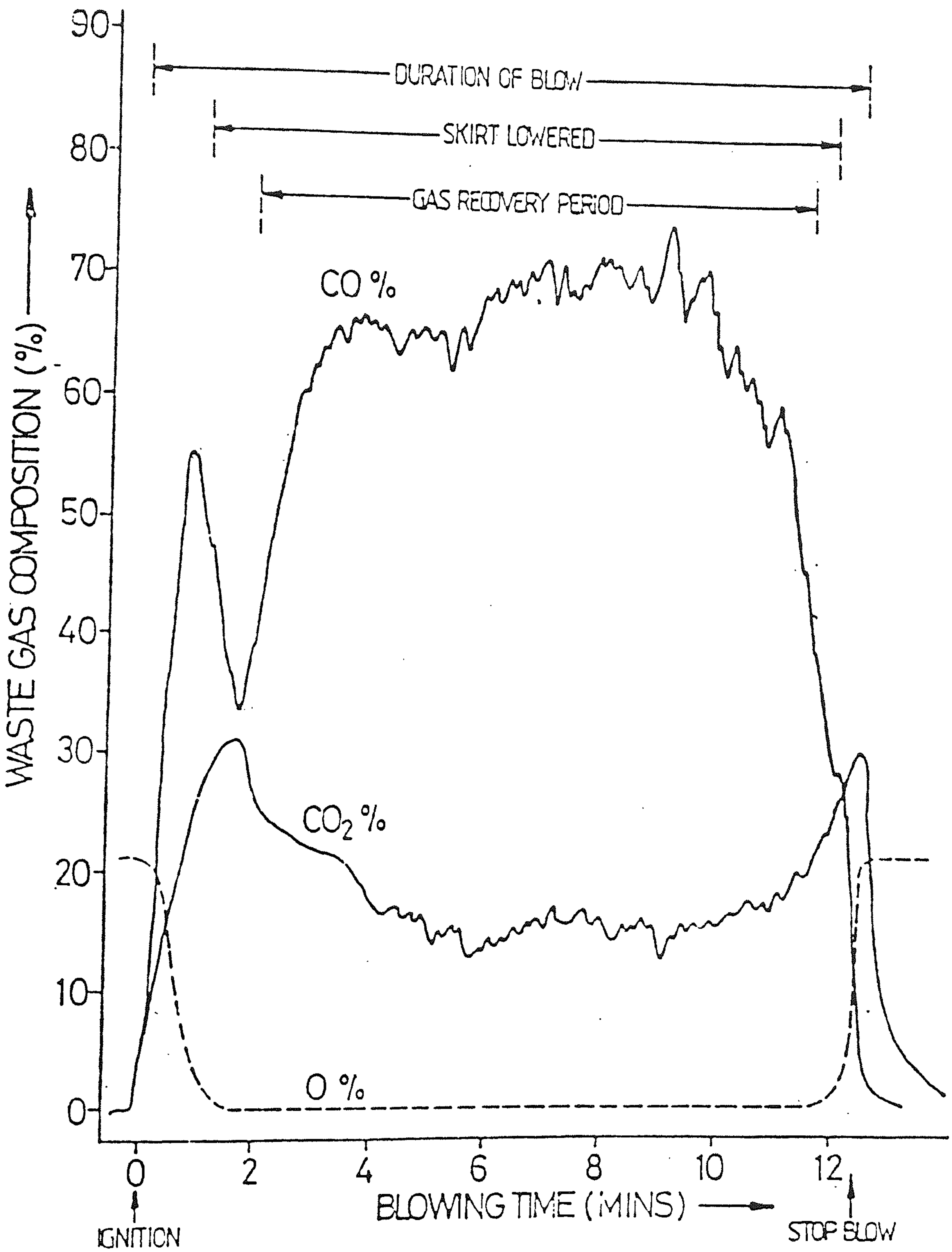
TABLE 5.2

Consumption and Energy Balance / Case Options	BASE CASE (No BOS Gas)		CASE 1 (with BOS Gas)	
	(MJ/h. x 10^3)		(MJ/h. x 10^3)	
	Wkday	Wkend	Wkday	Wkend
ENERGY CONSUMED:				
Cleveland				
Ferro stoves (GAS ONLY)	105	105	105	105
No.3 Primary Mill (GAS ONLY)	73	18	73	18
Boilers (GAS)	110	165	219	363
Boilers (OIL)	446	391	337	194
CLEVELAND ENERGY CONSUMPTION =	734	679	734	680
Lackenby				
No.1 Primary Mill (GAS)	-	-	118	30
No.1 Primary Mill (OIL)	118	30	-	-
No.2 Primary Mill (GAS)	-	-	-	-
No.2 Primary Mill (OIL)	244	244	244	244
LACKENBY ENERGY CONSUMPTION =	362	274	362	274
TOTAL ENERGY CONSUMED	1096	953	1096	954
ENERGY SUPPLIES				
Ferro Gas Make	288	288	288	288
BALANCE =	808	665	808	666
BOS Gas Make	NIL	NIL	232	232
BALANCE (OIL REQUIREMENTS) =	808	665	576	434

CASE NOTE:
Assumes CLEVELAND (2 x Ferro BF's (2700 tpw) +
Boilers (20 MWhr) +
No. 3 Mill +
LACKENBY (No.1 Mill + No.2 Mill)

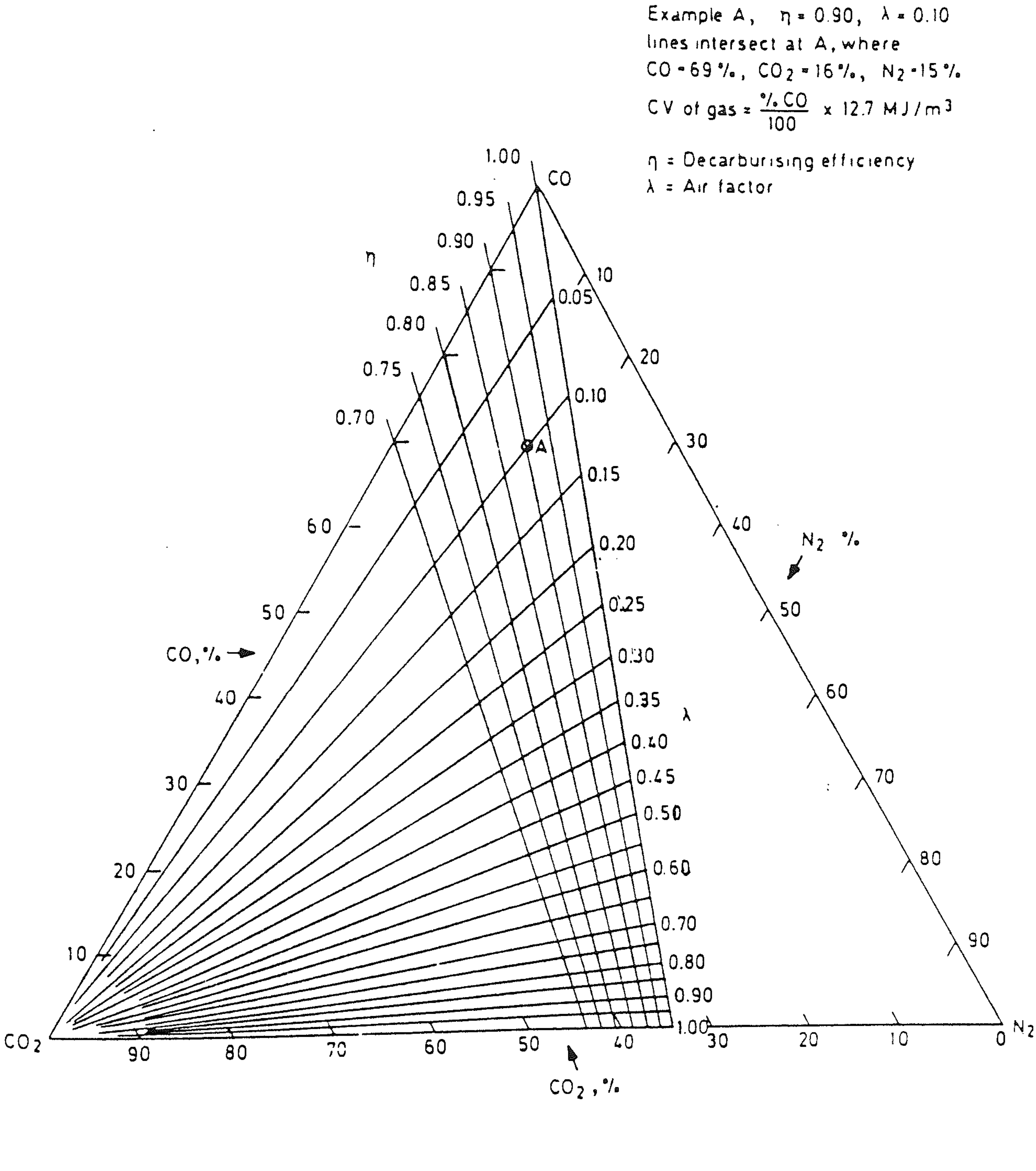
BOS OPERATING STRATEGY SHOWING OIL SAVINGS

TABLE 5.3



TYPICAL PATTERN OF BOS WASTE GAS GENERATION
AGAINST BLOWING PRACTICE

FIGURE 5.1



BOS GAS ANALYSIS

FIGURE 5.2

Aston University

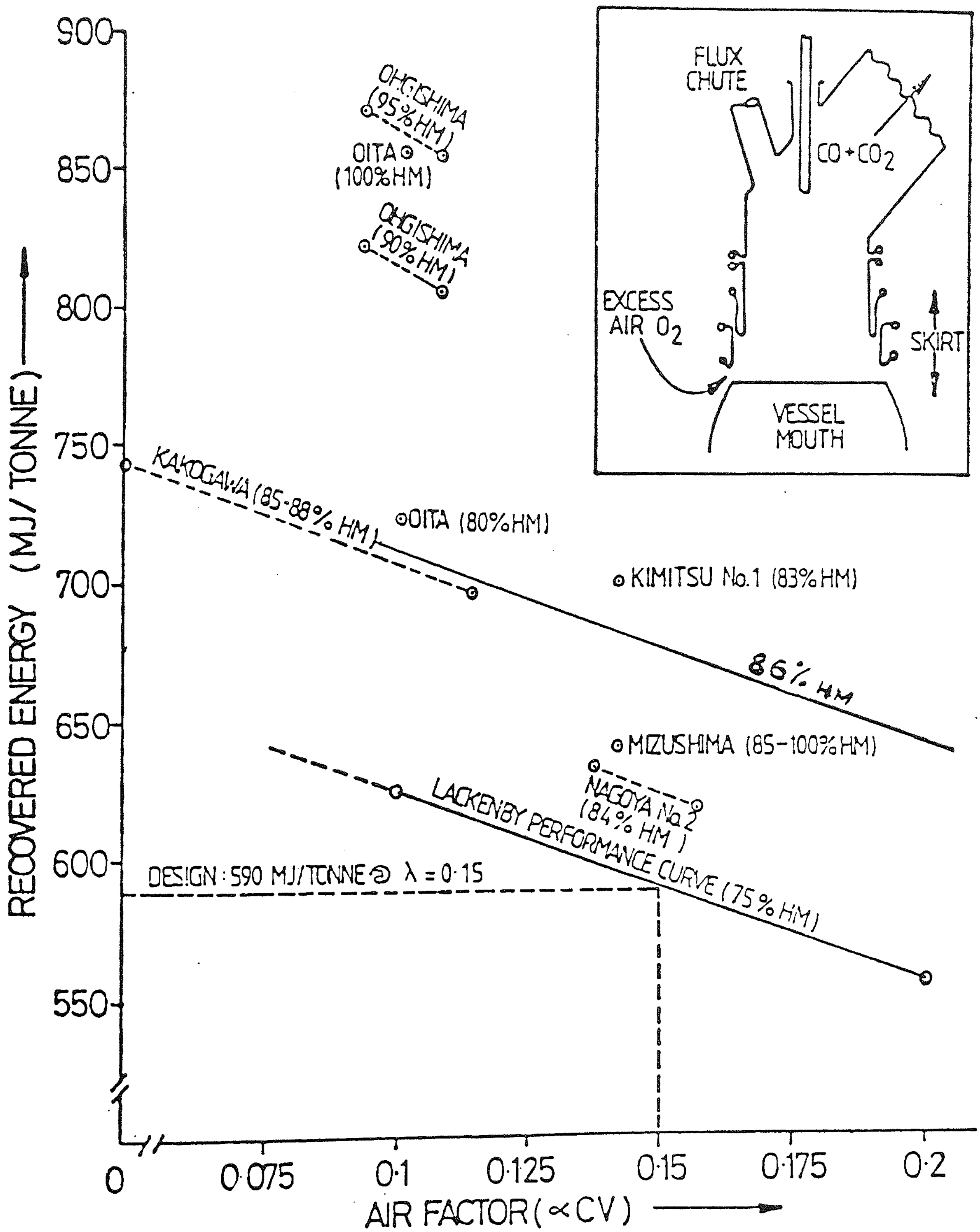
Illustration removed for copyright restrictions

Aston University

Illustration removed for copyright restrictions

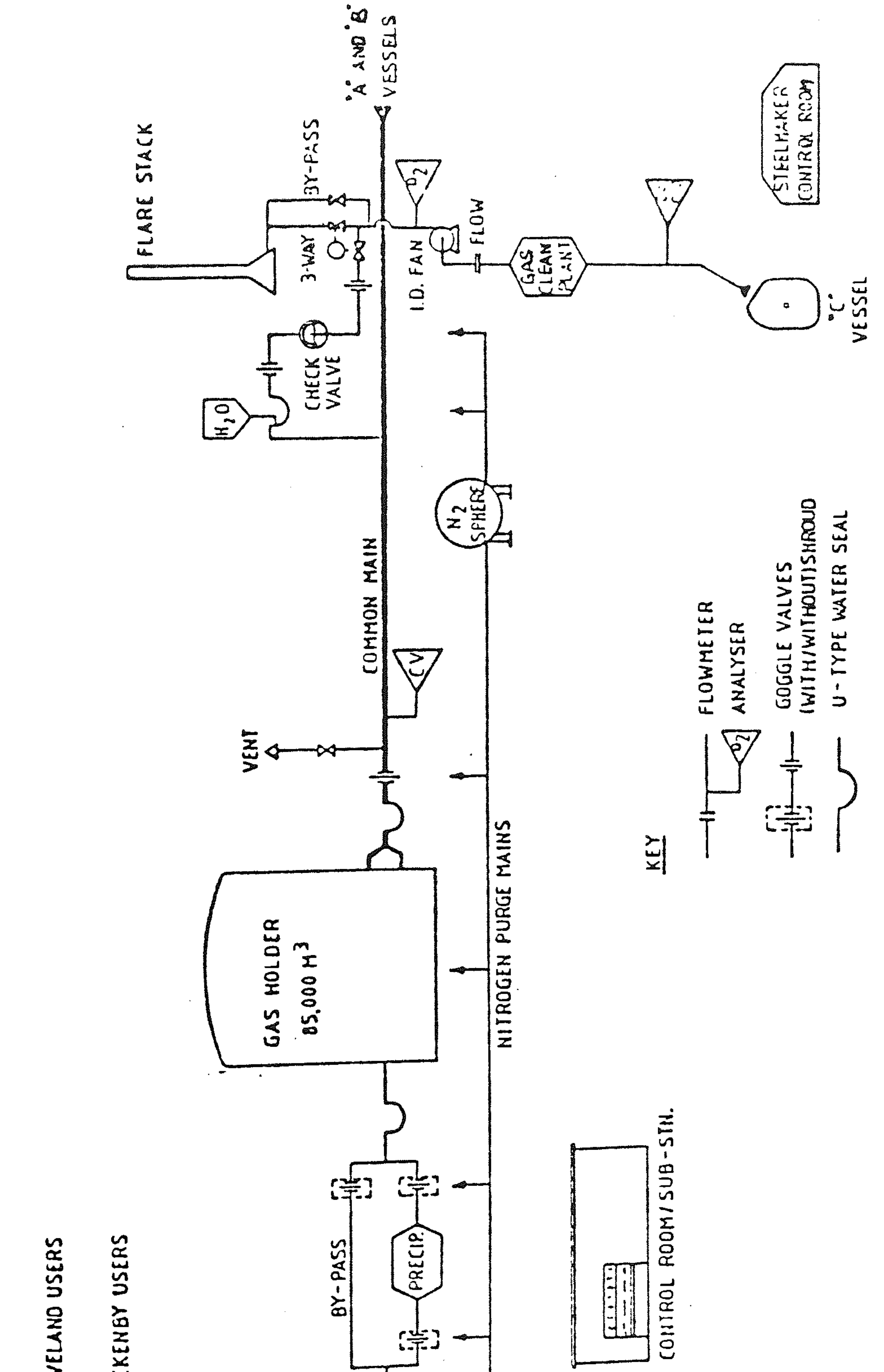
GENERAL LAYOUT OF THE OG SYSTEM

FIGURE 5.3



EFFECT OF AIR FACTOR (λ) ON RECOVERED ENERGY

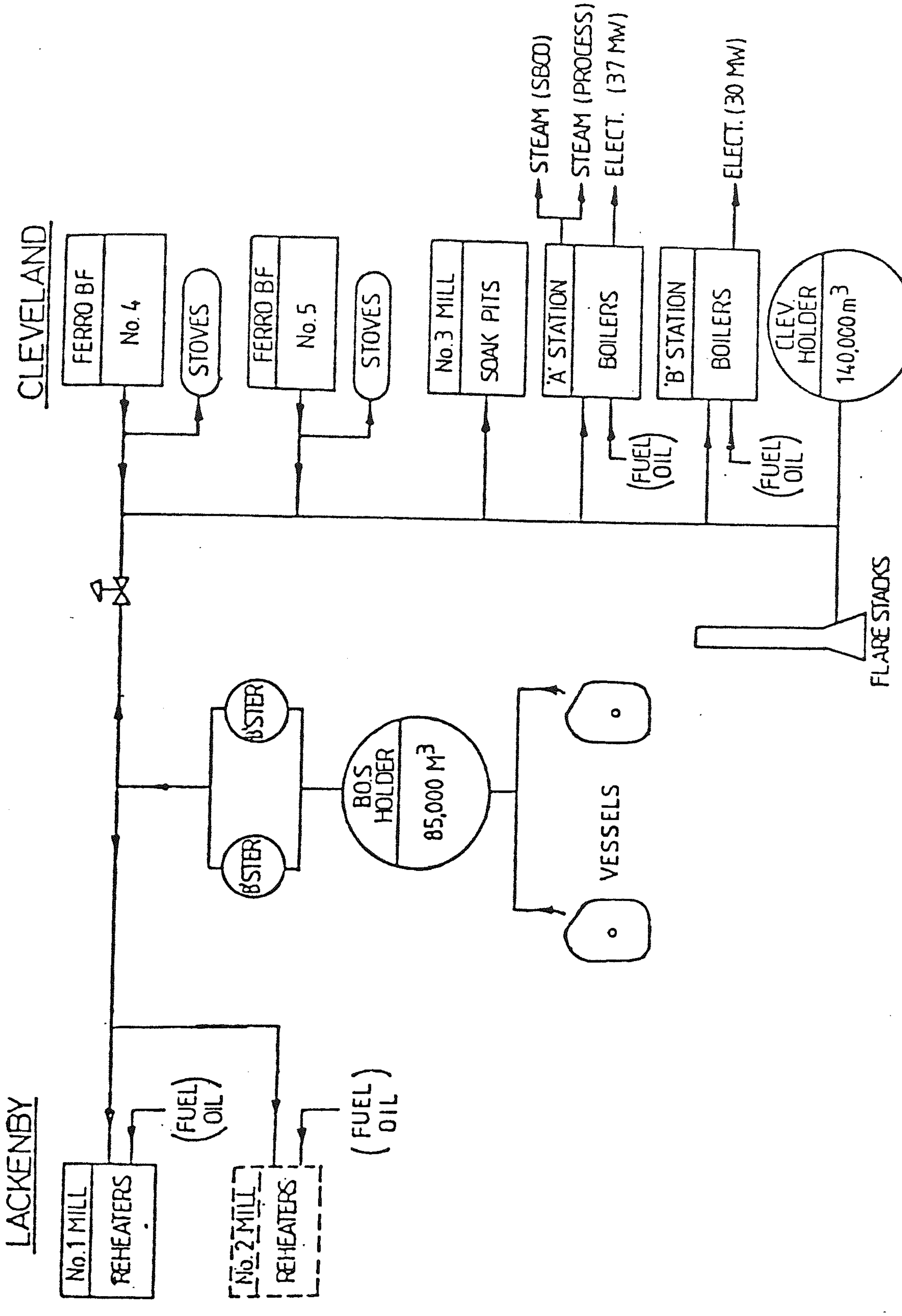
TABLE 5.4



Lackenby BOS gas collection, storage and distribution diagram

LACKENBY BOS GAS COLLECTION
STORAGE & DISTRIBUTION DIAGRAM

FIGURE 5.5



BOS gas "user" plants at Lackenby and Cleveland

BOS GAS USER PLANTS AT LACKENBY AND CLEVELAND

FIGURE 5.6

CHAPTER 6

ENERGY RECOVERY FROM WASTE GASES

6.1 INTRODUCTION

In general the iron and steel industry uses energy to raise materials to temperatures suitable for chemical reactions and to permit mechanical working. Thring[169] noted that there are three approaches in material manufacture:

(i) the manufacture of material being the primary objective without regards for energy or other inputs.

(ii) the process itself remains unquestioned; however the need to minimise energy is considered.

(iii) the whole problem is re-examined and the objective to achieve the desired production with a minimum of energy expenditure is considered.

With the escalating energy costs and low demands the first approach in manufacturing industry has disappeared. Howevever with the financial restraints still remaining, the second approach is still persistent with us, though effort to achieve three is being pursued.

As an energy intensive industry, the BSC is aware of the need to maximise the utilisation of its arising gases, such as blast furnace gas and coke oven gas, in addition to achieving a reduction in energy consumption by improved process control.

There are three sources of energy rejection which are believed to offer commercially viable recovery opportunities in the short term and at low cost. These sources are gas bleed losses; inadequate pipe insulation and steam leak losses; and improved recuperation on soaking pits and reheating furnaces.

The magnitude of bleed gas losses is a function of three main problems, namely, that of maintaining adequate standards of gas cleaning plant and consuming plant, that of accommodating the widely fluctuating gas make into the works energy balance, and that of fundamental changes in the works energy balance due to commissioning of new or closure of old plant. In the latter case timing of closure and phasing in of new plant is very difficult to accomplish without creating periods of energy imbalance.

Some degree of bleed gas loss is unavoidable due to temporary fuel supply demand imbalances and the prohibitive cost of additional gas storage, but it is believed that there is room for improvement in flexibility of fuel usage within works.

The costs of inadequate pipe lagging thickness and unrepaired steam leaks is constantly increasing and high returns are likely to be available from modest expenditure in this area. As an example a single steam leak through a 5 mm diameter hole can cost £10,000 per annum on a medium pressure line.

However, as illustrated in (Fig. 6.1) as much as 19% of the total energy input is dissipated to atmosphere through waste gases offering a potential energy recovery source. Therefore increased emphasis is currently being placed by the Corporation to harness some of this waste heat.(170-171)

This chapter outlines areas and quality of waste heat available and suggests some novel methods of recovery. A case study assessing the potential advantages of various recuperation systems on No.9 soaking Pit at Ravenscraig Steelworks has been carried out.

6.2 ENERGY RECOVERY FROM LOW TEMPERATURE GASES

The major problems in attempting to remove useful heat from such low temperature sources as coke oven waste gas, sinter plant gases and blast furnace gases are five fold, namely:

i) the need for very large heat transfer surface areas to compensate for the low thermal driving force between source and likely receiving temperature,

ii) the loss of bouyancy with falling temperatures calling for increased use of compensatory fan power. Many systems exhaust currently employ natural draughting,

iii) the likelihood of severe corrosion of stacks and exchangers by cooling gases already close to their acid dewpoint,

iv) the problems of fouling of exchanger surfaces with soot and dusts, and with liquid phase deposits such as tars and light oils,

v) that any heat potentially recoverable would be at low temperature, and would therefore, be of very limited use.

In the light of the complexity and interaction of the above problems, low temperature gas sources offer little if any real opportunity for recovery, even if considerable R and D effort were to be expended. In addition it is generally accepted that the cost of energy recovery increases with decreasing temperature.[172]

One limited exception may be in the application of refrigerant turbines (Freon) to clean low temperature gases at temperatures in excess of 200°C. Various workers, in the USA, UK and in Japan, have suggested that low grade waste heat (sensible heat in gases or liquids) could exchange heat with a closed loop Freon cycle, whereby the freon is evaporated in a heat exchanger and the expanded vapour is used to drive a turbine for the production of electric power.[173] Cold water may be used to condense and cool the recycling Freon. The major advantage of Freon relative to steam as a cycle fluid appears to be in the greater reduction in size possible with the Freon system. However, problems exist in the large size of the heat exchanger between waste gas and Freon and in the Freon cycle condenser, and in fouling and corrosion of the waste gas heat exchanger. Considerable design and development work is needed to produce a practical system that may be considered for the steel industry. A system currently being tested by IHI in Japan has resulted in a package unit of 3,8 MW.[174]

Theoretical consideration has also been given to the development of novel heat exchangers employing fluidised beds and heat pipes either alone or in combination. The general conclusion is that no practical use could be made of these techniques because of a number of fundamental design problems.

In the case of the heat pipe, the prime advantage of high thermal conductivity with small temperature differential between evaporator and condenser, plus the relatively narrow limits in working temperature of particular heat pipe systems are incompatible with the wide temperature spectrum encountered with soaking pits and to a lesser extent with reheating furnaces from which the major portion of high temperature gases are lost. In addition, at the lower end of the temperature range, the process of transferring heat from the gases into the heat pipe is still governed by convection across the gaseous boundary layer around the evaporator end of the pipe thereby calling for excessively large transfer surface areas.[175]

In the case of fluidised beds the obvious advantage of greatly enhanced heat transfer between the gas and heat transfer tubes suspended in the bed is negated by the strict gas flow velocity limitations for satisfactory fluidisation. In general, prohibitively large bed areas are needed because of the need to minimise bed depth in order to keep pressure drops (and hence power) to a reasonable level.[176-177] Furthermore, the operational difficulties of running fluid bed heat exchangers intermittently and with widely varying gas flows are considerable.

6.3 ENERGY RECOVERY FROM HIGH TEMPERATURE GASES

Most of the high temperature gases are generated in the soaking pits and reheating furnaces. In typical current practices the waste gases leaving these furnaces are at temperatures in the region of 900°C to 1200°C. Thus these flue gases offer a major and readily accessible energy recovery source.

A well established method of recovering this energy is through the use of waste heat boilers from which steam can be generated for the works' requirements.[178-181] However, in a number of cases because of the plant layout, installation of such an equipment within the vicinity of the furnaces is not possible. Thus the waste heat boiler needs to be installed either a distance away from the furnaces or above the furnaces. In both these cases the capital cost for installation becomes prohibitive. Additional costs of distribution further adds to the costs of the scheme.

In addition, effective use of the generated steam needs to be found to make the recovery scheme viable. In some steel plants, the overall energy balance is such that substantial quantities of steam cannot be absorbed internally without making internally generated fuel gases e.g. BF gas and CO gas, redundant at the boiler plant or elsewhere in the system.

In the light of this, energy recovery in the form of combustion air preheat at the furnace may offer better incentives.[182-184]

6.4 ENERGY RECUPERATION

Taking a general case of a fossil fuel fired furnace (soaking pit/ reheating furnace) given a fuel with a calorific value of cv, the fuel consumed Wf per unit time produces a quantity of heat Wf x cv. This heat is then expanded by producing

(i) heat to furnace stock Hs

(ii) heat loss by radiation Hr

(iii) heat loss to waste gases Hw

$$Wf \times cv = Hs + Hr + Hw \qquad \text{—— 6.1}$$

and

$$Hw = Wf \times Qf \times hg \qquad \text{—— 6.2}$$

$$Qf = \eta (1 + x) + 1$$

$$hg = \int_{Tm}^{Tg} Cpg \; dT$$

6.4.1 Energy Recuperation : Steam Generation

When a portion of the flue gas heat available, say dHw is utilised (clear of the furnace system generating it) for steam raising in a waste heat boiler, etc., then the full value of fuel savings available is given by

$$dWf(\alpha) = \frac{dHs + dHr}{cv} \qquad \text{—— 6.3}$$

6.4.2 Energy Recuperation : Air Preheat

In comparison the use of Hw for air preheat can be considered as follows:

Equation (6.1) shows Wf. cv as the heat release from the combustion of fossil fuel.

$$\text{i.e. } Wf * cv = Hs + Hr + Hw$$

$$= Hs + Wf * Qf * hg + Hr$$

$$= Hs + Wf\left[n(1+x) + 1\right]hg + Hr \quad \text{— 6.4}$$

Heat recovered for air preheat

$$= Wf' * cv + n(1+x)\, Wf' * ha$$

$$= Hs + Wf'\left[n(1+x) + 1\right]hg + Hr \quad \text{— 6.5}$$

where ha', the sensible heat of combustion air is given by

$$ha' = ha + \int_{Tm}^{Ta} Cpa * Tm * dr$$

Due to combustion air preheat, to maintain a constant value of Hs (the heat to the furnace stock), the fuel rate will be cut back.

$$Hs = Wf * cv - Wf\left[n\ (1+x) + 1\right]hg - Hr$$

Equating equation 6.5, the reduced fuel rate Wf' is given by

$$Wf' = \frac{Wf\left\{cv - \left[n(1+x) + 1\right]\ hg\right\}}{cv + \left[n(1+x)\ ha\right] - \left[n(1+x)+1\right]\ hg} \quad \text{— 6.6}$$

Now the proportion of flue gas heat recovered

$$dHs = Wf'\left[n\ (1+x)\right]\ ha - dHr$$

$$\text{or} \quad dHs + dHr = Wf' * n\ (1+x) * ha$$

Substituting in equation 6.5 the fuel savings dWf for air preheat may be stated as follows

$$dWf\ (\alpha) = Wf - Wf' = \frac{dHs + dHr}{cv - \left[n\ (1+x) + 1\right]\ hg} \quad \text{— 6.7}$$

Comparing equation 6.3 and 6.7 it can be concluded that combustion air preheat offers the maximum fuel savings in any waste heat recovery system. This is because heat fed back into the system via combustion air carries otherwise heat rejected in hot flue gases.

$$\frac{\text{Fuel saved/unit heat recovery for air preheat}}{\text{Fuel saved/unit heat recovery in other systems}} = \frac{cv}{cv - [n(1+x)+1]\ hg} > 1$$

Using equation 6.6 a fuel economy chart is drawn as shown in Figure 6.2

$$F = \frac{Wf'}{Wf} = \frac{cv - [n(1+x)+1]\ hg}{cv + [n(1+x)]^{*}\ ha - [n(1+x)+1] hg}$$

$$= \frac{cv - [n(1+x)+1]\ hg1}{cv - [n(1+x)+1]\ hg2}$$

where suffix 1 refers to the flue gas leaving the furnace flue

and suffix 2 refers to the flue gas at exit from combustion air preheater.

Therefore advantages of Combustion Air Preheat can be defined as:

i) Enhanced Fuel Economy: As clearly demonstrated above heat recovery via air preheat offers an enhanced fuel recovery system when compared with any other heat recovery system.

ii) Improved Combustion: Use of preheated combustion air produces a hotter flame and thus less unformed residual carbon and increased fuel efficiency.

iii) Economic Installation: Since the system requires no additional utilisation and distribution system, the installation cost can be minimum.

iv) Improved Environment: The flue gas temperature entering the atmosphere through the utilisation of a waste heat recovery system is substantially reduced thus eliminating the undesirable energy input to the surroundings.

v) Noise Reduction: A very considerable reduction in noise level can be achieved through the installation of a heat recovery system on the furnace which previously operated on natural draught.

6.5 RECUPERATIVE SYSTEMS

In many current practices the waste gases leaving the furnace at high temperatures are passed through radiative or convective metallic recuperators to preheat incoming combustion air to temperatures around $400^{o}C$, although $500^{o}C$ is achieved at a few installations.[186-187] Whilst making significant savings in furnace fuel requirements these metallic recuperators frequently require dilution of the incoming waste gases with cold air in order to protect the metal surfaces from overheating and, in addition, they still exhaust the waste gases at comparatively high temperatures in the region of $400^{o}C$ to $800^{o}C$. Furthermore, it is established that most of the existing type of metallic recuperators have very high (30%) air leakage in the system.

6.5.1 BSC Ceramic Recuperator

In order to improve heat recovery from these major sources and having established the limitation of the existing metallic recuperators, the Corporate Laboratories of British Steel Corporation embarked on the development of a novel ceramic heat exchanger to provide higher air preheat temperatures to give direct fuel savings.[188-190]

An intensive development programme was initiated to determine suitable tube and seal materials. The selection of the tube material was based on the following criteria:

- (i) Strength to withstand high temperature
- (ii) High resistance to thermal shock
- (iii) Low permeability
- (iv) Good thermal conductivity
- (v) High resistance to abrasion and corrosion
- (vi) Ease of manufacture and availability

From the design study silicon-carbide tubes were selected. A novel seal design was developed to provide low air leakage, high thermal expansion and a long seal life.

Following extensive design, selection and testing of components, and the development of a computer model of heat transfer, a first prototype was build on a 160 tonne oil-fired soaking pit at Llanwern Works in November 1973 (Fig.6.2). This quickly achieved its design full flow air preheat of 480°C. Subsequent addition of ceramic cruciform inserts to the tubes increased the preheat at full flow to 570°C. The air leakage from the recuperator was about 3% at full flow.[191]

The experience of the Llanwern Works prototype recuperator has led to the development of a much improved fibre sealing system and also to improvements in the structural design.[192]

A batch of six ceramic recuperators have now been built on new 100 tonne gas-fired soaking pits at the Normanby Park Works.[193] These recuperators were built in hybrid configuration (Figure 6.3) with conventional convection air and gas recuperators, and incorporate the latest design features leading to a much improved design performance compared with the Llanwern Works prototype. The pits and ceramic recuperators, which were commissioned in late 1976 have been built by Priest Furnaces Limited. The hybrid recuperators have been designed to operate on an inlet waste gas temperature of 1200°C with a maximum heat input to the pits of 21 GJ/h. The air inlet temperature into the ceramic recuperator from the metallic recuperator is designed at 460°c for full flow and 520°C at turndown. The resultant full flow air preheat to the burners from the ceramic recuperator is designed for 660°C with plain tubes, and 780°C with cruciform inserts. The preheat temperature with and without cruciform inserts is limited to 800°C during turndown by the use of hot air bleeding. When combined with the 415°C gas temperature predicted, these high air preheat temperatures offer a fuel saving of 15% to 18% over the best available metallic only recuperator system. The predicted thermal performance of the hybrid recuperators, starting with a cold charge, is shown graphically in Figure 6.4. Figures 6.5 and 6.6 show the predicted and actual trial results for the hot charge hybrid recuperator. Extensive leakage tests have been carried out on all the new pits with very encouraging results. The total air leakage at full flow, hot condition, through the ceramic recuperators when new varied from 1.6% to 3.2%. No significant increase in air leakage above the new condition has been noted on any pit.

Studies have been made for incorporating ceramic recuperators in addition to metallic recuperators on existing soaking pits at a number of BSC works. Several of these studies have shown potential for installing ceramic recuperators with paybacks in the region of 1½ to 3 years. Two further ceramic recuperators, also in a hybrid configuration, have been built by Priest Furnaces Limited on new gas-fired pits at the Appleby-Frodingham Works. The acceptance trials results follow closely the computer prediction performance curves showing the system to be fully proven.

Thermal Efficiency Ltd., and Metallurgical Engineers Ltd., have been granted manufacturing licences in addition to Priest Furnaces Ltd., and these licencees are planning to exploit the recuperator overseas as well as in the UK.

6.5.2 BSC Ceramic Rotary Regenerator (C R R)

At the time of its inception it was hoped that the ceramic recuperator concept could be applied to reheating furnaces as well as soaking pits. Subsequent experience has shown however, that the huge number of tubes necessary to cool the much greater waste gas flows of reheating furnaces would create serious problems and lead to a massive and costly structure. As a result the British Steel Corporation is currently developing a ceramic rotary regenerator to provide a compact, highly efficient air preheater suitable for use on reheating and other high output fuel-fired furnaces.[194]

The basic concept of the rotary regenerator is simple. Heat is transferred by storage in, and recovery from, a porous ceramic matrix which is rotated across the flow paths of hot waste gases and cold combustion air. In practice, the energy storage matrix takes the form of a thick disc rotor and the gas streams are arranged to flow axially through the rotor in a counter-current mode. Rotor structures are subjected to a complex non-symmetric thermal loading which results in both high hoop and tangential stresses within the matrix.[195]

A predicted waste heat recovery applied to an oil fired soaking pit installation is shown in Figure 6.7.

The subject is discussed in detail in the following chapter.

<u>6.5.3 High Temperature Air Recuperator (HTR)</u>

Metallurgical Engineers have developed a high temperature metallic recuperator which they claim will achieve the same air preheats as thos currently being achieved with BSC Ceramic Recuperator.

The recuperator consists of 'U' assemblies manufactured from spun cast tubes in ET 50N essentially a 50/45 Cr/Ni steel. The chrome content ensures good resistance to corrosion by hot gases including sulphur, alkali and vanadium bearing gases, in both oxidising and reducing conditions. The high nickel content provides strength and this material has been used for stressed structural applications at temperatures of up to 1150^{o}C. The bends are formed from spun cast tubes and incorporate corebuster to promote heat transfer. The bends are thus welded to the two straight lengths to form the U assemblies. The tube sheet is made from 10mm thick Met 9 stainless steel and lined with a wet blanket ceramic fibre of grade 1450^{o}C. The headers, casing and transitions are made from carbon steel. The headers are bolted to extended tube sheet and lined with 75mm of ceramic fibre, the outer layer being a wet blanket.

Technical assessment of the unit was carried out in October 1980 by installing the free unit supplied in a soaking pit at the Scunthorpe steelworks.

The initial results during the trials show that designed preheats of 750^{o}C, similar to that being achieved by the ceramic recuperators, could be achieved offering a reduction in specific fuel consumption of 200 MJ/t on a pit throughput of 2400 t/week offering an energy saving of £50,000 p.a., at 1979/80 prices. However the unit failed after only two months operation due to material failure.

The system, if successful, would have offered a metallic recuperator of low cost, free of air leakage, and more acceptable to industry.

6.5.4 Howden - Metallic Regenerator

Howdens have recently developed a high temperature metallic regenerator which they claim will provide air preheats similar, or even better than, those currently being achieved by the ceramic recuperator with no air leakage.[196]

The design is based on the similar principle as that of the BSC - Rotary Regenerator except that the rotor is made of a metallic mesh. A general layout of the unit is shown in Figure. 6.8

Heat is transferred from the flue gas to the air via the heating surface which slowly rotates through each stream. The heating surface elements absorb heat from the flue gases and release it to air in a continuous process as the surface is rotated. Uniform heat transfer is thus obtained with the rotary regenerator, unlike the cycling temperature patterns associated with static brick regenerators.

The rotor comprises a hollow central shaft of fabricated construction allowing air cooling with radial partition plates arranged spokewise attached at the outer periphery to a continuous cylinderical shell. The central shaft is insulated from high temperature fluids and the radial partition plates are attached to it at the cold end only, thus allowing freedom for expansion of these in the axial direction. The rotor is contained within a cylindrical housing construction from steel panels supported off the preheater and pillars. The housing is provided with end plates to which the gas and the air ducts are connected. Between the gas and air openings at each end of the housing are sector shaped plates forming sealing surfaces for the radial and circumferential seals.

The rotor is supported through the main bearings at the top of the preheater on to a twin beam arrangement, these beams being built into the preheater end pillars. The end pillars are tied at the bottom by two box section girders. The whole support thus forming a rigid built-in structure.

The selaing between the gas and air sides is acheived as follows:

(i) Radial Sealing: Strips at each end of the rotor are attached to the division plates of the rotor and are capable of adjustment, thus forming a seal with the end closing plates of the housing situated between the gas and air ducts.

(ii) Circumferential Seals: Strips at each end of the rotor and at the outer and inner preiphery are attached to the rotor. These are capable of adjustment and forming a seal with the end closing plates of the housing.

(iii) Hub Seals: Strips at each end of the rotor, attached to the end closing plates of the housing and forming a seal with the periphery of the rotor shaft forms hub sealing.

In order to maintain the elements in a clean condition, a motor driven single nozzle sootblower is mounted in the gas outlet duct, blowing against the gas flow.

6.6 CASE STUDY: TECHNICAL ASSESSMENT OF A RECUPERATOR INSTALLATION ON No.9 SOAKING PIT AT RAVENSCRAIG STEELWORKS

A design study for Ravenscraig Works, to establish the most efficient waste heat recuperation system to replace the waste heat boiler on No. 9 pit providing air preheat of 760°C, was carried out and this is presented below.

6.6.1 Technical Data of the Soaking Pit (Oil Fired)

Air Flowrate	=	8840 Nm^3/h
Thermal Input	=	8.3 MW
Pit Area	=	169.55 m^3
Charge	=	6 ingots
Ingot Dimensions	=	2.286m x 1.422m x 0.762m
Hot Charge Track Time	=	2-5 h
Cold Charge Temperature	=	20°C
Stripping Temperature	=	1100°C
Charge Tonnage	=	117 tonne
Fuel Flow	=	750.7 kg/h
Escess Air	=	10%
Waste Gas Flow	=	9300 Nm^3/h

Waste Gas Analysis (Vol)	=	N_2 = 75.0%
		H_2O = 10.0%
		CO_2 = 2.96%
		SO_2 = 0.2%
		O_2 = 1.85%
Waste Gas Pressure	=	1 atm
Air Pressure (gauge)	=	1 bar
Pit Set Point	=	1330°C
Soak Temperature	=	1250°C

6.6.2 Ceramic/Metallic Recuperator

A mathematical model of soaking pit and recuperator performance was developed. The model assumes that there are no pit inspections, and that the fuel/air ratio remains constant during the cycle. This gives both the pit and waste gas temperature steady condition.

Typical pit operation without regeneration was modelled for both hot and cold charging for No.9 Soaking Pit and the results are presented in Figure 6.9 and 6.10. In order to assess the best possible recuperator design the following configurations were studied.

a) a full twin Ansteel-Escher type inverted metallic recuperator

b) a two pass ceramic recuperator with the convective part of the twin Ansteel-Escher type forming a hybrid recuperator

c) a two pass ceramic recuperator with the full twin Ansteel-Escher type recuperator.

Figures 6.11 to 6.13 show a comparative thermal performance of the metallic hybrid and ceramic recuperator when hot charging. It can be seen that were it possible for the metallic recuperator to sustain waste gases at temperatures of 1200°C, air preheats of the order of 660°C could be achieved. However, use of ceramic tubes improves the air preheats to 760°C due to an improved heat sink property of the material. The ceramic recuperator offers maximum air preheat in the configurations considered.

A similar assessment for the cold charging has been carried out and the performance curves are presented in Figures 6.14 to 6.16.

From the results obtained the study concluded:

1. Were it possible to push high temperature hot gases directly to the metallic recuperator, very near designed air preheats could have been achieved at low costs. Unfortunately, the system requires air dilution to drop the maximum gas inlet temperature to the recuperator to 1000°C thus offering an air preheat of 500°C only.

2. Since the soaking pit in question has no existing recuperator, for economic reasons ceramic recuperator installation is recommended in preference to the hybrid configuration although the latter offers a better energy savings.

Design Specifications of the unit are given below:

Ceramic Recuperator	-	8 Tubes
		4 Rows
		2 Passes
Standard Tube Dimensions	-	O/D = 0.1397 m
		I/D = 0.1016 m
		Span = 1.524 m
		Length = 1.829 m
Thermal Conductivity		= 6.7 W/mK
Pitch/Dia.		= 24/11
Waste Gas/Preheat Temp.		= 755°C

Capital Cost (Updated to 1980 prices) = £250,000

6.6.3 Ceramic Rotary Regenerator

In designing the regenerator the following assumptions were made:

(i) In order to keep the support grid stress to an acceptable level, the depth of the matrix should not exceed 800 mm

(ii) For ease of manufacture, the hydraulic dia. of pores should be a minimum of 10mm and the wall thickness 3mm.

(iii) To avoid acid dew point, matrix cold face temperatures to be 200-400°C.

(iv) A leakage factor of 10% was assumed. Incorporation of the rotary generator was achieved by assuming the presence of the metallic recuperator whose quoted performance was actually that of the rotary regenerator. It was recognised that there were extended assumptions in this technique.

(a) there may not be comparable variation of heats delivered with waste gas temperatures for the two heat exchangers. This variation was later established to be linear.

(b) the relationship between full flow and turndown performances may differ. This was established to be so. It was therefore accepted that all preheats and fuel savings calculated will be marginally less than those obtained in operation.

Performance curves for soaking pit operation with regeneration are shown in Figures 6.17 and 6.18.

It can be seen that using a 4m rotor, air preheats of up to 1000^{o}C can be achieved, giving an energy saving of 647 MJ/t (54% of the total) for hot charge and 1451 MJ/t (50% of the total for the cold charge) at a capital installation cost of about £500,000 (Rotary Regenerator cost £300,000).

6.6.4 HTR Recuperator

Metallurgical Engineers' HTR design specifications for BSC Ravenscraig are presented below:

Tube Size	106mm ID and 120 mm O/D
Tube Material	E T 50 N (50% Cr)

Tube Ferrules	127mm nominal bore 6mm thick
Core buster	Type 2, Met 9, 2mm thick
Number 'U' Assemblies	15
Effective length of Recuperation	4770 mm
Straight tube length	2000 mm
Total Outside Surface Area	27 m^2

The engineering design shows that a 4 pass recuperator will offer an air preheat of 750°C for the unit cost of £120,000.

However, since the collapse of their first unit after only two months operation the reliability of the unit at this stage is highly questionable.

6.6.5 Howden's Metallic Regenerator

Figure 6. 19 shows that the application of the Howden's metallic regenerator can offer air preheats of 760°C at a capital cost (turn-key) of £200,000. Should the predicted data be correct, the unit can provide high energy recovery at low cost. However, it should be recognised that no such unit is currently or otherwise in operation. Thus the unit under soaking pit operating conditions may find itself clogging, in spite of soot-blower, thus giving a very high maintenance cost and/or unreliability

6.6.6 Comparative Assessment of Recuperative Systems

Selection of any type of recuperator is very much based on[197]

- initial cost of the installation
- operating temperature range
- its ability to meet the designed air preheat and turndown
- operating pressure and pressure drop through the recuperative system
- campaign life and its reliability and thus min of down time
- ease of maintenance and site considerations
- resistance to corrosion/fracture

In comparative analysis of the systems discussed it can be concluded that regenerators would ideally meet the air preheats required with a lower capital costs but are still in the development stages. Therefore of the newly developed proven technologies BSC developed ceramic recuperators though more expensive would be the ideal unit.

6.7 CONCLUSIONS

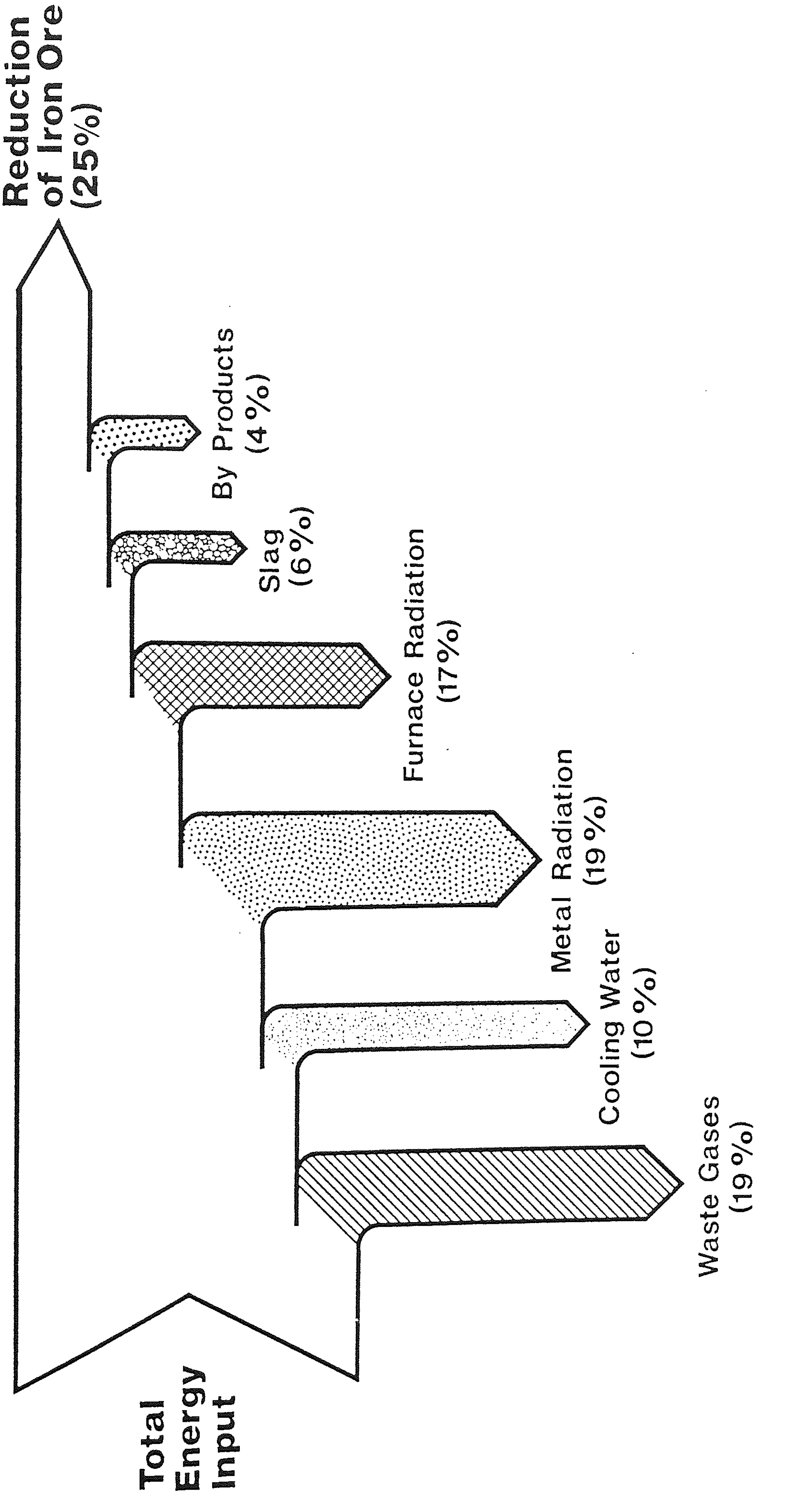
From the study it is concluded that as much as 19 per cent of the total energy input to a steelworks is dissipated to atmosphere through waste gases offering a potential energy recovery source.

Because of a number of problems e.g. corrosion, fouling etc. associated with the low grade waste heat, it is difficult to recover this energy at an economical cost without substantial R & D effort. On the other hand, high grade waste heat available from soaking pits and reheating furnace waste gases offer substantial energy recovery opportunities.

The conventional method of recovering heat through waste heat boilers is only successful where there is a works need for the steam generated. In general it can be said that air preheat offers the best mode of recuperation at a minimum of cost.

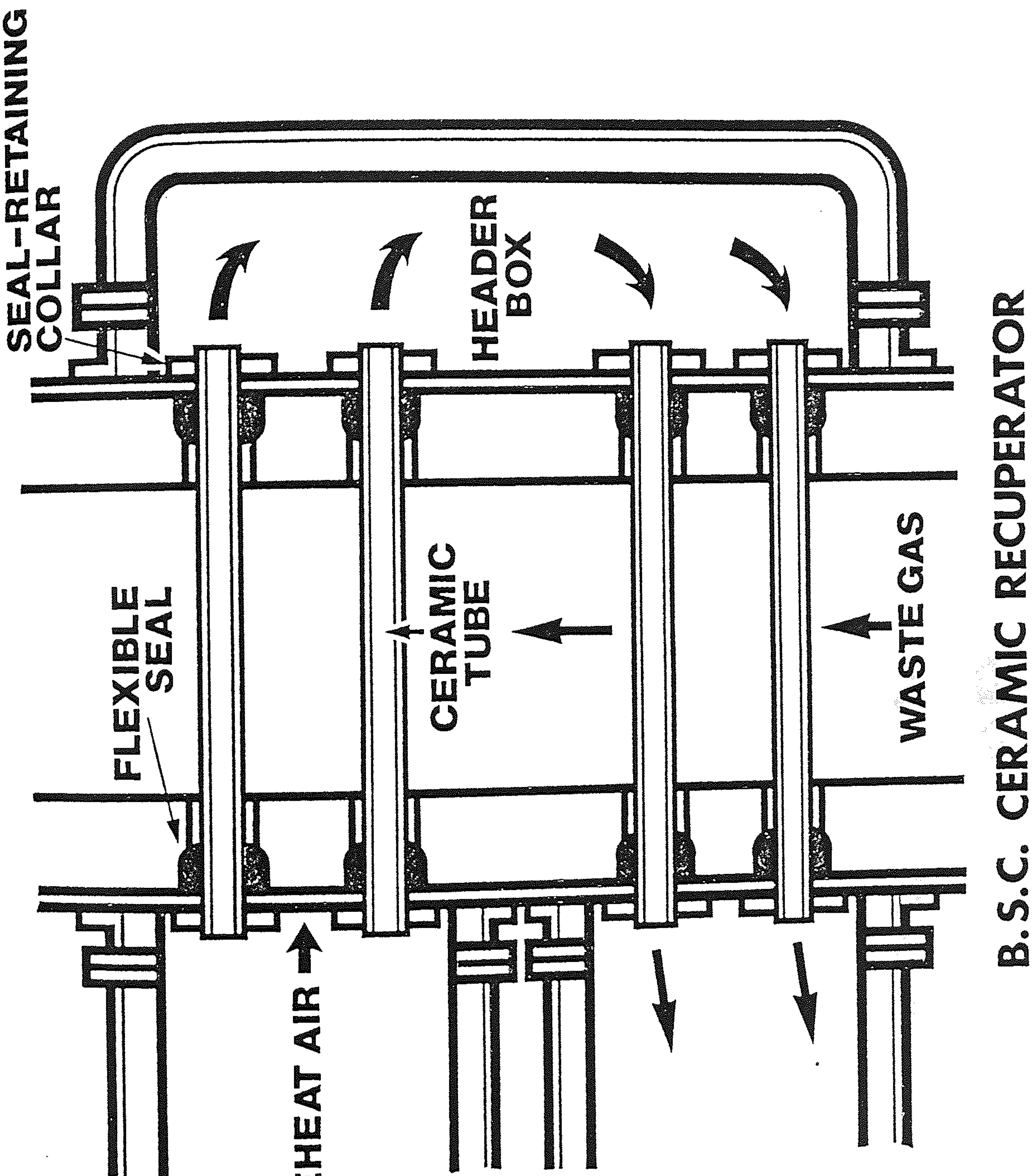
Metallic recuperators have been available in the field for a long time. But because of their limitations to cope with high temperature waste gases the newly developed BSC ceramic recuperators offer substantially higher air preheats and thus increased savings through this recuperation system. The system is now fully proven. However, due to the industry's conservative attitude towards ceramics and their preference to metals, the ceramic recuperators have not been fully exploited. Case studies carried out for a new recuperator installation show that BSC ceramic recuperator would be the best to install.

Further developments to make the recuperative units more compact are in hand and success of these units will offer more efficient and economical recuperative systems.



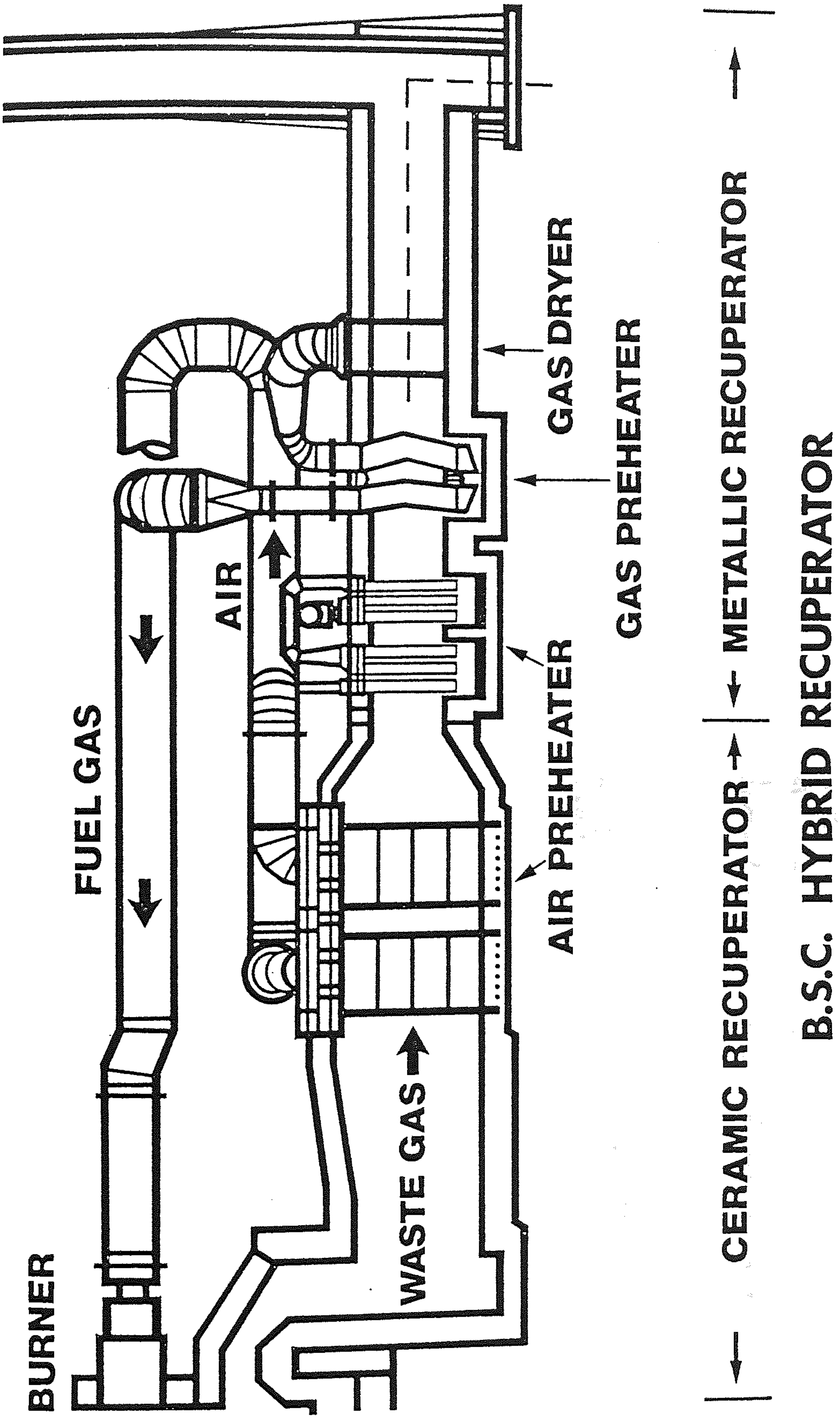
Energy Distribution in Steel Making

FIGURE 6.1



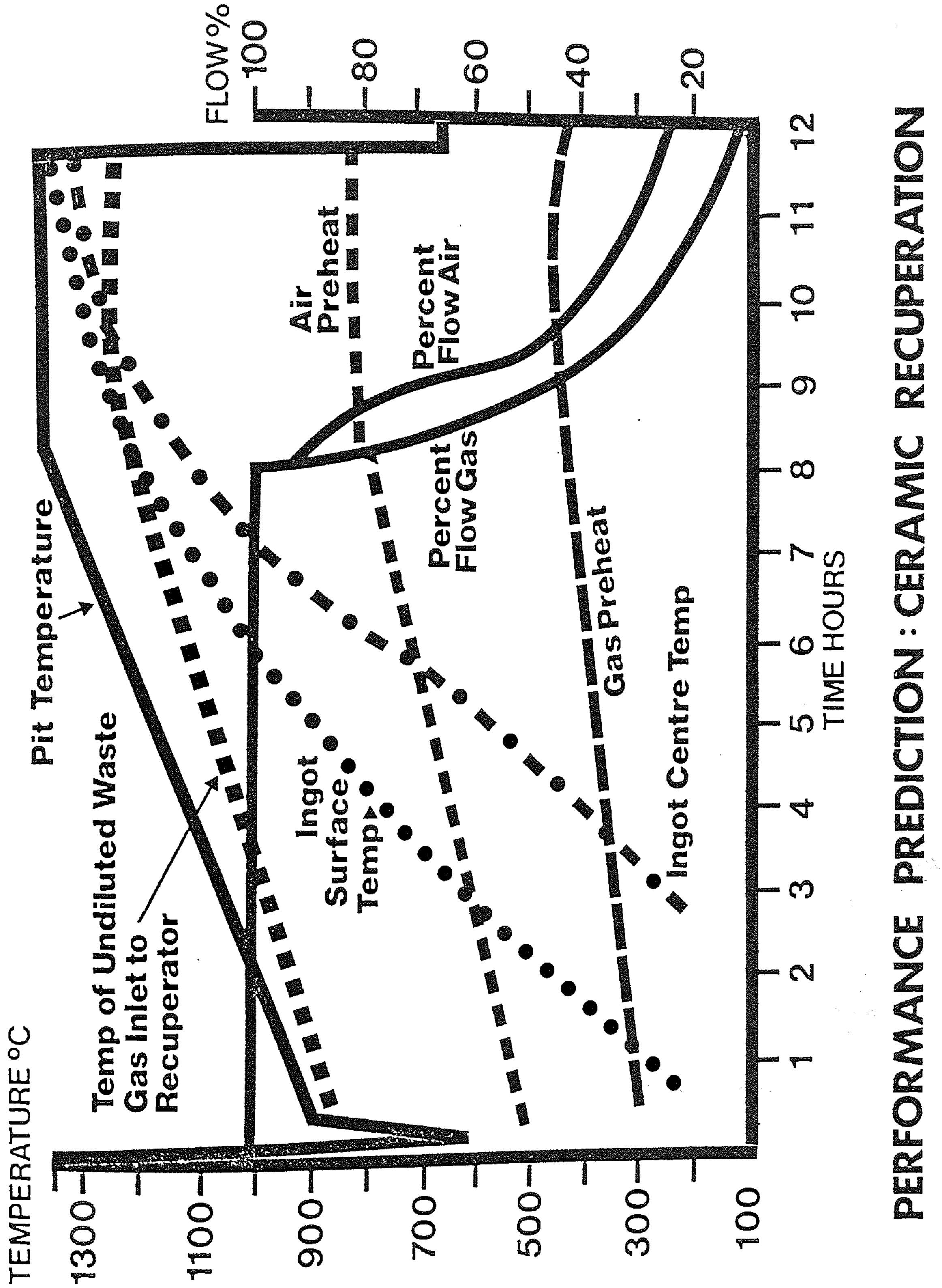
BSC CERAMIC RECUPERATOR

FIGURE 6.2



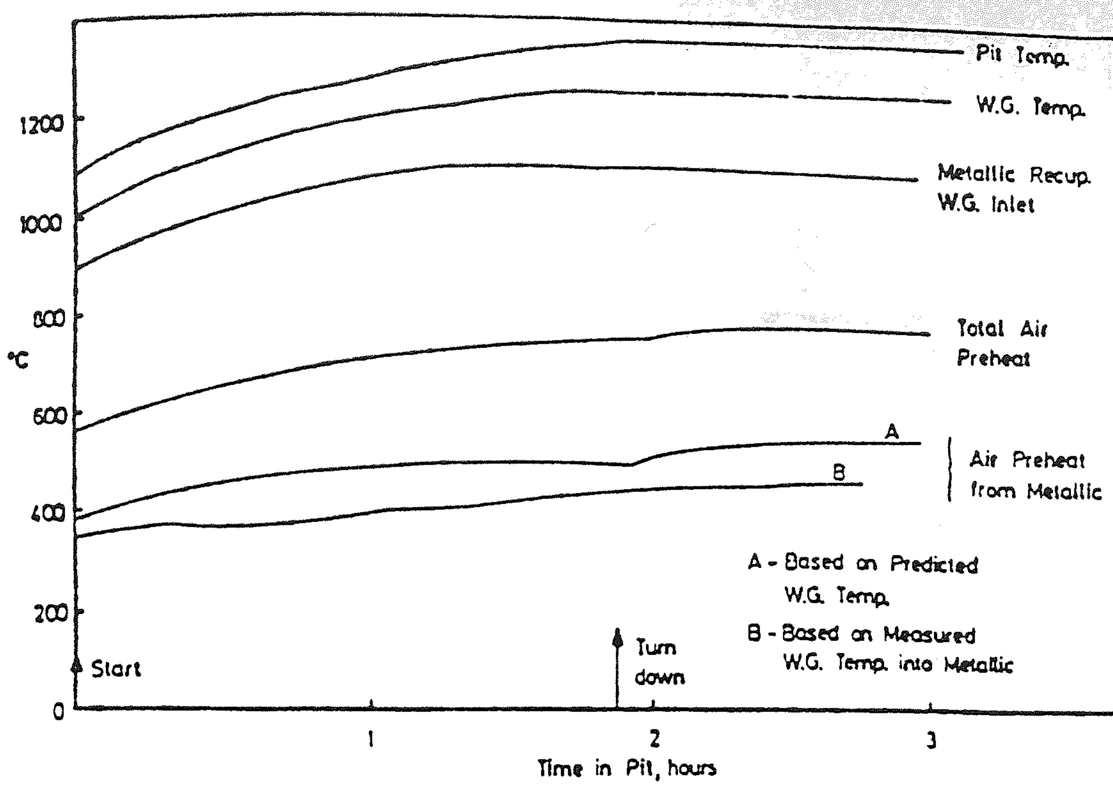
BSC HYBRID RECUPERATOR

FIGURE 6.3



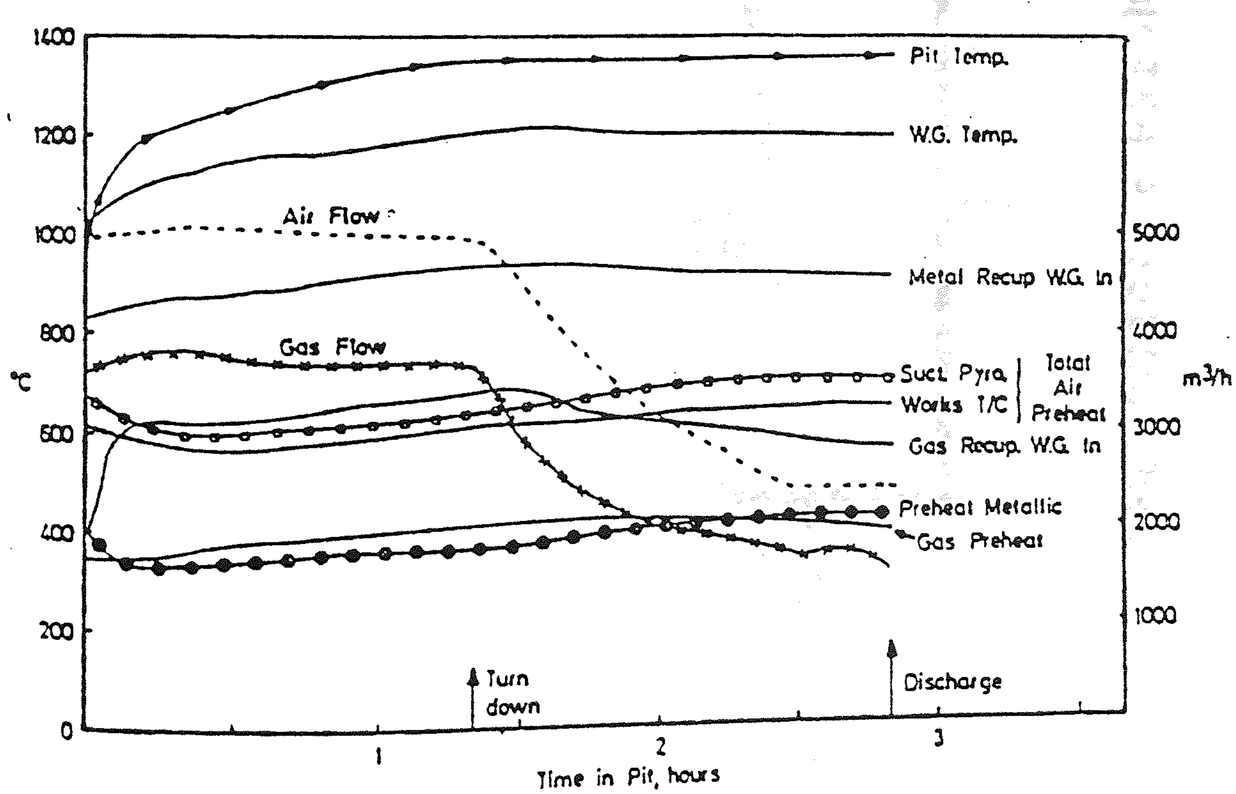
PERFORMANCE PREDICTION: CERAMIC RECUPERATOR

FIGURE 6.4



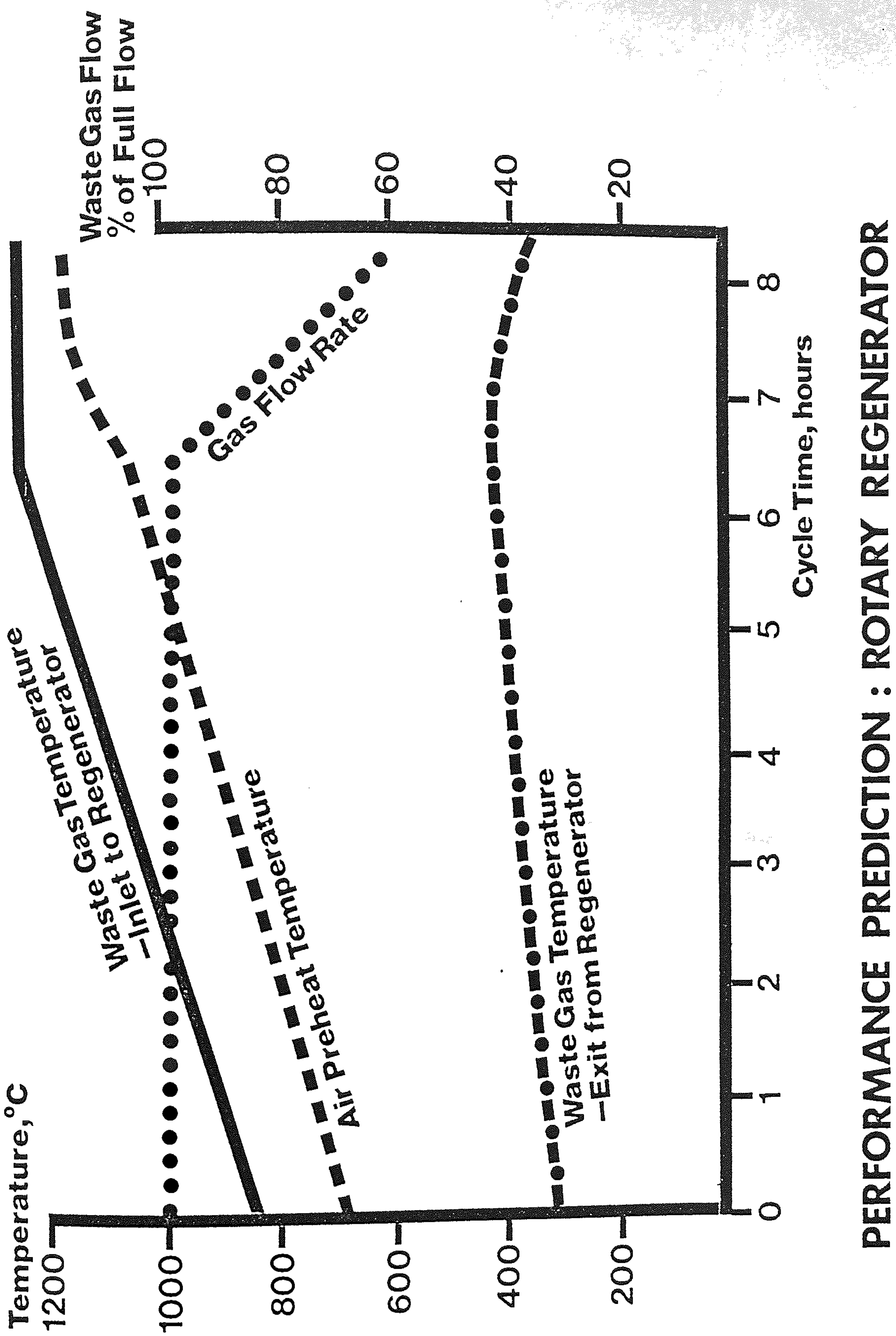
COMPUTER MODEL TEMPERATURE PREDICTIONS - HOT CHARGE - 16 INGOTS

FIGURE 6.5



NORMANBY PARK - HOT CHARGE RESULTS-16 INGOTS.

FIGURE 6.6

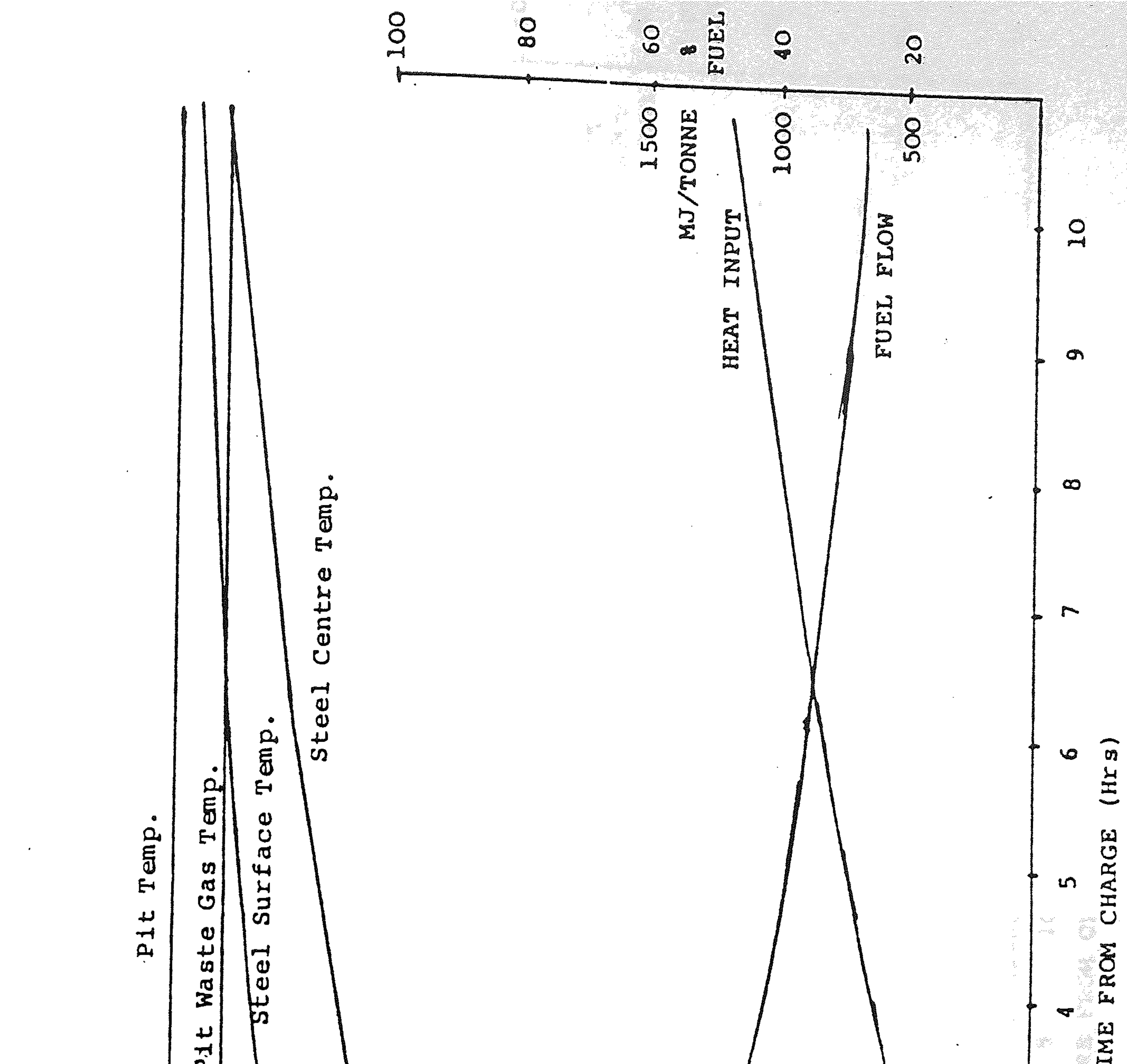


PERFORMANCE PREDICTION: ROTARY REGENERATOR

FIGURE 6.7

HOWDEN'S METALLIC ROTARY REGENERATOR

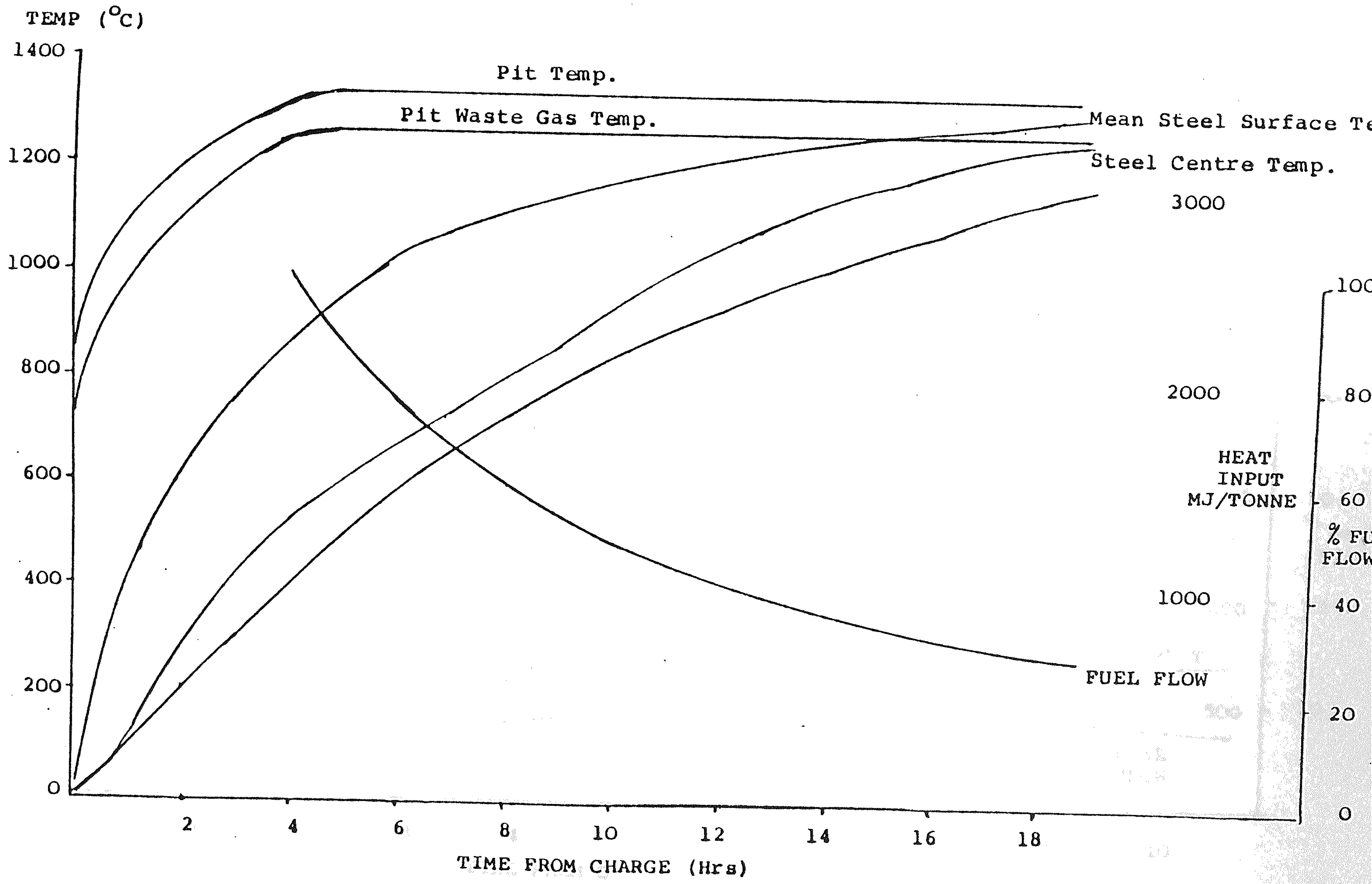
FIGURE 6.8



HOT CHARGE (2 HRS TRACK TIME) NO RECUPERATION OIL FIRING AT BOTTOM

SOAKING PIT WITH NO RECUPERATION - HOT CHARGE

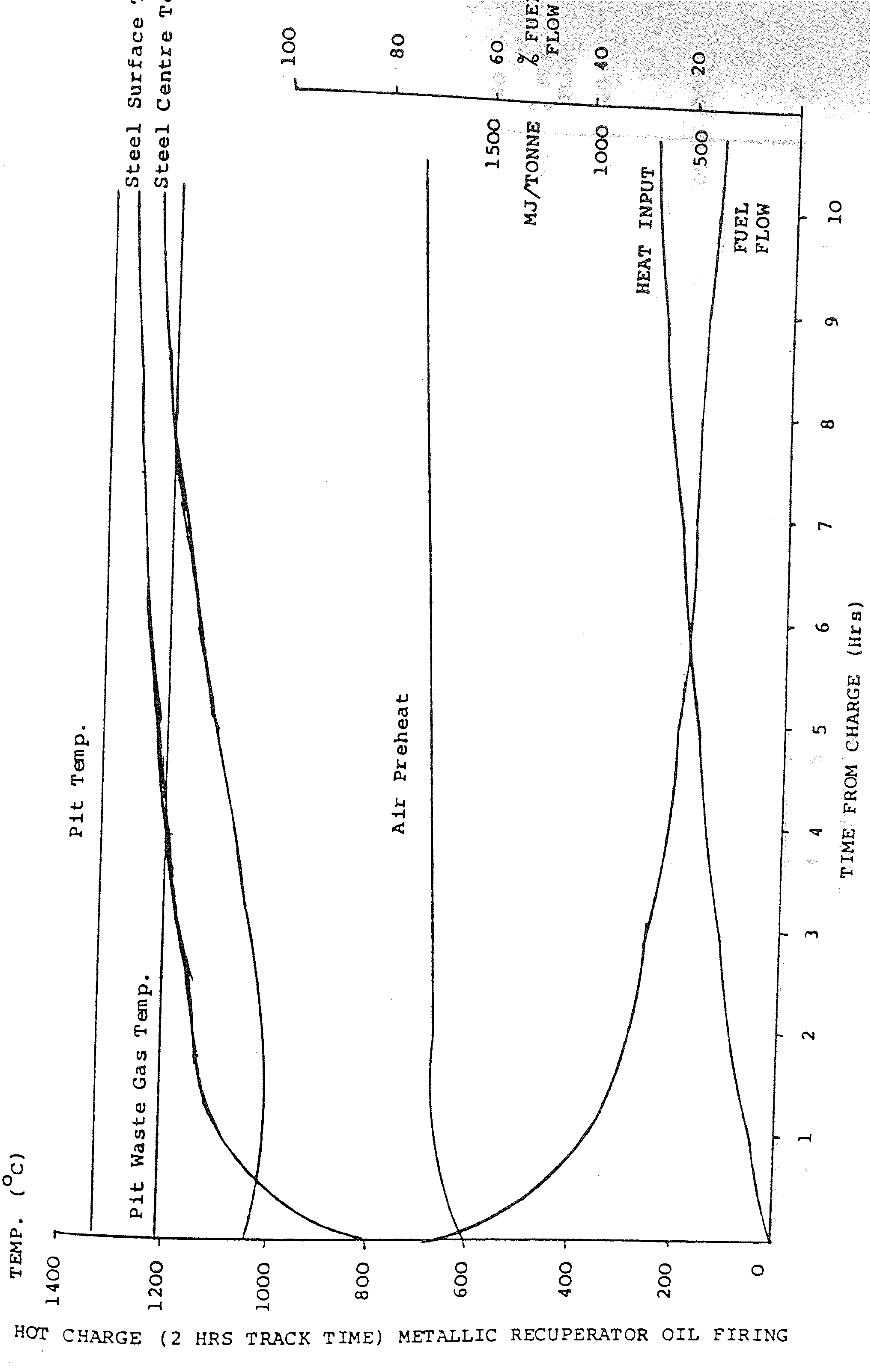
FIGURE 6.9



COLD CHARGE 20C - NO RECUPERATION OIL FIRING

SOAKING PIT WITH NO RECUPERATION- COLD CHARGE

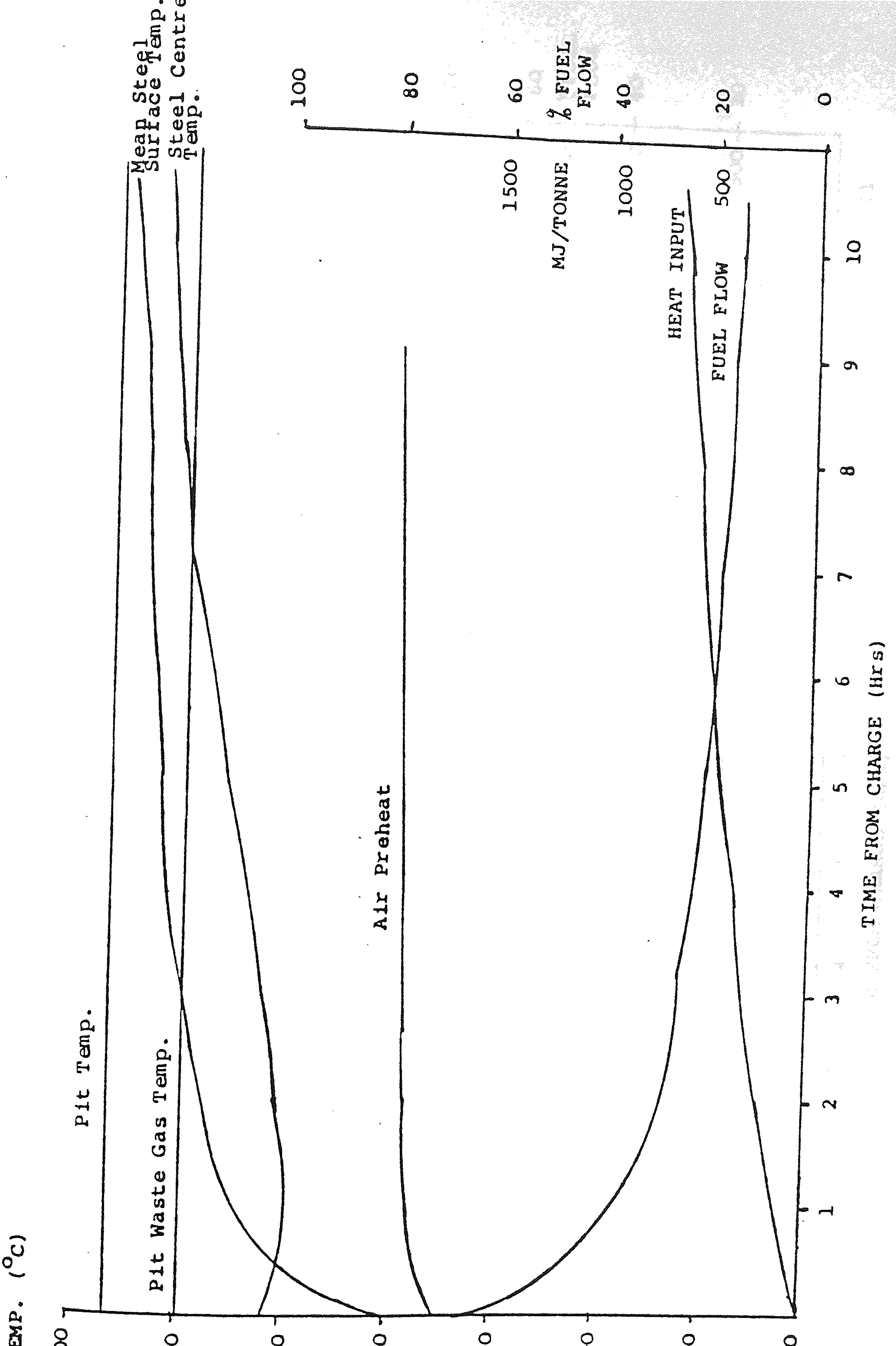
FIGURE 6.1



HOT CHARGE (2 HRS TRACK TIME) METALLIC RECUPERATOR OIL FIRING

SOAKING PIT WITH METALLIC RECUPERATOR - HOT CHARGE

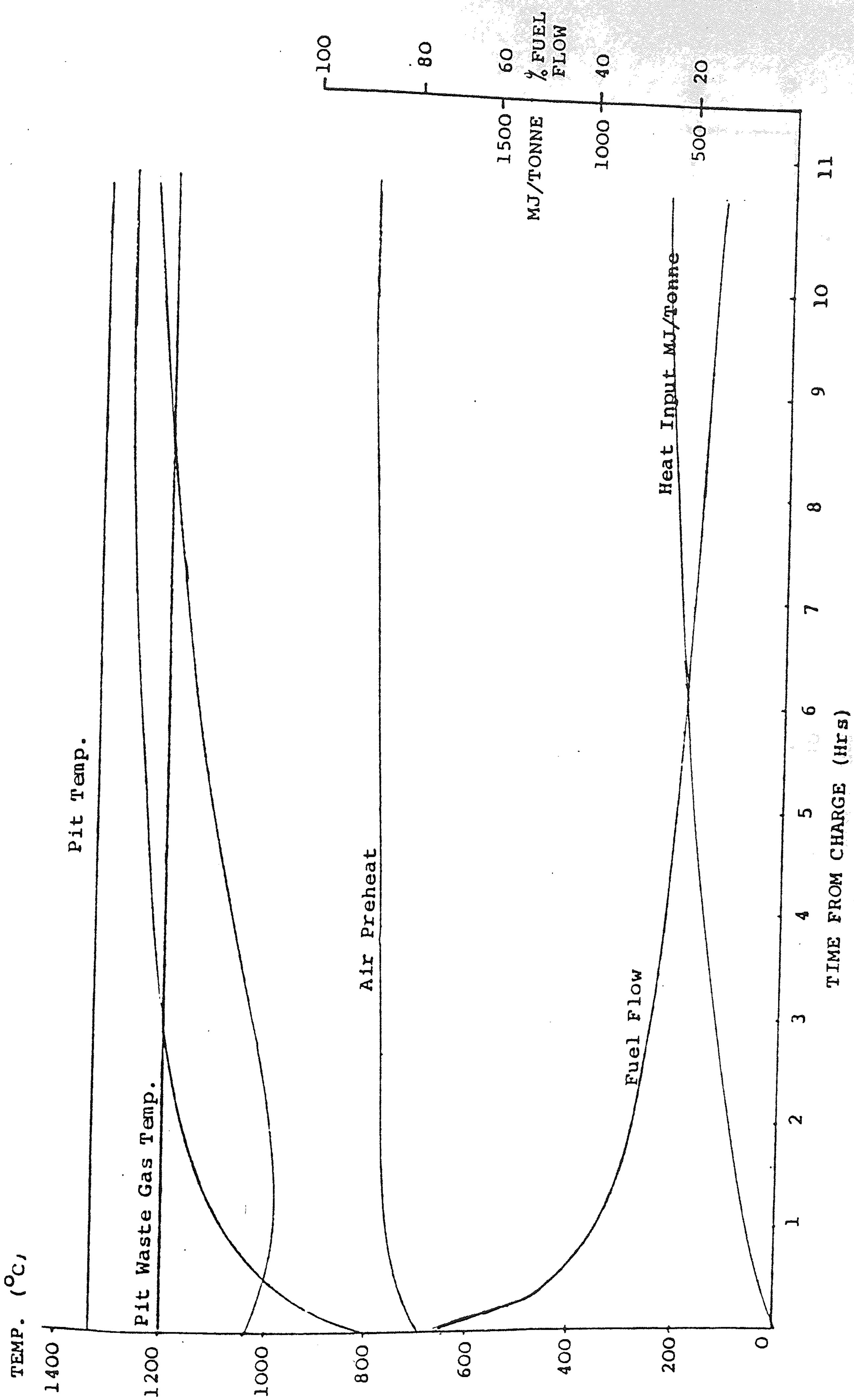
FIGURE 6.11



HOT CHARGE (2 HRS TRACK TIME) HYBRID RECUPERATOR OIL FIRING

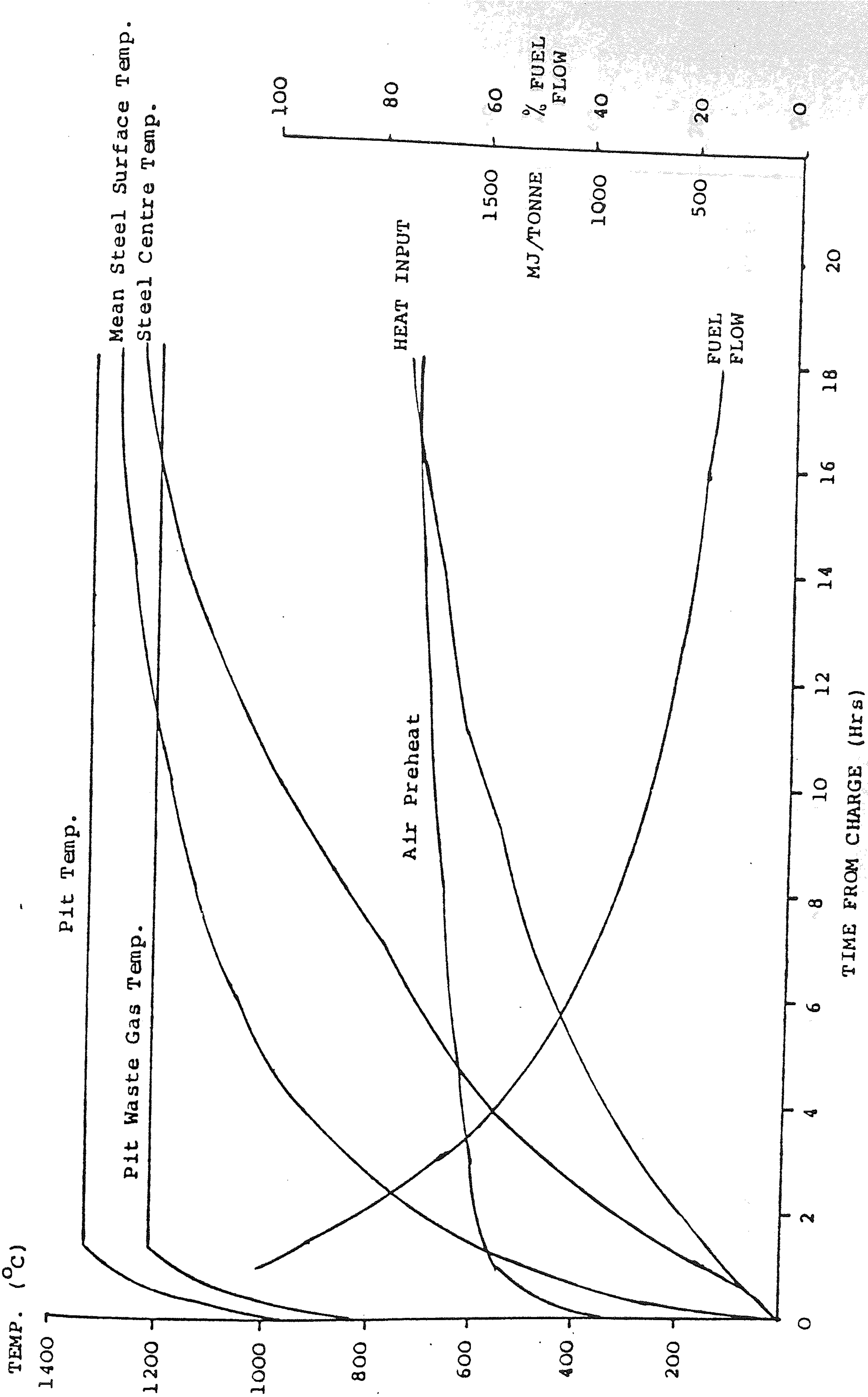
SOAKING PIT WITH HYBRID RECUPERATOR - HOT CHARGE

FIGURE 6.12



SOAKING PIT WITH CERAMIC RECUPERATOR - HOT CHARGE

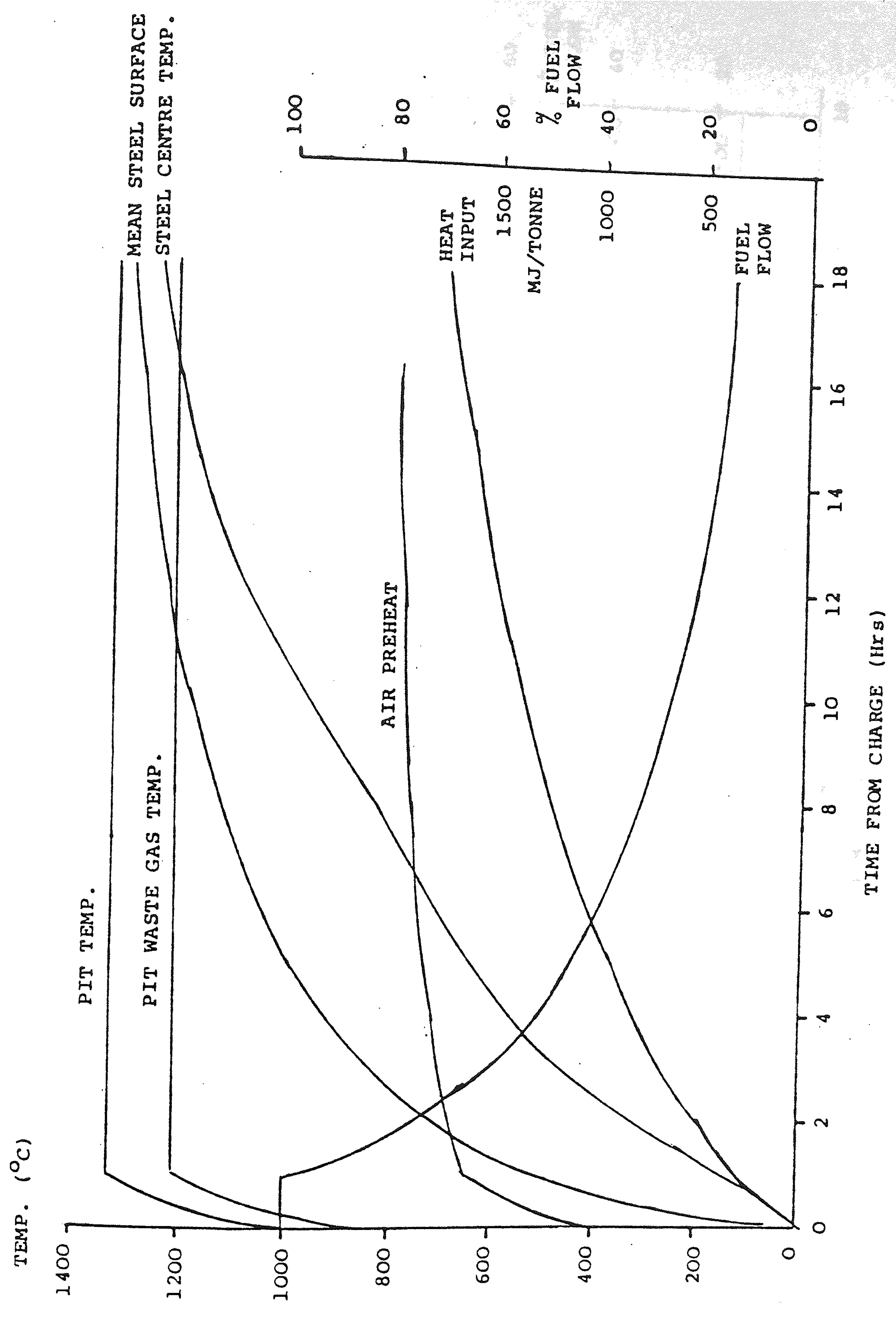
FIGURE 6.13



COLD CHARGE 20C - METALLIC RECUPERATOR OIL FIRING

SOAKING PIT WITH METALLIC RECUPERATOR - COLD CHARGE

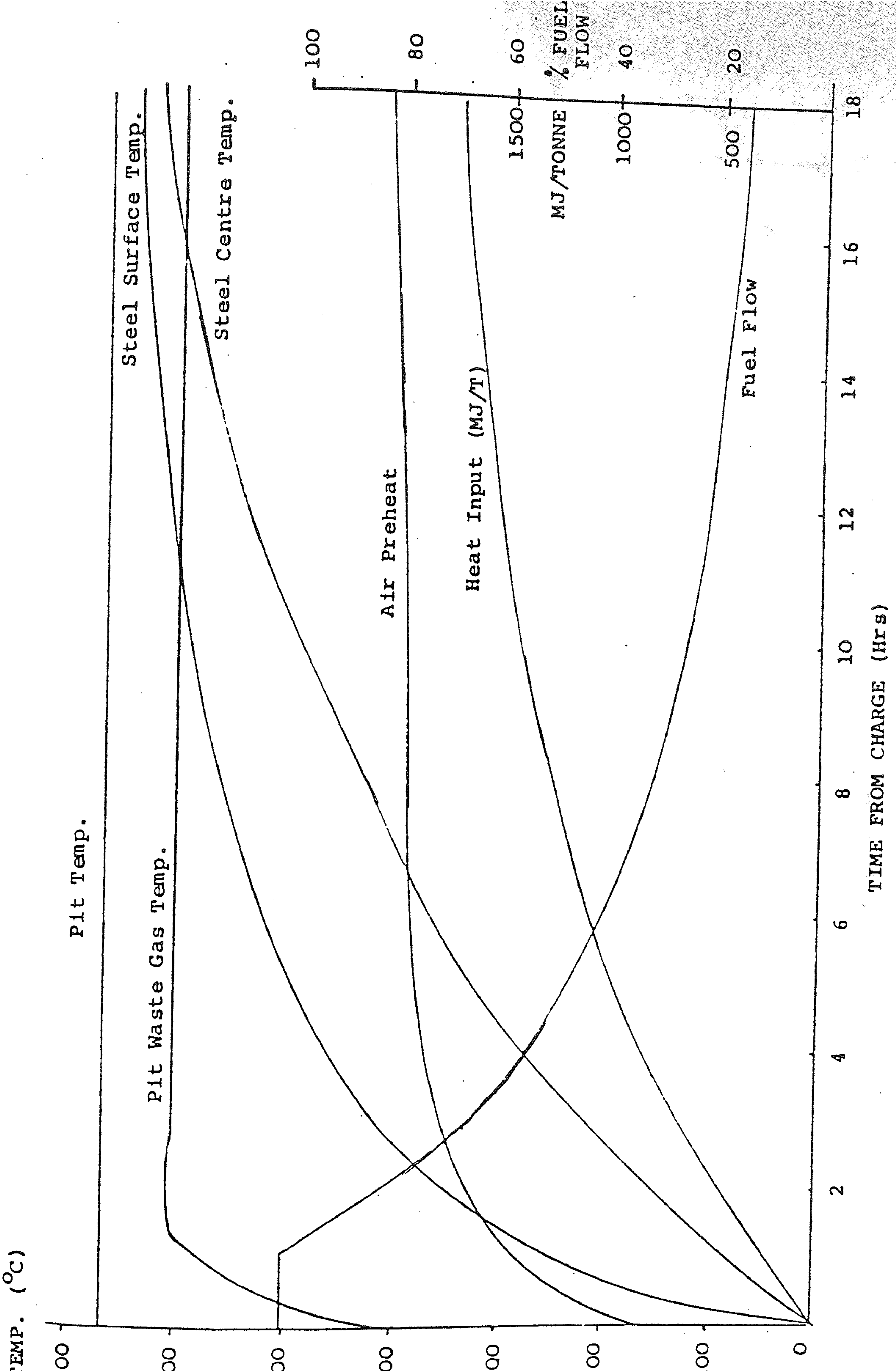
FIGURE 6.1



COLD CHARGE 20C - HYBRID RECUPERATOR

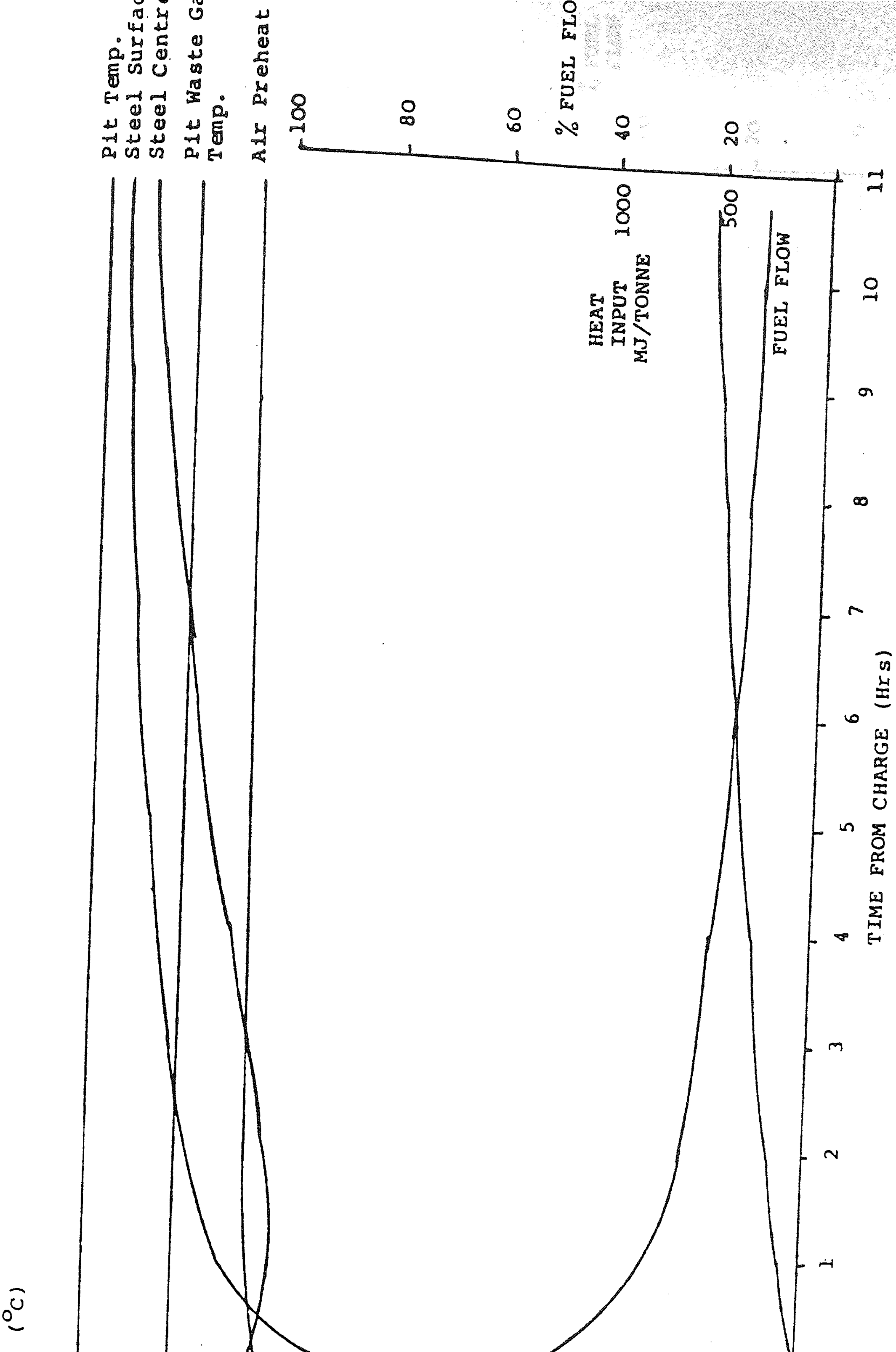
SOAKING PIT WITH HYBRID RECUPERATOR - COLD CHARGE

FIGURE 6.15



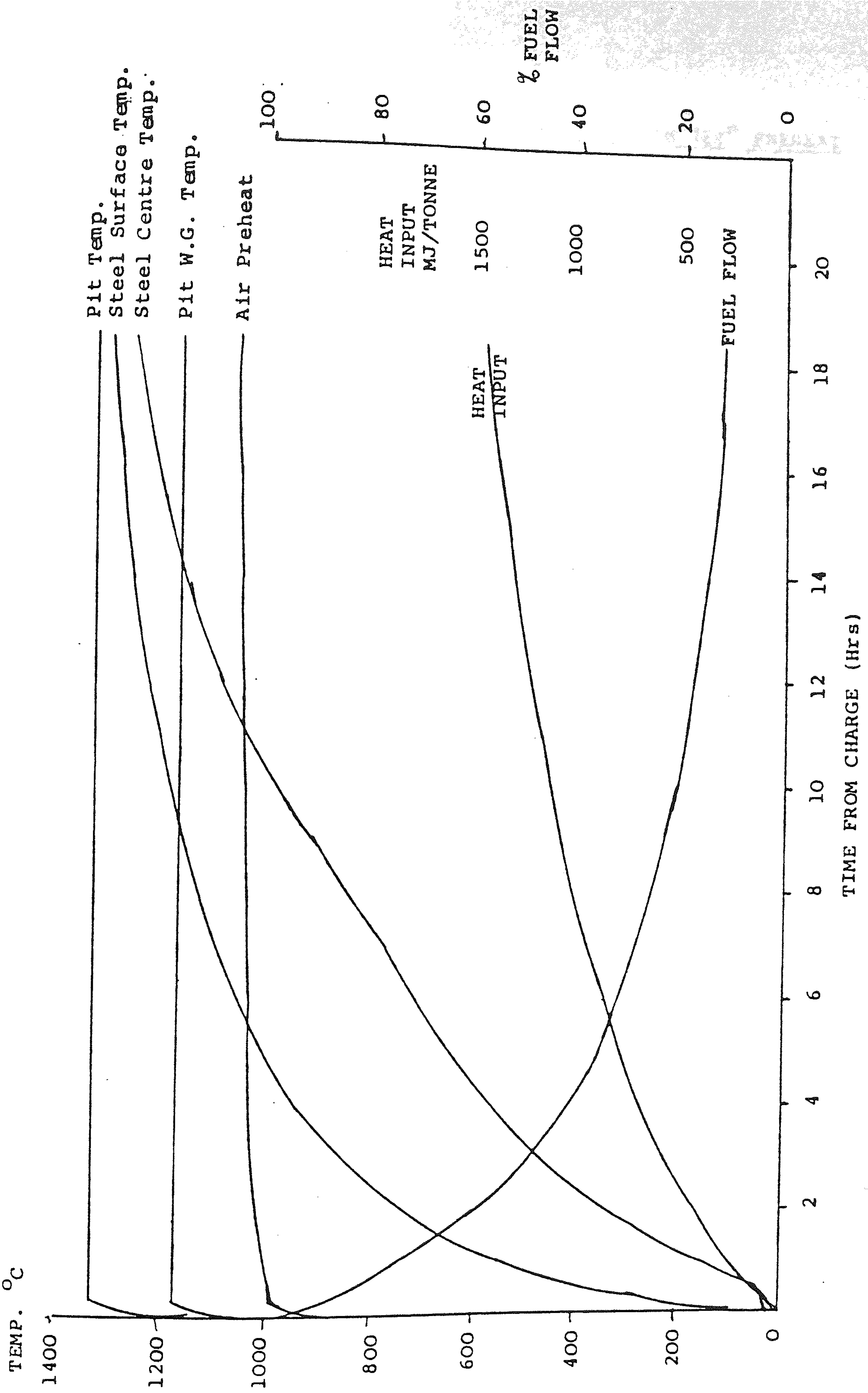
COLD CHARGE CERAMIC RECUPERATOR OIL FIRING

SOAKING PIT WITH CERAMIC RECUPERATOR - COLD CHARGE



HOT CHARGE (2 HRS TRACK TIME) ROTARY REGENERATOR OPERATING
OIL FIRING

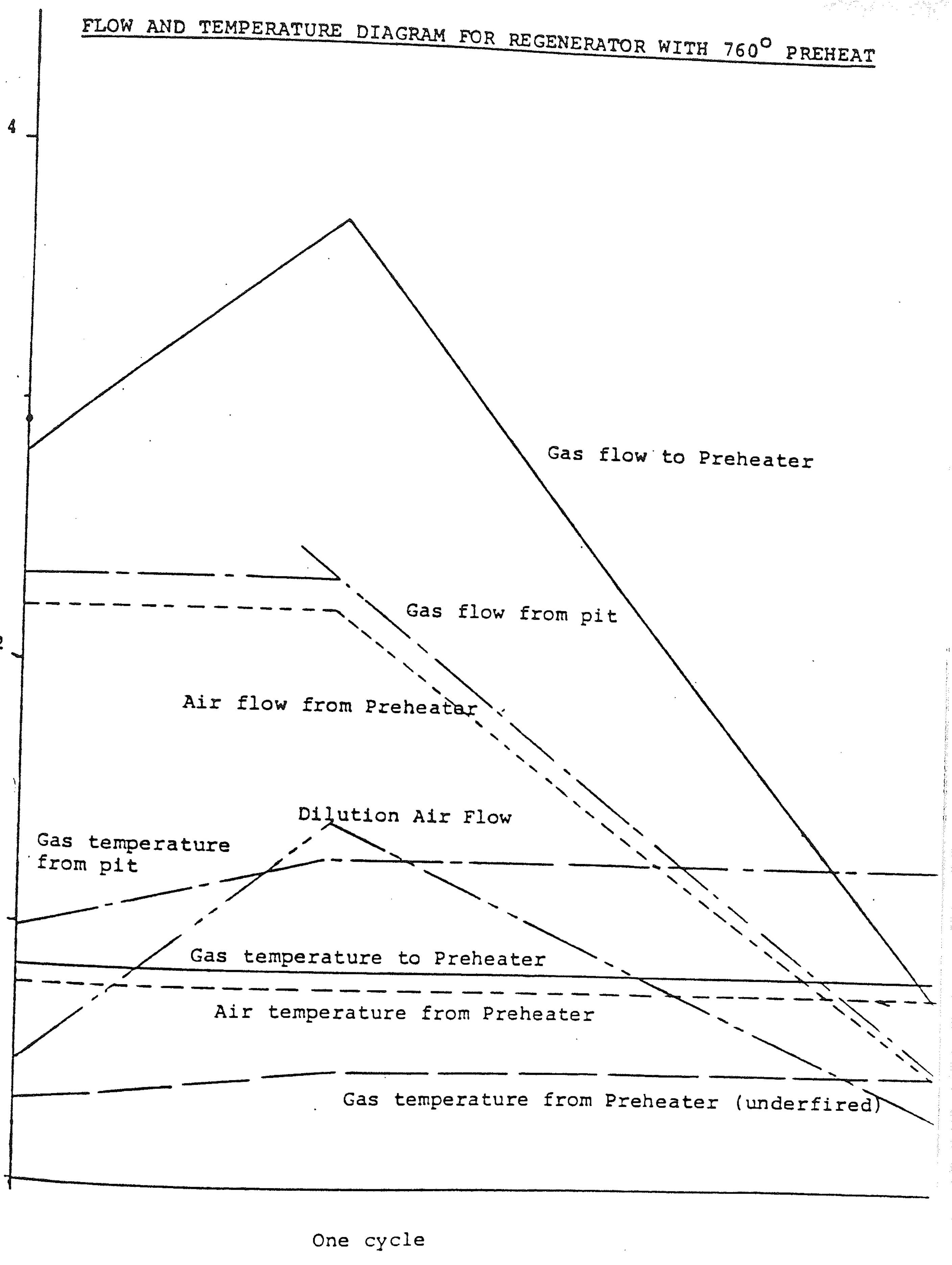
SOAKING PIT WITH ROTARY RECUPERATOR - HOT CHARGE



COLD CHARGE - ROTARY REGENERATOR OPERATING OIL FIRING

SOAKING PIT WITH ROTARY REGENERATOR - COLD CHARGE

FIGURE 6.18



THERMAL PERFORMANCE OF HOWDEN'S METALLIC REGENERATOR

FIGURE 6.19

CHAPTER 7

CERAMIC ROTARY REGENERATOR

7.1 INTRODUCTION

7.1.1 Introduction

Steel is produced for industrial use in many forms - from strip for the car industry to girders for the building industry - and most require hot rolling as part of their finishing process. To heat the steel for the rolling mill, a wide range of reheating furnaces are in use throughout the world, heating blooms, slabs and billets of many sizes. Of these the two most common type are pusher and walking beam furnaces and range from single-zone furnaces heating a few tonne of steel in an hour to large multiple-zone furnaces heating hundreds of tonne steel every hour. Fig.7.1.

Typical sizes of these furnaces are:

Hearth length	- 33m max.(pusher type); up to 50m (walking beam)
Height	- 6 m
Width	- 20m

An alternative way of heating is through a soaking pit where the bloom is left to soak in a box type furnace until its temperature is reached for rolling.

After ironmaking plants, rolling mills are the main energy consumers at iron and steelworks. Reheating furnaces heat input could range 2.0 - 3.0 GJ per tonne of steel pushed for slabs.

The furnaces are lined with conventional refractory materials with typical drop out temperatures for slabs at around 1230^{o}C for conventional carbon steels and a little higher for low/high alloy steels.

Tables 7.1 and 7.2 [198] indicate that sensible heat losses in waste gases from soaking pits and reheating furnaces represent a major source of energy loss. Waste gas temperatures at the stack can vary from 200^{o}C to 800^{o}C, depending on the degree of recuperation.

Another area of energy rejection in the reheating furnace is the skid cooling (Table.7.3)

Only through extensive use of continuous casting can this process be replaced to some extent. However, at the present time where only 30% of the total BSC's steel is produced through the continuous casting route and with the present market and investment situation pertaining, that day is a long way to come. Furthermore because of the nature of demand for steel products, both soaking pits and reheating furnaces are here to stay.

In the present times of increased cost of fuel and diminishing world energy resources, an efficient operation of reheating furnace can considerably contribute to a reduction in energy requirements and fuel bills.

In order to achieve a high furnace efficiency and thus a low specific energy consumption, it is necessary to aim simultaneously on the following:

NOTE: In the following text use of the work reheating furnace equally applies to soaking pits.

1. Low Heat Loss: In general losses, by conduction through the walls and radiation losses through various openings, particularly the charging and extraction doors represent 5% of total energy input.[199-200] These can be reduced by improved roof insulation and by minimising opening of doors during charging. Skid cooling losses which can amount to 20%[198] can be minimised by improved insulation and waste heat recovery in the form of steam through the evaporative cooling of skids where the water enters the furnace cooling system at approximately the saturation temperature and during its passage through the cooling system some is converted to steam, which is separated off in a steam drum and re-used on the furnace for burner atomisation.

2. A high adiabatic combustion temperature: The adiabatic temperature is the temperature which would be reached by products resulting from complete combustion of the fuels which would occur without any heat exchange with the outside environment. This is very much dependent on the type of fuel used, air factor (λ) and the combustion air temperature. Therefore an increase in combustion air temperature through recuperation offers improved furnace performance.[201]

3. Waste gas temperature as low as possible: It goes without saying that any heat loss to atmosphere is a waste and should be minimised.

Therefore an energy conservation programme on a reheating furnace can be achieved through:

1. Reduced Energy Input to the furnace.
2. Maximise waste energy recovery.

7.1.2 Reduction in Energy Input

The BSC is currently actively engaged in installing micro-process control systems on the reheating furnaces[202], thus eliminating the wasteful energy during delays in the rolling mills. Through the computer control, each zone is kept to its correct temperature and during any delay periods when the slabs/blooms are held back for longer periods in the furnace, the burners from individual zones are switched off, thus saving heat. In addition every effort is made to operate the burner correctly with a regular burner maintenance programme. The furnace is kept gas tight with good fitting doors. With poor fitting doors, the furnace under positive pressure can lose as much as 8% of the total heat input through flames and heated gas leakage. High furnace pressure can cause damage to doors, frames and structures and low furnace pressure can pull in cold air, chill the hearth and thus give rise to cold ends. For an ideal operation the pressure in the furnace should be balanced at sill level of the stock discharge door.

Fuel is used to heat the furnace and maintain the stock at the required temperature. If the hearth is not loaded to capacity this results in excess use of fuel due to maintaining furnace capacity at temperature and then not utilising the capacity.[203] For any energy conservation it is essential that the furnace loading should be maintained at as high a level as possible with a maximum number of shifts in operation. It is also important that when loading, stock sizes are not mixed.

In the present times when the furnace utilisation factor is low due to the depressed demands, energy conservstion can be achieved through

(a) maximum loading of the furnace with reduced number of shifts;

(b) partial loading of the furnace with some or all of the burners off in the preheat zone, together with increased furnace pressure.

Therefore it can be concluded that the manner in which the furnace is 'controlled', particularly with respect to selection of:

- fuel distribution in the various heat zones;
- the levels of 'indexing temperatures' and other furnace values for the heat flow distribution along the furnace;
- timing (output per hour);
- furnace loading factor (slab spacing and length);
- extraction temperature

all have a direct effect on waste gas temperature and furnace energy consumption.

7.1.3 Energy Recuperation

In typical current practice the waste gas leaves these furnaces at temperatures in the region of 900°C to 1200°C and are passed through radiative or convective metallic recuperators to preheat incoming combustion air to temperatures around 400^{o}C, although 500^{o}C is achieved at few installations.

Whilst making significant savings in furnace fuel requirements, these metallic recuperators require dilution of the incoming waste gases with cold air in order to protect the metal surfaces from overheating and, in addition, they still exhaust the waste gases at comparatively high temperatures. Heat losses from an integrated steel-works of 2 million tonne/annum capacity exceeds 100 GJ/t from the soaking pits and 170 GJ/t from the reheating furnaces.

In order to improve heat recovery from these major sources, the Research Laboratories of British Steel Corporation developed a novel ceramic recuperator to provide higher air preheat temperatures to produce direct fuel savings (Chapter 6). At the time of its inception it was hoped that the ceramic recuperator concept could be applied to reheating furnaces as well as soaking pits. Subsequent experience has shown, however, that the huge number of tubes necessary to cool the much greater waste gas flows of reheating furnaces would create serious engineering problems and lead to a massive and costly structure. As a result, the British Steel Corporation embarked on the development of a ceramic rotary regenerator (CRR) to provide a compact, highly efficient air preheater suitable for use on reheating and other high output fuel fired furnaces.

This chapter describes the design and development of the BSC Ceramic Rotary Regenerator together with an economic assessment and potential savings that can be achieved.

7.2 LITERATURE SURVEY

7.2.1 Introduction

Waste heat recovery has always been an important part of industrial and engineering practice - never more so than now, with todays very high fuel prices. Throughout industry, in fields such as steel-making, aluminium processing, glassmaking, ceramics and many others using high temperatures, heat recovery equipment to fulfil very demanding requirements is needed. Recuperation through air preheat and in some cases preheating of fuels, reduces fuel consumption by returning waste energy back to the process thus achieivng a direct energy savings and improved process efficiency (Fig. 7.2)

The greatest energy savings are available in the recovery of high grade waste heat, and it is in this area that suitable equipment is lacking. The three types of high temperature (above 500^{o}C) heat recovery devices currently available are recuperators, switched regenerators and rotary regenerators.[204] Switched regenerators currently used in the blast furnace stove operation have enormous drawbacks in terms of size, poor efficiency and complex operation. To operate efficiently, there must be at least three stoves operating in conjunction and coupled with the need for all of the ancilliary equipment, switched regenerators have dropped from general usage.

Metallic recuperators are the most widely used of industrial air preheat devices today.[205-207] Due to material temperature limitations, it is usually required to dilute gases at temperatures over 1000^{o}C with air or water to lower them to acceptable temperatures. This material shortcoming prompted the development of ceramic recuperators. The newer, ceramic recuperators still have disadvantages in terms of low surface area to volume ratios and low efficiency.

The rotary regenerator scores highly where recuperators and switched regenerators fall down. Due to its very high surface area to volume and the continuous nature of its operation its efficiency is very high. Rotary regenerators also occupy less space than recuperators for the same heat transfer performance.

Regenerators with metallic rotors, or large diameter low temperature ceramic devices are widely used, particularly for power station air preheaters. The potential for ultra-compact rotary regenerators was realised by the automobile industry for use in gas turbine powered vehicles. As conventional metallic regenerators have been applied to higher temperature situations, special problems have been encountered. As the rotor undergoes thermal cycling of several hundred degrees, so it expands and contracts and as this distortion occurs it becomes very difficult to maintain the necessary radial and circumferential seals to isolate the two streams. Due to the potential gains, research was initiated into new, low thermal extension materials to replace metals.

Over the past 30 years, a great deal of research has been implemented into ceramic rotary regenerators for use in gas turbine applications and more general waste heat recovery, for example in the aluminium industry[208-210]

7.2.2 Materials and Design Development

A development programme on glass and ceramic materials for this type of application was begun as early as 1952 by Corning Glass Works of New York, having seen the potentially low cost and improved properties of ceramics over metals in heat regeneration. The earliest regenerator matrices (or "cores") were composed of bundles of Pyrex, and later "Pyroceram" tubes, bonded with cement or fused together. These designs simply could not withstand high temperature operation, even at 800°C.[211]

Around 1960, Corning developed a new material, Cercor, which was manufactured in thin, flat or corrugated sheets, which were subsequently wound around a solid hub, forming a matrix with typical passage diameters of about 1 millimetre.

This design is currently the basis of most regenerators for automotive gas turbines and industry.[212] Sizes for industrial regenerators are up to 70 inches diameter and 5 inch depth, accepting waste heat up to a claimed 1080°C.

Up to 1967 the main difficulties were related to material instability rather than mechanical design aspects. Aside from materials problems, design aspects have proved troublesome in relation to sealing, plugging of the matrix and large, avoidable, thermal and mechanical stresses.[213-216]

Sealing experimentation has involved the use of both clearance and rubbing seals. Early attempts were disastrous resulting in enormous leakages. Wear faces have been tried, utilising many different materials, with nickel oxide the most promising[217-218]; however, at temperatures of 760°C, this could no longer be used.

Thermal and mechanical stressing was a major problem due to the design of the drive and mounting of the matrix. The early designs were mounted with a horizontal axis; this will increase stresses due to weight concentration and distribution. Later designs are mounted with a vertical axis, eliminating many tangential and hoop stresses.

Driving the matrix has also proved a problem, with a rim-drive proving most successful. The early designs incorporated a heavy cast circumfernetial rim, fused to the outside of the matrix to provide a strong mounting element for attaching the driving gear. Due to differential thermal expansions this design caused peripheral cracks in the matrix circumference with subsequent failure as the matrix jammed against the cast rim.

More recent designs have incorporated semi-circular cast ceramic pins, fused at intervals into the periphery of the matrix, through which the drive can be transmitted. Ford have developed a system where they can bond the drive gear directly to the matrix periphery with a silicone elastomer.[219]

The main problem with CRR's has been the matrix materials. The basic compositions tried so far have been Lithium Aluminium Silicate (L.A.S. or Cercor), Magnesium Aluminium Silicate (M.A.S. or cordierite) and Aluminium Heatite (a development of L.A.S.).

Lithium Aluminium Silicate (LAS), marketed by Corning Glass as Cercor, (Fig.7.3-phase diagram) is based on B - Spodumene ($Li_2O.Al_2O_3.4SiO_2$). It is quoted[214] as having a melting point of 1300°C and a thermal expansion of only $0.6 \times 10^{-6}\ K^{-1}$.

The low melting point of LAS is enough to eliminate it, and this is coupled with severe sulphuric acid corrosion above 700°C.[210]

An ion-exchange reaction, Li^+ for H^+ occurs when LAS is contacted by H_2SO_4, resulting in contraction and eventual fracture. SInce sulphur products are to be expected in the application environment, this would be a serious drawback.

A development of LAS, utilising the acid reaction, is aluminous heatite, code 9460, formed by treatment of LAS with sulphuric acid. This has similar properties to the parent LAS compound, but at temperatures over 1000°C transformations to mullite and cristobalite occur. Despite its improved corrosion resistance, those transformations, and firing and manufacturing difficulties eliminate Aluminous keatite as a candidate material.

Magnesium Aluminium Silicate (MAS), otherwise known as cordierite, $MgO.2Al_2O_3.5SiO_2$, is variously quoted as being impervious to chemical attack, and as being leached by Sulphuric acid under certain conditions.[220]

Cordierite melts at 4160°C, and some sources [219] rate its maximum working temperature as only 1000°C, due mainly to its high thermal expansion. MAS is therefore a very good material for regenerators up to 1000°C but its properties are not adequate for extended use at 1300-1400°C.

It can be summarised that LAS and MAS suffer from sulphuric acid attack; MAS also suffered from thermal instability. Aluminous heatite has proved difficult to manufacture in a dense strong form in bulk, and all of these types would not be refractory enough for operation above 1200°C.

So, material selection remains of great importance, especially for applications at higher temperatures than those experienced in gas turbine regeneration.

7.2.3 Market Needs

Before embarking on a major development programme of work it is necessary to assess its market needs. It is well known that the steel, aluminium and glass industries discharge gases at very high temperatures (> 1000°C) from its furnaces.

In assessing the BSC operations alone, it can be seen (Table 7.4) that the 27 mills with throughputs varying from 165 to 600 tonnes per week discharge exhaust gases at around 1100°C with heat recuperation providing air preheats at around 310°C. Improved recuperation through a ceramic regenerator can offer air preheats of 770°C offering energy savings of as much as 40 per cent. On 1981/82 fuel prices this would offer energy savings of £1.2 million per year over and above the existing recuperation savings, thus making the project worth pursuing.

7.3 BSC - CERAMIC ROTARY REGENERATOR (CRR) DESIGN

The basic concept of the rotary regenerator is simple (Fig.7.4). Heat is transferred by storage in, and recovery from a porous ceramic matrix which is rotated across the flow paths of hot waste gases and cold combustion air. In pracitce, the enrgy storage matrix takes the form of a disc rotor and the gas streams are arranged to flow through the rotor in a counter current mode. These rotor structures are subject to complex non-symetric thermal loading which results in both high hoop and tangential stresses within the matrix. Therefore the selection of matrix material and engineering design is an important part in the successful development of a rotary regenerator.

Already we have seen a number of ceramic regenerators fail due to either the wrong selection of ceramic material or/and due to engineering design failures such as rotation of the matrix on the horizontal axis, high rotational speeds. etc.; thus increasing both the hoop and tangential stresses on the matrix. The existing moving regenerators provide a compact heat exchanger but it is difficult to seal sufficiently to prevent leakage loss to the environment and between exchanging streams.

BSC in their current development have given considerable thought to the selection of matrix materials and engineering design concept and this is discussed in detail in the following pages.

7.3.1 Material Demands

In selecting matrix materials it is established that the ceramic selected must have certain physical and mineralogy properties to operate within the prescribed temperature band and be able to withstand the hostile environments of its application. The main properties thus required are outlined below:

A. Refractoriness and Strength

The material will be required to operate at hot face temperature of around 1300°C. Cold face temperature would be about 600°C, and load-bearing members would be located in this colder region of the rotor assembly.

A stress analysis has indicated that tensile stresses of 30MN/m^2 and compressive stresses of 26MN/m^2 of thermal origin can be expected.[221-222]

The requirements are therefore for a material which retains its fracture strength well above these limits when operating in temperature range of 0-1400°C.

B. Slag attack resistance

The material must be inert to the air stream and waste gas stream. It must therefore be oxidation resistant at all temperatures, and be able to withstand attack by a waste gas containing CO, CO_2, Ferrous oxides, sulphur oxides and acids, alkali metals and their compounds and Vanadium Pentoxide. These compounds may exist in vapour or liquid form, and at temperatures between 600 and 1400°C.

Good corrosion resistance is essential as attack will first affect the surfaces and result in lower abrasion and thermal shock resistance, due to decreased fracture strength.

C. Thermal shock resistance

Good resistance to sudden temperature change and thermal gradients is required due to the thermal cycling and asymetric heating of the rotor.

In steady state operation the maximum temperature change would be expected at the cross over point from the waste gas outlet side to the air inlet side where air at ambient temperature impinges upon a matrix at about 500°C. A BSC computer performance prediction programme indicates a drop of about 50°C across the face with a maximum temperature gradient of about 130°C over a 300 mm radius (Appendix A)

D. Abrasion Resistance

The rubbing seals must wear preferentially to the matrix material, and since the seal material must be very resistant to abrasion. This problem is aggravated by the perforated structure of the matrix, and a Mohs hardness of at least 7 would be desirable.

E. Fabriaction

Requirements for the regenerator matrix are that the wall thickness and passage diameters are as small as possible, for optimum heat transfer; this must be balanced against a high probability of plugging with small passage diameters. These considerations indicate that a method of fabrication capable of high dimensional accuracy, the production of long thin passages and uniformity of density is required. Typical dimensions would be between 8 and 15 mm for passage diameter and between 5 and 15 mm for wall thickness. These requirements impose certain restrictions on fabrication. When coupled with the need for a smooth surface finish, high strength, controlled particle and pore size and low cost; the problem is considerable.

Some of the possible methods of fabrication are thus described below:

Slip-casting: Many well established techniques exist, the oldest of which is slip casting. This method involves the combination of water with raw materials to form a slurry which is then cast in a porous mould. The mould removes most of the water, and the product is then dried. This technique is not suited to the production of articles with complex internal geometry.

Dry Pressing: Dry pressing involves the compaction of a mix of raw materials with a small percentage of water to promote a degree of plasticity.

Vibro-casting: Vibro-casting is a method of dry pressing carried out in a vibrating mould which ensures densification. This is another well established technique offering high strength, uniform structure, good dimensional tolerance and the ability to mould complicated shapes.

Isostatic-pressing: This method uses a rubber mould in a liquid medium through which the compacting pressure is applied. The technique again cannot produce complex internal geometries.

Extrusion: Extrusion techniques, involving the extrusion of a plastic mix offers uniformity, close tolerances, good finish and speed of manufacture - but at a high initial tooling cost.

The most suitable techniques for producing long thin uniform passages would appear to be extrusion and dry-pressing (preferably with vibro-casting).

G. Other properties

In addition to above, consideration has to be given to the following properties of ceramics when selecting materials.

Thermal expansion

For accurate sealing, thermal expansion must be low; thermal expansions range from $3 \times 10^{-6} - 11 \times 10^{-6}/^{o}C$ for materials considered. This gives an expansion of between 2 and 9mm on an 800mm matrix, an acceptable limit being about 8 x $10^{-6}/^{o}C$.

Porosity

This has various effects on other properties of the ceramics such as, for good corrosion and abrasion resistance. To avoid plugging low porosity is preferable, but for good thermal shock resistance high porosity is preferable due to "stopping" of crack propagation.

Porosity may also have an effect on heat transfer e.g. high porosity would increase turbulance and heat transfer, but would also increase the pressure drop across the matrix.

Density

It is desirable that the rotor does not exceed weight limits dictated by the engineering design aspects of the installation.

Thermal conductivity

To improve thermal shock resistance and heat transfer characteristics of the rotor a high thermal conductivity would be desirable.

Specific Heat Capacity

A high specific heat capacity would seem to be required but is not a major consideration.

7.3.2 Material Selection

An extensive materials review was undertaken (223-229) and some materials were immediately rejected on grounds of poor strength, or refractoriness or other properties. These include ceremets (due to their irregular thermal expansion), carbon refractories, fireclay, lime and many ternary and quaternay oxide systems. Other materials such as LAS and MAS were also eliminated due to their earlier failures.

Table 7.5 shows a summary of a group of materials, consisting of several "special" super-refractories. e.g. Zirconium diboride and some cheaper traditional refractories together with the reason(s) for their rejection. They range from expense to poor corrosion resistance. Many of the special refractories require fusion casting or high temperature sintering for manufacturing rotor blocks, which is difficult and expensive, and thus have been rejected.

The selection programmme has thus shortlisted 3 materials having potential for further investigation, and possible utilisation; i.e. aluminosilicates with a range of Alimina content, Sialons (including silicon nitride) and silicon carbide.

A. Alumino-Silicates

Members of the Al_2O_3 - SiO_2 binary system have been the basis of refractory technology since its earliest beginnings. As a result there is a large amount of manufacturing and

and technological experience available in this field[230-233]

As one could deduce from the phase diagram (Fig.7.5) a material with certain properties can be obtained by selecting the correct proportions of alumina and silica. Alumina-silicates can be classified into four categories according to their chemical composition;

Percentage Alumina	Type
1 45-65	Sillimanite, and other $Al_2O_3.SiO_2$
2 65-75	Mullite
3 75-90	Bauxite based
4 90-100	Corundum

Category 1 materials are generally made from minerals of composition $Al_2O_3.SiO_2$, such as sillimanite, andlusite and kyanite. The usual manufacturing method involves crushing and grading the ore, mixing with claybond, forming, drying and firing. The sillimanite type materials form mullite and cristobalite (a form of silica) when heated above 1300^oC.[234]

Category 2 materials are manufactured either from natural mullite ($3Al_2O_3.SiO_2$) ore or from synthetic mullite. To increase alumina content bauxite ($Al_2O_3.H_2O$) is added to the raw materials.

Table 7.6 shows some representative examples of a range of alumino-silicates.

Refractoriness and Strength

There is a general trend of increasing strength and refractoriness with increasing alumina content throughout the Al_2O_3 -SiO_2 system. These properties are subject to considerable variation depending on the method of manufacture, microstructure and testing.

Some general results are shown in Fig. 7.6 and 7.7. Great improvements in strength can be made by promoting a continuous, well developed matrix of mullite, rather than a discontinuous multiphase structure.[235] This is achieved by careful selection of particle sizes, moulding pressures and firing routines, so that high alumina products can attain even greater superiority over low alumina materials.

Slag and corrosion resistance

This group is considered to have excellent slag and corrosion resistance which further improve with increased alumina content, until pure alumina is reached, which is claimed to be impervious to most chemicals.[236-239] Research indicates that mullite content improves alkali resistance.[240] (Fig. 7.8).

Thermal shock resistance

The best product for thermal shock properties will be a compromise between strength and thermal expansion, since other relevant properties do not vary greatly.

This property can be predicted using certain parameters known as R-values . The value of R is dependent on Young's modulus, fracture stress, thermal expansion, Poisson's ratio and thermal conductivity of the material. A study of alumina refractories[241-242] indicate that the best quality specimens are obtained in the 70-72% alumina region. Products with less than 50% and more than 72% alumina have a reduced resistance due to higher thermal expansion. (Fig. 7.9 and 7.10)

The free silica content of a product has a detrimental effect on thermal shock resistance. Indeed it is said that free silica in a mullite is the direct cause of thermal shock failure.[204] Thus the ideal product would be fired to remove all free silica and consist of a continuous mullite matrix, with a low enough percentage of corundum to avoid an undesirably high thermal expansion.

Thermal expansion

The effect of thermal expansion on thermal shock resistance has been discussed above. The thermal expansion coefficient for various ceramic oxides is presented in Fig.7.10 and the effect of alumina content in thermal expansion/resistance is given in Table 7.7. An expansion of up to 8×10^{-6} at $800^{\circ}C$ would be acceptable, and this limit includes those materials with up to 80% alumina content.

Abrasion resistance and hardness

Both these qualities improve steadily with increased alumina content, but are dependent on manufacturing methods. In general, low alumina products (40%) have a Mohs hardness of 6, and corundum has a hardness of 9, and is very resistant to abrasion. Cold crushing strength correlates closely with abrasion resistance. (Fig. 7.7.)

Other properties

(a) The thermal conductivity of most alumino-silicate is around 2-3 $Wm^{-1}\ K^{-1}$ at $1000^{o}C$ and this does not increase greatly until pure alumina. Table 7.6. (Fig. 7.11).

(b) The specific heat capacity of most materials is almost constant at elevated temperatures. So little variation is seen amongst alumino silicates. (Fig. 7.12).

(c) The density of these materials is rather dependent on fabriaction methods, but the general trend is an increase in density with alumina content. In the materials selected all these densities are well within design criteria.

(d) Increased porosity reduced alumina strength (Fig.7.13). Porosity is also dependent on the method of fabrication. It is easier to achieve lower porosity with higher alumina materials than with low alumina materials.

Fabrication

Alumino-silicates can be manufactured by most methods and no difficulty should be experienced in dry pressing or extruding. Difficulty of manufacture does, however, increase with increased alumina content. i.e. 90% alumina bodies must be hot-pressed or sintered. For materials containing up to 85% alumina, the main fabrication difference will be in the firing temperatures i.e. higher alimina products requiring progressively higher firing. Typical temperatures of 1350°C for 80% alumina though, in general, firing must be high enough to ensure that no free silica is left and that the mullite content is well developed.

Therefore it can be summarised that an alumino-silicate containing only mullite and corundum would be the best compromise of strength, corrosion resistance, thermal shock resistance and ease of fabrication. This composition would be achieved with a 70-80% alumina product. Above this content, thermal shock resistance decreases and manufacturing is more difficult. Below this range, the refractoriness, strength and corrosion resistance become inferior.

B. Other shortlisted materials

This group comprises the various forms of silicon carbide, silicon nitride and Si-Al-O-Ns. A vast body of research has been undertaken into these materials and their methods of manufacture both by the aerospace and gas turbine industries.[243]

The research reveals that this class of materials has outstanding properties of strength, refractoriness, abrasion resistance and thermal shock resistance. Usual methods of manufacture are hot pressing or sintering, and these are not as suitable as extrusion or dry pressing for the manufacture of large matrix blocks. Hot pressing involves temperatures of about 1600-1700°C and reaction-sintering usually takes place at over 1400°C.

Refel Silicon carbide however is claimed to be fabricated by many cold moulding methods, although these can only produce articles of up to 280 mm diameter[242].

Alumina-bonded silicon carbide, containing 3-10% clay bond, retains many of the parent silicon carbides excellent properties with easier manufacture and improved oxidation resistance. However, the clay bond is a weakening factor, and complicated behaviour occurs in high temperature oxidising conditions.

The major drawback with this class of ceramics is their poor oxidation resistance. Research is continuing and Sialons (compounds consisting of alumina and other molecules in a B-Si_3N_4 matrix) may prove to be a rewarding field. At present no material in this group challenges the alumino-cilicates for overall suitability, but advances may be made to produce a very promising material.

<u>Conclusion</u>

After considering a very wide range of materials, the most promising proves to be one of the most common of the traditional refractories - an alumino silicate of around 65-80% alumina content. The only other promising candidates silicon nitride and carbide and sialons are ruled out due to severe oxidation problems.

However, careful watch should be placed on the aerospace and gas turbine industries for further developments in oxidation resistance in this field, where it is widely hoped that sialons, by selection of different molecules to be incorporated in the B-Si_3N_4 structure will become a very strong, tough material well suited to our needs though at a cost.

Having specified the material choice, the final selection of particle size, moulding methods and firing temperature must be made in close co-operation with the manufacturer. Although Mullite (60-80% AlO) was considered to be the most promising candidate, due to manufacturing difficulties in extrusion to meet the design tolerances required, a material containing only 45% alumina was used for the pilot model of the CRR. It is hoped however that with advances in production technology a matrix of higher alumina content can be developed.

7.3.3 Rotor Assembly Design

Considering a general design of a ceramic rotary regenerator the energy storage matrix takes the form of a disc-like rotor and the gas streams are arranged to flow axially through the rotor in a counter-current mode. Hence rotor structures are subjected to complex non-symetric thermal loading which results in high hoop and tangential stresses whithin disc matrix structures. For this reason and for ease of fabrication, installation and maintenance, large-diameter rotors are usually designed as multi-element matrices assembled within metallic cage support structures.

The BSC rotor assembly design uses an assembly of rectangular sectioned heat storage blocks supported on a strategically positioned metallic grid. The assembly is mounted in a horizontal plane and is rotated about a vertical axis by a drive mechnism acting on the cirumference. By arranging for the two gas streams to flow in a counter-current mode with the hot gas flowing axially downwards, a relatively cold region is obtained at the bottom end of the rotor and it is in this cold region that the metallic support grid is located. Mathematical modelling studies have shown that, even for high-temperature heat recovery duties, suitable sizing of the rotor and heat storage elements will produce adequate cooling at the bottom end of the rotor to allow the use of stainless steel for the support grid structures.

The ceramic blocks are supported in the rotor assembly by square-sectioned metallic bars which are passed through V grooves cut in opposing faces of the blocks. The metallic supports form a simple parallel bar grid which runs in the rotor plane near the bottom of the rotor (Fig.7.14). V Grooves are used to ensure that the crushing load at the bar/block interface is spread over a large area and also that stress concentration problems in the load-bearing regions are minimised. The parallel bar grid is simply supported on the rotor rim ring structure (Fig. 7.15) and the entire block assembly is compacted by mechanically or pneumatically actuated radial compression rams. For large-diameter rotors, support grid creep strength limitations will require the use of auxiliary support beams to reduce the unsupported span length in the parallel grid system (Appendix B). The rotor can be divided into quadrants, as shown in Fig.7.16, by rectangular beams which are simply supported and located circumferentially on the rotor and on a hub support. Each quadrant can thus carry individual support grids as illustrated. Blanking ceramic blocks are employed to protect the auxiliary support beams from exposure to high-temperature gases.

7.3.4 The Sealing System

Allied with the need for a composite rotor is the requirement for a flexible sealing system to prevent excessive leakage loss at the interfaces of the rotor and header systems. On the BSC high-temperature regenerator, the seal assembly on each face consists of a circumferential barrier against loss to the environment and a seal acting radially between the two gas streams (Fig. 7.17). Interface flexibility is achieved by using barriers composed of assemblies of seal blocks which are arranged in juxta position along static guide members and are biased into rubbing contact with the rotor face. Seal block size will, in the longitudinal dimension, be determined by the degree of rotor distortion or irregularity and in the lateral sense by matrix pore size and pitch.

Seal system conceptual design work has resulted in applications for patent cover on two types of seal block element[244-245]. In the first of these designs, piston-like blocks are arranged to ride, in lap-jointed contact with each other, in channel-shaped water - or air-cooled static housing members (Fig.7.18). The effectiveness of such a system will depend on the quality of surface finishes and the degree of control over guide channel and seal block mating surface dimensions.

The alternative flexible seal design envisages the use of blocks which ride as shoes on the outside of static guide rails. The main features of the design are illustrated in Fig. 7.19. The blocks, which may be fabricated in ceramic material, are arranged to ride on rectangular-sectioned guide rails which may be air - or water-cooled as required. In the case of blocks fabricated in ceramic material, a metallic lining may be employed to give improved cross-sectional bending strength and to reduce friction and wear at the shoe/guide rail interface.

Before embarking on the design of seal systems for high-temperature performance assessment, a cold test rig was designed to evaluate the concept of a multi-element piston block assembly functioning as a seal barrier.[246] Fig. 7.20 shows schematic details of the rig test section which consists of a cylindrical chamber made up of a cover bell which bears on a circumferential seal on a rotor base plate. The bell is fitted with a seal block guide structure which provides location for a radial barrier of seal blocks acting in a plane perpendicular to the base plate and in so doing divides the chamber into two compartments. Barrier evaluation can thus be carried out by leakage flow measurements between the two compartments, at various pressure differentials.

A test rotor surface is provided by a 280 mm diameter steel plate which is supported on the rig turntable and is driven by an infinitely variable speed motor. The working rotor area is a 250 mm diameter central section on the plate surface which, in the case of this investigation, was dome-profiled to simulate the thermal distortion of hot rotor surfaces. A circumferential seal between the bell and the rotor was effected by mating a shoulder, machined on the inside of the rim of the mild steel cover bell, against a 3.5 mm width of PTFE bearing material bonded round the periphery of the central working area of the rotor plate. The assmebled chamber is provided with a spring-loading arrangement on the cover bell which ensures the efficiency of the circumferential seal arrangement over a wide range of test pressures.

Table 7.8 summarises block set test conditions for the entire test series. Results obtained with Type A and Type B block sets are presented in Fig. 7.21 and 7.22 respectively, with leakage data being quoted in terms of mass flow rate per unit length of seal. The actual significance of these results with respect to regenerator seal performance on a soaking pit application may be illustrated from the following estimations of leakage loss rate from the air system on a full-sized rotor.

Soaking Pit Application:

Air flow rate:	3 kg/s
Air pressure:	1.03 bar gauge
Waste gas pressure:	1.0 bar gauge

Atmospheric pressure: 1 bar gauge.

Regenerator port area ratio: 1/1

For operation at flux densities of 1 $kg/m^2.s$:

Rotor diameter = 2.75 m

Total air system seal length = 2 x 2.75(3/2 + 1)

= 14.2 m

Air to atmosphere pressure diff. = 25.5 m bar

Air to waste gas pressure diff. = 25.80 m bar

From the performance curve for test M1 at P = 25.8 m bar

Leakage = 5.2×10^{-4} kg/s/m

% Leakage = $(5.2 \times 10^{-4} \times 14.2 \times 100/3)$

= 0.25% of total air flow

From table 7.9 it can be seen that, with the exception of tests K4 and K5, leakage loss rates, at a pressure differential of 25.80 mbar, are below the equivalent of 1.0% of total air flow rates. This compares favourably with the level of 10% assumed for preliminary thermal design work on high-temperature regenerator systems.

Tests K4 and K5 were performed with the block set having zero end float in the seal housing. This lack of flexibility could cause interference problems in the lap-jointed block set and so could have been responsible for the relatively poor performance recorded in these two tests.

For a high-temperature installation, the piston blocks will have to be produced in a ceramic material and the housing structure will require cooling to control thermal distortion to permit the use of low-cost materials. Because of surface finish and dimensional tolerance limitations encountered in ceramic fabrication work and because of thermal distortion problems, a high-temperature piston seal barrier will not be capable of achieving the performance levels obtained in these tests. Also, with the high-temperature rotor being produced as an assembly of matrix blocks, there will be a certain amount of additional leakage between matri blocks which will depend on the effect of thermal distortion on mating surfaces within the rotor assembly.

Recognising the problems associated with rubbing seals (i.e. wear) an alternative system of proximity or close tolerance seal is assessed.[247] A test rig was designed and fabricated in wood. A cross-section of test rig together with the flow diagram is presented in Fig.7.23.

To calculate the leakage rate through the seal the volume-in equals the volume-out, if pressure remained constant, is assumed. Therefore a measure of volume required to maintain a pressure on the other side will provide a leakage rate.

Tests were carried out maintaining an air flow rate of 2000 liter/min. Differing seal widths (60mm, 90mm 120mm) were used to assess the effect of seal widths on the leakage rate, as it is generally accepted that an increased seal width will improve sealing. However the results obtained showed little relation to the above concept except at high pressure differences (Table 7.10).

Using the results, leakage rates for a typical 3 m diameter CRR commercial installation are estimated (Fig.7.24) and the method of calculation is presented in Appendix C.

From the results it can be concluded that though for a 3 m diameter rotor with a gap of 3mm, an acceptable leakage rate of 9% can be produced (Higher than from rubber seals), it is unlikely that the matrix even in its cold state can be manufactured within these tolerances. Taking account of the thermal expansion when hot these gaps are inadequate to consider. In addition the leakage rates predicted are much higher than the ones obtained in rubbing seal concept.

Therefore rubbing seal was accepted as the best to pursue.

7.3.5 Seal Wear

A. Introduction

In the BSC design of a ceramic rotary regenerator, sealing is achieved through the rubbing action of static ceramic shoes onto the rotating ceramic matrix. In such a system, it is obvious that ceramic wear would occur both due to abrasion and adhesion.

In a successful CRR design it is essential that this ceramic wear should be minimum. Furthermore, due to the high costs of matrix, any such ceramic wear should preferentially occur on the ceramic shoe rather than the matrix material.

The preferential wear of shoe material can be achieved through the correct selection of shoe material related to the matrix material. However, data is needed to assess the total ceramic wear in such a system so that the life expecatncy of the ceramic shoe could be predicted.

This part of the thesis describes the experimental procedure and then assesses the results obtained.

B. Ceramic Wear

(a) Abrasion Wear: Fragments of a moving charge or dust particles carried in the fast moving waste gases, impinge on the refractory surfaces and thus produce an abrasive wear. The rate of wear can be given by:

Rate of Wear $\quad K \propto R$ —— 7.1

It is generally accepted that increase in gas temperature has little effect on abrasive wear (248) and in high alumina ceramics the abrasion wear is minimum.

(b) Adhesion Wear: The rubbing of two surfaces through adhesion produces the major wear. Similar to metals it is very much dependent on the friction coefficient of the material.

i.e. $$H = \frac{F \ell \rho}{m} \qquad \text{—— 7.2}$$

and $$G = \frac{m}{S \rho L} \qquad \text{—— 7.3}$$

$$\text{Wear Rate} \propto \left[\frac{K \tan^2 \theta}{H} - \frac{1}{E} \right] \qquad \text{—— 7.4}$$

where K is independent

and θ is the angle of contact

From the above it can be said that the wear rate is very much dependent on the material hardness, grainsize (i.e. density) and on the load per unit area and thus can be given as:

$$\text{Wear Rate} \propto \left[\frac{Gs^2}{H} - \frac{1}{E} \right]^{T,L,M} \qquad \text{—— 7.5}$$

$$= K.\ T.\ L^2.M \left[\frac{Gs^2}{H} - \frac{1}{E} \right] \qquad \text{—— 7.6}$$

Increasing the ambient temperature can frequently result in an increase in friction coefficient and thus increased wear (Fig. 7.25.)[249] For some alumina ceramics there is evidence that in rubbing, the dust produced in the early stages adheres to the surface and thus prevents any further abrasion (polishing effect). There is also evidence that through the selection of nature and grain size of a refractory material, the wear can either be retarded or accelerated.[250]

C. Experimental Procedure

The apparatus consists of an electrically heated furnace. The matrix block is placed rigidly inside the furnace using refractory brick and is shielded against radiation from the furnace heating elements. The shape and size of the ceramic block and shoe under investigation is shown in Fig. 7.26. The shoe is fixed onto a shoe holder, made of stainless steel (Immaculate 5), using a refractory adhesive cement called 'Fortafix Curomix'. The shoe holder is linked to a set of rollers which in turn is attached to a cam connected to gearbox drive unit (Fig.7.27). This mechanism causes the shoe and holder to reciprocate on the surface of the matrix block.

The temperature of the furnace is controlled and can either be kept constant or varied during the test period as required. The shoe is made to slide on the matrix surface at a constant speed. The duration of each test was varied and the results are presented in Table 7.11.

A temperature range of 250^{o}C to 700^{o}C was covered, which is supposed to provide the maximum wear in the alumina based refractories wa covered in these earlier tests.

Little published information is available on the effect of ceramic abrasion at high temperature. Since the unit being developed is to operate at high temperature, ceramic abrasion and hence seal shoe wear was also assessed, for the selected silliminite based materials, at these high temperatures. The results of these tests are presented in Table 7.12. Because of the high temperature operation each ceramic sample was connected on to a firebrick using 'Monocon Super 80' cement and every effort was made that the surfaces of the sample to be tested were clean and free from any cements.

The shoe and matrix were both examined and wear measurements taken at the end of each test. A typical wear calculation is presented in Appendix D.

D. Discussion and Conclusion

Contrary to belief that hard materials with fine grain size (mullite) offering a minimum of coefficient of friction would offer a minimum of wear, this was not substantiated. Fine, hard grain size during the rubbing action at high temperatures tended to dislodge from its top surface and, with no escape, tended to rub into the opposite surface making a gouging effect. (Fig.7.28) Microstructure analysis of the sample clearly confirmed this (Fig.7.29). Therefore a true wear that may be experienced in the pilot plant will very much depend on rate and the passage fine powder produced will follow. Given the circumferential movement of the holed matrix together with gas flows (i.e. some gas leakage) it is possible that acceptable wear will be ahcieved.

The results also show that the wear rate increases with the increase in temperature reaching maximum at around 1100°C and then decreases with temperatures for the silliminite materials tested (Fig.7.30). High alumina coarse bonded material against silliminite offered a minimum of wear.

The results show that acceptable wear rates were achieved with the material selected. A prolonged test period shows a decrease in wear rate. This could be due to the polishing effect of the ceramic as discussed earlier in this chapter. While loading had a marked effect on the wear rate, temperature variations play very little part at lower temperatures.

A typical wear test calculation shows (Appendix D) that for a full scale application with a rotor of diameter 2.5 m, a seal shoe life expectancy of 632 days would be expected.

As a result of these tests, to obtain a minimum of wear and thus a prolonged matrix life, it is recommended that compressed high alumina ring segments be instituted on the periphery of the matrix face.

Therefore, it can be concluded that the material selected provides an acceptable seal wear rate with the design recommendations of changing the worn shoes at the annual inspection/maintenance of the regenerator.

7.3.6 Computer Program

As an integral part of the development of CRR it is essential that a computer program is written which will provide a mechanism for evaluating various empirical designs of the CRR and thus offer an optimised design.

A: Objectives

The main objective of the program is to predict the performance of a given rotary regenerator in the form of thermal recovery (i.e. preheat) and leakage factor for various operating conditions such as rotational speed, waste gas temperatures, flux density, etc.

B: Computer Program

The body of the program consists of a suite of three sub programs defined as STIR, MASH and POUR. The function of each program can be described as follows:

STIR PROGRAM

- reads the input data and records the relevant items
- calculates BLOCKAGE, PASSAGE LENGTH, AIR PERIOD
- provides output STIR data file to MASH & LINE PRINTER
- calculates leakages
- alters air and gas flux densities accordingly
- provides leakage file outputted to MASH

MASH PROGRAM

- procedure CP WALL - This is the main body of program and comprises : calculates specific heat of material
- procedure DETERMINE GAS PARAMETERS - calculate ECOEFF (emissivity coefficient independent of temperature) from gas mass fractions
- produce CALCULATE FACTORS - calculate the Hausen coefficient using calculated values of convective heat transfer coefficient, radiative heat transfer coefficient and the friction factor.
- reads in STIR data and leakage data
- determines initial matrix temperatures in the selected steps through the rotor.
- calculates CRCA & CRCG and sets relocation parameters and timesteps
- sets up initial boundary conditions.
- enters flow cycles and calculates S, SS & T

- enters gas temperature iteration and compensates gas temperature for leakage
- gas cycle
- enters gas flow distribution iteration
- enters gas pressure equalisation iteration
- calculates initial pressure
- calculates the viscosity, specific heat, thermal conductivity and density of the gas at iterated temperature from library progress for each axial step
- calculates refractory specific heat
- calls CALCULATE FACTORS and from the value of Hausen HT COEFF and the above values of the variables, the gas and matrix temp changes can be calculated and correspondingly temperatures for the next axial step (gas temp change) or the next gas timestep (matrix temp change) are predicted
- calcualtes the corresponding pressure change and pressure
- exit gas pressure equalisation iteration
- calculates pressure drop and designates gas outlet temperature
- designates the initial pressure drop to equal reference pressure drop
- calculates pressure drop ratio as ratio of pressure drop to reference pressure drop
- repeats calculation with relaxed gas flux density if pressure drop ratio not within $\pm$ 0.5%
- exits gas flow distribution iteration
- compares sum of individual gas flux densities with mean gas flux density used initially and if not within 0.5% repeats gas flow distribution iteration using modified GFD

- enters air cycle
- enters airflow distribution iteration
- calculates the viscosity, specific heat, thermal conductivity and density of air
- calculates the specific heat of rotor
- call CALCULATE FACTORS and from the value of convective heat transfer coefficient and the friction factor and the above values of the variables, the gas and matrix temp changes are calculated in the appropriate direction of iteration - as per gas cycle except that radiation is ignored.
- exits air pressure equalisation iteration
- exits airflow distribution iteration with the same qualifications as the gas cycle
- CONVERGENCE CHECK carried out using relaxation factors and matrix temperatures. Cycle count to be 180 before No Convergence
- calculates CARRY-OVER FLUX to give the amount of gas/air entering the matrix but exiting in the air/gas duct due to the revolution of the matrix
- calculates MEAN TEMPERATURES
- calculates EFFECTIVENESS. AIR - the ratio of the enthalpy change of air preheated to preheat temperature with the enthalpy change of air heated to gas inlet temperature. GAS - the ratio of enthalpy change of gas from exit to inlet temperatures with the enthalpy change of gas from air inlet to gas inlet temperatures
- refractory utilisation calcualted - this is a measure of the ratio of the heat extracted from the matrix by the air over the matrix weight

POUR PROGRAM

- Sets of the MASH output

Detailed program together with theoretical assumptions and approximations used is presented in Appendix E.

During the workings, it was found that because the program was developed using sub-routines from BSC Recuperator program, it is slow and cumbersome. Therefore a similar but simplified program developed by Dr. Wilmott from York University was purchased. The program is modified to provide similar results as that of the BSC Program.

A test data run showing the design performance of a 3 m diameter CRR (Commercial Unit) is presented in Appendix E.

7.4 THERMAL DESIGN OF A REGENERATOR

7.4.1 Introduction

The successful development of a compact high - temperature regenerator system hinges on the solution of mechanical problems related to matrix structures, the efficiency of sealing systems, thermal designs of the regenerator and finally its compatability with the installation environment. We have so far looked at the materials problems and developed a novel sealing system. With the help of the computer program developed we are now in a position to provide a thermal design for the rotor.

A rotary regenerator has a wide range of variables which influence the final level of its performance. It is not feasible to consider all the possible combinations of design in this report. However, effort has been made to cover as many likely combinations as possible, in sufficient depth, to allow a sensible estimate to be made for any particular design.

7.4.2 Design Parameters

The design parameters are defined as those that constitute

A: The physical shape and design of the rotary regenerator i.e.

1) Rotor speed (n) in revs per minute

2) Length (ℓ), meters. Taken as the length of the air or waste gas passage through the rotor and can therefore be considered as the depth of the rotor.

3) Hydraulic diameter (d) meters. The hydraulic mean diameter of the individual matrix holes. The holes can be of any shape consistent with ease of manufacture and the basic mechanical integrity of the rotor. Items (2) and (3) are usually combined to give a length/diameter ratio as one of the primary performance control parameters.

4) Wall thickness (t) meters. Taken as the minimum material thickness between any two adjacent matrix holes.

5) Air port area (A), m^2. Defined as the working face area of rotor inside the seals that is subjected to the airflow.

6) Air to gas area ratio (λ). The ratio of the air port area to the waste gas port area.

7) Matrix block material, which in turn specifies the material conductivity and density.

and B: the required inlet flow conditions i.e.

1) Air flux density (λ_A) kg/sm^2. The air flow in kg/s per unit air port area.

2) Gas flux density (λ_G) kg/sm^2. The waste gas flow in kg/s per unit waste gas port area.

Both (1) and (2) use the working face area of the rotor. (In some literature the flow flux densities use the actual flow area of the rotor, being the sum area of the holes in the matrix).

3) Waste gas inlet temperature (T_{gl}), ^oC.

4) Air inlet temperature (T_{al}), ^oC.

5) Waste gas inlet temperature (P_{gl}), N/m^2.

6) Air inlet pressure (P_{al}), N/m^2.

7) Waste gas composition, using the mass fractions of H_2O, N_2, O_2, CO_2, SO_2.

8) Leakage (L), kg/s. There are six potential leakage paths for waste gas and air as shown diagramatically on Figure 7.31 i.e.

   a) Hot face air to gas leakage through the central dividing seal.

   b) Cold face air to gas leakage through the central dividing seal.

   c) High pressure air leaking direct to atmosphere.

   d) Low pressure air leaking direct to atmosphere.

   e) High pressure waste gas leaking direct to atmosphere.

   f) Low pressure waste gas leaking direct to atmosphere.

Items (e) and (f) under certain operating conditions, where a waste gas stack is operating under suction or induced draught, may result in air being drawn into the waste gas through leaking seals.

7.4.3 Performance Parameters

The performance parameters are those defined from calculations using the design parameters and are essentially the output from the rotary regenerator mathematical model.

(a) Air side effectiveness (η_A). The heat recovered in the preheated air as a percentage of the theoretical maximum heat recoverable.

(b) Gas side effectiveness (η_G). The heat given up by the waste gas as a percentage of the theoretical maximum heat available.

(c) Solid/air capacity rate ratio (C_A). The ratio of thermal capacity flux of the rotor to that of air. This can be considered as the ability of the solid matrix to transfer heat to the air.

(d) Refractory utilisation (U), W/kg. A measure of the heat transferred per unit mass of refractory. This can be used as an assessment of the cost effectiveness of the rotor.

(e) Waste gas pressure loss (ΔP_g), N/m^2. The pressure drop through the waste gas side of the matrix.

(f) Air pressure loss (ΔP_a), N/m^2. The pressure drop through the air side of the matrix.

(g) Matrix temperature (T_x), ^oC. The matrix temperature is calculated for various planes through the rotor at a number of circumferential points, so giving a complete temperature picture of the rotor for any particular flow condition. For the BSC design of rotor the most significant temperature is the maximum temperature occurring at the cold face of the rotor since it is in this region that the matrix block support bars are positioned.

The calculated air preheat temperature and waste gas outlet temperature can be determined from the effectiveness values.

7.4.4 Thermo-mechanical Design of the Matrix

In sizing the heat storage blocks for a multi-piece rotor, factors such as the nature of typical steelworks waste heat sources, acceptable pressure drops, matrix fabrication limitations and the effectiveness of refractory material utilisation is considered. Steelworks high temperature waste gases tend to be laden with considerable amounts of slag and iron oxide debris which would tend to clog fine-pored matrices and thus cause an increase in flow pressure losses. This may be off-set to some degree by the self cleaning effect of the incoming counter flow clean combustion air in the rotating matrix. Further considerations of material suitability, ease of fabrication, and overall costs (on a consistent basis) dictates the selection of material, hole size and wall thickness of the matrix. Design of high refractory utilisation is important in that is implies designing for compactness and hence for lower overall plant costs.

In the designs below, each of the main design parameters are thus varied over a useful working range and the corresponding effects on performance are predicted using the model, assuming the air port area to be $2m^2$, an arbitrary figure since the flows are taken as flux densities.

A. Effect of Rotor Speed

The main effect of changing the rotor speed is a change in solid/air ratio capacity (CA). Earlier work[201] shows that air side effectiveness (η_A) is sharply affected if the absolute value of CA is dropped. Fig.7.32 shows that this change occurs when CA value is around 2, above which the thermal performance of the regenerator remains virtually unchanged. For the purpose of this study the value of CA is thus maintained at a level of 2 or more.

From the operational point of view, it is essential that seal-shoe/matrix wear is kept to a minimum, thus a prolonged seal life can be attained. Therefore it is desirable that the rotor speed be kept to a minimum, maintaining a CA value of 2.

Analysis of Fig. 7.33 shows that with an increase in flux density, the minimum rotor speed must be increased if an acceptable value of CA and thus high air side effectiveness is to be maintained. Because the value of CA is inversly proportional to the air port area, very large diameter rotors will need to operate at higher speeds to maintain acceptable thermal performance. In addition it can be seen that although the change in hole size has little effect, an increase in wall thickness can significantly increase the value of CA.

B. Effect of Flux Density & Rotor Design

The relationship of the three variable (λ_A , λ_G & l/d)

which have a significant effect on regenerator performance are presented in Fig. 7.34 and 7.35.

For a given value of gas flux density, a decrease in air flux density improves the air side effectiveness i.e. performance. Similarly for given flux densities (air and gas) an increase in the ratio of l/d provides an improved performance although the benefit of increasing the ratio of l/d tends to diminish at the higher values of l/d.

Fig. 7.36 demonstrates that the refractory utilisation (U) can be related to the ratio of air/gas flux densities (λ_A / λ_G) but the high refractory utilisation is generally incompatible with a high air side effectiveness.

An increase in the ratio of l/d produces an increased pressure drop across both the air and gas sides. Fig.7.37 shows typical levels of air side pressure loss. The absolute levels will vary according to the air temperature and the plots shown are consistent with an air preheat of about 1000°C at l/d = 100. Fig. 7.38 gives a similar set of data for the waste gas side with an outlet waste gas temperature of about 700°C for the same l/d ratio.

Figures 7.39 and 7.40 give the predicted maximum cold face matrix temperature (T_x) for a range of λ_A, λ_G and l/d. The value of T_x has a direct bearing on the structure of the rotor since it is representative of the temperature at which the rotor support bars will be required to operate.

The support bars are metal and so the maximum cold face matrix temperature must be chosen with respect to the environment, the choice of bar material and the bar stress. As λ_G increases so for a given value of l/d and λ_A, the matrix temperature increases.

C. <u>Effect of Port Area Ratio,</u>

It is found that over the range of conditions being considered, the maximum air side effectiveness is obtained with a port area ration (α) of about one, and that the most likely use of α in design studies would be in its effect on the cold face matrix temperature. The effect of α is therefore analysed for a given air and waste gas flow and a given wheel size. The air port area is then varied to give a range of α. This has the effect of varying the air and gas flux densities whilst maintaining a constant value of solid/air capacity rate ratio. Figure 7.41 shows the way air side effectiveness varies with α for four particular cases of λ_A, λ_G and l/d at α = 1. For any other cases, the value of η_A at α = 1 can be determined using values from Figures 7.34 and 7.35 and plotting these in the above graph at $\alpha/(\alpha+1)$ = 0.5. The shape of the nearest curve can thus be used as a guide to assess the effect of changing the port area ratio. Figure 7.42 gives the corresponding plots of maximum cold face matrix temperature (T_X), and it can be seen that as α increases the value of T_X drops quite rapidly.

The inflections on some of the curves are caused by the flows through the matrix entering a transitional flow regime between turbulent and laminar.

The air and waste gas pressure drops can be determined from Figures 7.37 and 7.38 using the revised flux densities consistent with the new port areas and the flow rate.

D. Effect of Matrix Hole Size (d)

The choice of hole size (d = 10 mm) for this study is based on manufacturer's information of their ability to produce consistently a satisfactory matrix block. Larger hole sizes may be required or found to be necessary for certain applications, especially where there is a risk of blockage due to material being deposited from the waste gas onto the matrix.

Figure 7.43 gives the ratio of air side effectiveness with a given hole size to the air side effectiveness with d = 0.01 m, plotted against hole size for range of λ_A and λ_G which are considered to be typical of the values likely to be encountered. Figure 7.44 give a similar plot with the value of l/d increased from 50 to 80.

Increasing hole size reduced the air side effectiveness over the ranges considered and at the same time generally increases the cold face matrix temperature (Figure 7.45). The variation of air and waste gas pressure drops is given in Figures 7.46 and 7.47 for the range of flows and geometries being considered.

E. Effect of Matrix Wall Thickness (t)

The wall thickness in practice is likely to be influenced by the hole size, the size of the rotor and by manufacturers' abilities to produce matrix blocks in the required material. Figure 7.48 describes the behaviour of the air side effectiveness as hole size increases for a wall thickness of 5mm and l/d = 50. The results are compared with the value of air side effectiveness with a 10mm hole and 3 mm wall thickness for a range of typical air and waste gas fluxes.

The plots show that any increase in hole size or wall thickness tends to cause a loss in air side effectivensss for low values of λ_A and λ_G. However, at higher flux levels the effect on η is less predictable, as can be seen for the plots with λ_A = 2.0, due to the variation in the nature of the flow through the matrix holes.

Figure 7.49 shows a typical range of cold face matrix temperature variation against wall thickness for two different hole sizes. It can be seen that at low flux densities the matrix temperature does not vary greatly with increasing matrix wall thickness, but at higher fluxes the change in matrix temperature is less predictable.

The effect on the pressure drop of increasing wall thickness for a given hole size is to effectively reduce the available flow area per unit rotor area. Increasing hole size alleviates the problem, so the net effect on pressure drop will depend on the resulting blockage per unit rotor area and the corresponding air to waste gas temperature. Figures 7.50 and 7.51 give specific examples of the behaviour of air and waste gas pressure drops with change in wall thickness.

F. Effect of Waste Gas Inlet Temperature (Tgl)

As the waste gas inlet temperature drops so the air side effectiveness is reduced. Figure 7.52 demonstrates the effect of waste gas temperature on η_A for a range of flux densities and Figure 7.53 presents the data as a difference in air side effectiveness from that at 1200°C. To a first approximation the maximum cold face matrix temperature can be ratioed according to the waste gas temperatures.

G. Effect of Leakage (L)

(i) The hot face air to gas leakage through the central dividing seal has the effect of partially diluting the incoming waste gas with hot air. This causes the effective gas flux density to increase and the ingoing waste gas temperature to drop. The net effect on air side effectiveness is consequently dependent on the flux density levels and air preheat levels. Figure 7.54 shows the effect of leakage on air side effectiveness for a range of flux densities and it can be seen that at low flux densities there is an increase in η_A with air leakage, but for higher flux densities the value of η_A decreases.

The effect of this leakage on the cold face matrix temperature is shown on Figure 7.55 and it can be seen that at low flux densities there is a rise in matrix temperature, but as flux densities increase the effect is minimal.

(ii) The cold face air to gas leakage through the central dividing seal dilutes the outgoing waste gas flow and reduces the ingoing air flux density. The effect of this leakage on performance can therefore be analysed at the reduced level of air flux density.

(iii) High pressure air leaking direct to atmosphere reduces the ingoing air flux density and this can be treated as (ii).

(iv) Low pressure air leaking direct to atmosphere has no effect on the regenerator performance but reduces the amount of hot air that is available.

(v) High pressure waste gas leaking direct to atmosphere reduces the ingoing gas flux density and for the purpose of analysis can be treated as a regenerator operating at a reduced gas flux density.

(vi) Low pressure waste gas leaking direct to atmosphere has no effect on regenerator performance.

A number of leakages together will tend to complicate the performance estimates and more particularly will make analysis of test data more difficult.

H. Effect of Matrix Block Material

The study is carried out assuming sillimanite material properties, and it is found that changing to a different material such as alumina based silicon carbide will virtually make no difference to the thermal performance of the regenerator. The choice of matrix block material can consequently be governed by its wear resistance and its ability to withstand thermal shock.

J. Other Effects

Air pressure, waste gas pressure and waste gas composition all have minor effects on regenerator performance and, for the purpose of general design studies, can be ignored over the ranges that are likely to be encountered in steelworks furnaces.

7.4.5 Concluding Remarks

A rotary regenrator has a wide range of variables which can influence the final level of performance. It is not feasible to consider all the combinations of designs, so this report attempts to cover as many likely combinations as possible in sufficient depth to allow a sensible estimate to be made for any particular design.

The data presented should only be used as a guide to give a general indication of a particular performance area or requirement. When the field of possibilities has been narrowed down from this guide, then a specific design study should be carried out, using the appropriate computer prpogram.

Using the above criteria a pilot plant CRR design was carried out giving a rotor diameter of 0.56 m with a depth of 0.237 m. Assuming a zero leakage and rotor speed of 2 revs per min. waste gas and support bar temperatures for a range of air/gas flux densities are predicted. These figures are presented in Fig. 7.56

7.5 PILOT PLANT:

7.5.1 Pilot Plant Assembly

In order to assess the theoretical design, engineering design and construction of a ceramic rotary regenerator was undertaken after in depth discussions with the manufacturers.

A schematic layout of the system together with a general pictorial view are presented in Figs. 7.57 and 7.58 respectively. The major items of the pilot plant are now discussed below:

A. Ceramic Matrix : The matrix is a circular disc of 0.56 m in diameter and 0.237m in depth, made up of twelve blocks separated by ceramic paper and supported horizontally in a matrix basket. (Fig. 7.59) The gases pass through the hexagonal holes of 6.25 mm. hydraulic diameter separated from each other by a wall thickness of 3.15 mm. A pictorial view of the matrix supported in the basket is shown in Fig. 7.60.

The blocks are made of ceramic material, namely 'Morgan' Triangle 45E sillimanite which can withstand the waste gas temperatures of upto 1400^{O}C. The properties of the material are presented in Table 7.13. The material is homogeneous mix of a fine grain size. The blocks are made using extrusion technique and fired to 1280^{O}C before being assembled and machined to provide a designed matrix.

In order to support the overhang of the seal shoes of the eralier design and to provide a prolonged life of the matrix ring segments made of higher alumina material are placed on the periphery of the matrix (Fig. 7.61). Use of higher alumina allows a preferential wear of the seal shoe.

B. <u>Matrix Basket Assembly</u> : A diagramatic view of the basket assembly is presented in Fig. 7.62.

The basket comprises a ring which encircles the matrix and the three stay bars which support the matrix. The bars are made of 310 stainless steel, 25.4 mm square in cross-section passing horizontally through the matrix, 187 mm from its upper surface. These bars in turn pass through the walls of the basket-ring in such a way so as to spread the load evenly. The ring is 12 mm thick and 140 mm in depth and is also manufactured from 310 stainless steel which was chosen for its physical strength even at high temperatures. At its top edge the ring has a rim which rests on the drive ring and is secured to it by small clamps.

C. Drive Ring and Motor : Mounted on the circumference of the drive ring is an extended gear wheel which mates with another smaller gear wheel fitted to the axle of a 1 h.p. DC electric motor. (Fig. 7.63). Control of the voltage across the motor provides a variable speed device, enabling rotation of the matrix at speeds between 0 and 6 r.p.m.

A torque limiter is installed to prevent excessive power being taken by the motor should anything obstruct the smooth movement of the matrix under the seals.

D. Seal Arrangement : To prevent mixing of the two gas streams, and loss to atmosphere, it is necessary to provide a sealing system between the rotating matrix and the static duct work. This is achieved by a flexible seal arrangement which allows movement of the seals over any irregularities in the matrix faces.

The seals are essentially for two purposes, the radial seals which prevent mixing of the two gases, and the peripheral seals which prevent leakage of the gas streams to atmosphere.

The seals are designed to be in rubbing contact with the matrix, the top set of 24 seals by virtue of their own weight through spring loading is provided, and the bottom 24 being spring assisted.

The 18 peripheral seals are in groups of 3, each group forming one side of a hexagon. The 6 diameteral seals then split the hexagon into two trapeziums of equal area Fig. 7.64.

The seals themselves consist of two parts, the ceramic shoe and their metallic shoe holders. The shoes are made of a fine grain sillimanite material similar to that of the matrix, but slightly softer (i.e. fired at lower temperature) so that through the rubbing action the seals will wear preferentially (Fig.7.65). The shoes are fixed, using Fortafix Chromic cement, to the shoe holders which are made of 310 stainless steel. The shoe holders are U-shaped in cross-section so that they may run on guide rails.

The U-channel through the holders is cooled using compressed air to prevent damage to the metal, and for the same reason the guide rails are water-cooled.

Push-rods pass through the guide rails and into the shoe holders to provide a pivot for flexible motion, and with spring assistance enable pressure to be applied to the bottom seals.

A secondary sand sealing is provided to avoid peripheral leakage both for upper and lower seal assemblies. (Fig.7.66).

E. <u>Expansion Bellows</u> : The bellows are provided to allow for expansion and contraction of the ductwork under varying temperature conditions, and to facilitate retraction of the rotor assembly.

It consist of two overlapping co-axial tubes, surrounded by Kaowool fibre, all housed inside a stainless steel bellows unit.

The central tubes are made of Nimonic 75 at the waste gas inlet, EN 58B stainless steel at the waste gas outlet and preheated air, and mild steel at the main air inlet.

To make sure that the bellows are not exposed to excessive temperatures, its surface temperature is monitored both through visual (i.e. temperature point applications on the bellows surface) and recording (i.e. thermocouple recording) system.

F. <u>Mixing Sections</u>: Before measurements are made of the conditions in the gas streams it is desirable for the duct environment to be homogeneous.

Perfect mixing is achieved using two baffles which first force the gases to the duct wall and then to the centre of the duct, thereby creating turbulence.

The baffles are made of a high alumina ceramic to withstand the high temperatures present.

G. <u>Burner</u> : The heat source for the CRR is a 0.5 MW Tornado Gas Burner. This provides hot waste gases at temperatures between 150°C and 1400°C and at flowrates of 0.02 kg/s to 0.15 kg/s.

Natural gas passes through a filter and gas booster before reaching the burner, where primary and secondary air is used to control flame shape. All flows are measured using turbine and paddle-wheel flowmeters. A schematic arrangement of the combustion system is shown in Fig. 7.67.

Burner ignition is directed by a Honeywell Controller via spark plug, infra-red detector and solenoid valves on the natural gas line. Automatic shutdown is also achieved should any of the safety trips be activated e.g. high rotor torque, flame failure, high waste gas outlet temperature etc.

H. <u>Measuring Section</u> : After passing through the mixing section the gases enter a section where measurements are taken of temperature, pressure and gas composition.

Temperature measurement is by means of Land SU 6 Suction Pyrometers, connected to a multipoint recorder.

Pressure is monitored on inclined manometers, using alcohol as the fluid, at a central panel.

Gas composition is found by drawing samples through water-cooled jackets and then to a Quadropole Mass Spectrometer, which provides an analysis of CO, CO_2 N_2, O_2, H_2 and Ar.

J. Instrumentation : Rotameters are used to measure flowrates of cooling water and air to the seals, and for cooling water to the rotor bearings.

Alcohol manometers are used to measure pressure in the ducts, and also inlet air pressure to the seals. Mercury manometers are used for Natural Gas, and Combustion Air pressures.

Turbine flowmeters are used to measure flowrates of Main Air, Combustion Air, and Natural Gas whilst a paddle-wheel type is used for the Primary Air.

Chromel-Alumel probe thermocouples monitor the cooling water outlet temperatures. Chromel-Alumel patch thermocouples monitor external skin temperatures of the ductwork. Chromel-Alumel thermocouples are used in the Waste Gas Outlet and Preheat Air suction pyrometers.

A Platinum - 13% Platinum/Rhodium thermocouple is used in the Waste Gas Inlet suction pyrometer, and the same metals are used in a patch thermocouple which monitors the internal skin temperature of the Waste Gas Inlet bellows.

All thermocouple readings are measured via Multipoint Recorders.

A Linear Voltage Displacement Transducer is used to detect movement of any individual seal push-rod.

Torque taken by the rotor motor is monitored by an ammeter on the field current.

The Control Panel houses the Honeywell Controller, Safety trips and Alarm trips, which ensure that temperatures in the CRR do not get too high, and to detect any supply failure to the CRR. (Fig.7.68)

7.5.2 Operating Procedure:

To make experimental procedure simple to follow, all lines and valves are marked. Assuming all valves and services are closed at the start up, starting/operating sequence to be followed is given below:

- Turn on power/electricity
- Turn on all instruments and check that they are all working.
- Turn on cooling water supply and check flows.
- Turn on rotor drive motor and check rotation.
- Turn on main air fans; open valves to set the required air flows and downstrean pressure levels.
- Turn on seal air cooling supply and set flow (during hot testing only)
- Turn on combustion air and duct cooling fan and djust required flows and pressure levels.
- Turn on suction pyrometer suction pump.
- Check instrument panel for any alarms
- Operate flame out ductor (F.O.D.) by-pass switch.
- Turn on igniter
- Turn on fuel gas booster fan and open valves to give the required gas flow rate.

- After 10 seconds release F.O.D. by-pass switch and turn off igniter (if combustion has failed to occur wait 2 minutes for lines to purge before trying again. If burner fails to light after 3 consecutive attempts follow fault finding procedure).
- Check all alarms
- Check that all pyrometers and thermocouples are responding.
- adjust fuel flowrate to provide required waste gas inlet temperature
- Check waste gas duct and bellow temperatures and adjust if necessary.
- Check downstream pressure of regenerator and adjust if necessary.
- Check hot air duct temperature and adjust if necessary.
- Check all alarms and trim if necessary.
- Allow temperatures and flows to stabilise and continue with test schedule.

<u>Rig Shut down procedure</u>:

- Turn off fuel gas booster fan and close fuel valves leaving all other systems running for 20 minutes
- Turn off combustion and main air fans.
- Turn off suction pyrometer suction pump.
- (Turn off seal air cooling supply).
- Turn off water supply and shut all other valves as necessary.
- Check rotor still operating satisfactorily and then turn off rotor drive.

- Check all instruments before turning off.
- Turn off power supply.

A. <u>Possible Areas of Hazards and Safety Precautions:</u>

Safety of operation has been an important consideration in the design and construction of the rig and all envisaged eventualities have been catered for in the provision of a system of automatic alarms and trips.

1. <u>Explosion:</u> The most serious foreseeable hazard to personnel is the danger of gas build up and subsequent explosion. Following measures have been taken to prevent this:

(i) An ultra-violet flame detector will automatically shut off the gas supply, by activation of two solenoid valves situated close to the burner, in the event of the flame failing. An alarm will be activated.

(ii) The gas supply may be manually cut off in emergency by valves, or by push button operation of the solenoid valves. Alarm is activated.

(iii) An air flow sensor will cause solenoid valve operation and gas cut off if the burner air, pneumatic air, or cooling air fails. Alarm is activated.

The gas supply will also be automatically shut off to prevent damage to the rig in the event of:

(i) Excess temperature in hot air chimney

(ii) Excess temperature in waste gas chimney

(iii) Excess temperature in waste gas duct below matrix

(iv) Rotor stoppage.

Alarms will be activated in the event of backflow of hot gas toward the dilution air fan, (in which instance the valves to this fan should be closed), or of inadequate cooling water supply to the rotor unit, in which event the pressure should be increased or the gas turned off.

2. <u>Fire:</u> Fire risks are considered to be minimal as the rig is constructed wholly from non-combustibles. Excessive burner length would lead to serious overheating of matrix with attendant waste gas temperature rise leading to automatic gas shut off.

3. <u>Electricity:</u> Armoured cable is used for 440V supply to motor and fans. Rig platform is earthed and all other instruments are supplied by 240V.

4. <u>Others:</u> Other risk areas are minimised as follows:

- waste gases ducted straight to atmosphere
- cooling water leaves luke warm (minimum pressure alarm and continuous temperature monitoring provided)
- possible trip/falling hazards minimised by guard rails
- operating personnel kept off rig platform especially during hot operation
- suitable signs posted
- rig operated by authorised personnel only.

7.5.3 Leakage Testing

For successful development of the unit it is inherent that between waste gases and combustion air and between waste gas/air and atmosphere a minimum leakage occurs, while still allowing the thermal wheel to rotate freely. In operating, the air side pressure is generally much higher than that on the gas side/atmosphere and thus there is always some leakage in such systems. To calculate the thermal performance of a regenerator it is essential that the leakage be measured accurately.

The most common method of measuring leakage is through a tracer gas technique where the tracer amount is measured at the inlet stream and then recorded at the outlet. From the difference in two values a leakage rate can be predicted. Use of oxygen as the tracer is most common in steelworks. However, in our test work, especially where the waste gas temperatures are low, because of very small changes in oxygen levels i.e. high excess air in the waste gas, the accuracy of the results was unacceptable. Therefore Argon which represents

- chemical stability at temperatures upto $1400^{\circ}C$
- relatively low concentrations in air
- similar density and diffusion rates to air and combustion parameters
- easy to measure, non-hazardous, cheap and readily available

was used. The gas analaysis is carried out using a mass-spectrometer. A computer program together with method of calculation is presented in Appendix G.

From the results obtained it is concluded that Argon Tracer technique is highly accurate for cold air and satisfactory for hot gases. Fig. 7.69 and 7.70.

7.5.4 Cold Commissioning:

Before any hot trials can be started it is essential that cold commissioning of the equipment is carried out so that a base line for comparative assessment can be identified. The main part of this test work is given below:

1. Gas Burner: Earlier experience has shown that the Tornado burners can be difficult to light up. In order to establish safe lighting procedure and to avoid 'hot starts' approximate flows of primary and secondary air for both starting and for operation at a given temperature are established by initially firing the burner to atmosphere (to avoid any damage to the matrix) and later in its propoer position inside the burner chamber.

2. Rotor: The rotor is checked for satisfactory rotation in its position in the rig and a preliminary torque/clutch setting is obtained. In addition the rotor is checked for trueness of rotation. Both these readings thus provide a basis for future checking of any distortion in the bearing assembly which may be caused due to overheating.

3. Rotor Plus Seals: The seal shoes are measured for their flatness and this reading is used to calculate the shoe wear at the end of the test.

The rotor is checked for smooth running with the seals in position to ensure no fouling between seals and matrix. The pressure on the seal spring loading is increased to provide better sealing but still maintaining smooth running by monitoring the torque/clutch recording. Any gross inbalance in rotor is thus eliminated.

4. Cooling Water System: Lines are checked for airlocks, steadiness of required flowrates and leaks as a regular part of operating procedure.

5. Fan Flows: Approximate valve setting for particular flowrates are determined. The effect of altering these valve settingson pressure levels and flowrates is then assessed.

6. Instrumentation: Satisfactory operation of all instruments is checked.

7. Seal Leakage Testing: With the rig outlet valves closed, air is introduced from the main air fan to pressurise the air side. All flanges and pipes are checked for leakage and a pressure/flow relationship is obtained. This provides an indication of cold static seal leakage. The tests are repeated with varying rotor speeds and seal pressures. The whole procedure is then repeated on combusiton side. Finally a cold leakage check is made with lines at their nominal operating pressures. Any subsequent change in cold seal performance is judged against this data.

8. Cold(Dummy) Run: With all systems operational, a cold run (burner unlit) is carried out to assess the ease of setting up the system and any likely problems that may need to be encountered are identified.

7.5.5 Hot Testing:

To avoid any cracking of the matrix, it is essential to heat the matrix through gradual steps. The burner is ignited and through careful control, waste gas inlet temperature of 500°C is maintained until any moisture that may have accumulated in the matrix while idle, has been removed. The waste gas inlet temperature in steps of 50°C is then raised to the required temperature.

The corresponding waste gas outlet conditions are monitored and gas flux density calculated. For the said conditions the air flowrates (both inlet and outlet) together with temperatures and pressure levels are recorded and air flux density is calculated.

Except for the earlier part of work where the rotor speed was changed; the rotor speed is kept constant at 0.5 revolutions per minute. This is because our earlier work showed that within the design criteria, matrix speed has little effect on the thermal performance of the unit. To minimise thermal stresses, a speed of 0.5 revs. per minute was selected.

Known amounts of Argon, as tracer gas, are injected in the air and gas stream and the gas analysis, using a mass spectrometer, is carried out. With the help of BSC developed model interstream leakage is thus predicted. It should be realised that it is difficult to predict as to what proportion of this leakage occurs at the bottom and/or top seals.

The procedure is repeated over a range of waste gas inlet temperatures.

At the end of the test period both the matrix and seal-shoes are inspected for fouling and the rates of wear are measured.

It is fully recognised that this procedure of hot testing where the matrix is heated and cooled all the time exposes the matrix material to extreme conditions because generally these units once in service run at a constant load for 24 hours a day. To assimilate works conditions it was hoped to run the test work an a 24 hour basis during the final phase, but this had to be amended because of shortage of time.

7.6 RESULTS

7.6.1 Thermal Performance

The results for waste gas inlet temperatures ranging 500°C to 1320°C are obtained and tabulated in Appendix G. The effect of major variables on the thermal performance of the pilot plant regenerator is assessed below.

One of the main variables that effects the thermal performance of a regenerator is the flux density. To establish a true relationship of the flux density and thermal performance, effort was made to minimise the number of variables in a given test. As such, the effect of rotor speed was first assessed.

A. Effect of Rotor Speed:

To avoid any damage to the matrix, the rotor is heated in step changes. This takes a long time for the heating cycle to reach and settle at the required temperature especially when operating at low rotor speeds. A quick way to stabilise the waste gas temperature is to increase the rotor speed to 1 rev per minute. Once the temperature is attained, the rotor is brought back to the required speed.

Figures 7.71 and 7.72 show that the rotor speed, within design limits, has little effect on the air preheat or thermal efficiency of the system. Therefore to minimise mechanical stresses and shoe wear in the system a rotor speed of 0.5 revs per minutes was fixed for all future runs.

It is realised that at low rotor speeds the thermal stresses could be higher for a single block design, which is compensated in the existing multi-block unit design.

B. Effect of Flux Density:

To establish the maximum air flowrates that are available from the fan, a graph showing a direct relationship between the main air outlet pressure and the flowrate is drawn (Fig.7.73)

To assess the effect of flux density of rotor performance, the gas pressures were maintained at a constant value using the waste gas outlet and main air outlet dampers. By varying the gas and main air flowrates the flux densities are altered. Air flux densities are changed from 0.3 to to 1.8 Kg/m^2 sec for fixed gas flux densities of 0.35, 0.55, 0.65 and 1.3 kg/m^2 sec and temperature range from 500^oC to 1150^oC in the first phase of test programme. The test work was repeated extending the temperature range to 1320^oC in the later part of this work.

As predicted Figures 7.74 and 7.75 show that the thermal efficiency of the unit is directly proportional to air flux density and increases both with an increase in gas flux density and gas inlet temperature. Correspondingly an increase in air flux density reduces the air preheat temperature and this is presented in Fig.7.76 for a range of gas flux densities.

A similar relationship showing an increase in thermal efficiency with an increase in flux density ratio and a decrease in air preheat is concluded in Fig. 7.77 and 7.78, for a range of gas inlet temperatures. Thermal efficiencies of upto 80% for the pilot plant are achieved from the data collected.

C. Effect of Pressure Difference in the two streams:

It is quite obvious that the thermal efficiency of the unit will be markedly effected by the leakage. As expected, results show that an increase in pressure difference between the two gas streams reduces the seal effectiveness and thus increases leakage. This relationship, for a range of flux densities, is presented in Fig. 7.79 and 7.80, and its effect on thermal efficiency is presented in Fig. 7.81. The results are similar to theoretical predictions.

The relationship between the flux density ratio and leakage is not clear define (Fig.7.82).

Data also shows that the pressure drop through the matrix for both air and gas side is minimalthus eliminating any need for larger fans for future industrial application.

D. Comparative Data Analysis Summary:

To assess the accuracy of the computer model, the recorded values of waste gas temperatures and air preheats are compared against the practical values for the same (Fig. 7.83 and 7.84). It is interesting to see that the results offered a correlation coefficient of around 0.85 with a confidence limit of 95%, proving our model to be operating successfully.

In conclusion, the data for the three most important variables for a regenerator design for the whole series of test work is presented in Fig. 7.85 to 7.87.

Figure 7.85 shows that the air preheat is directly related to the waste gas inlet temperature and the results for a waste gas inlet temperature range of between 500°Cand to 1350°C show a correlation coefficient of almost 70 per cent.

Figure 7.86 again shows a direct relationship between the interstream pressure difference and leakage for the whole range of tests with a correlation coefficient of nearly 70 per cent.

Figure 7.87 represents the effect of flux density ratio on the thermal performance of the pilot plant regenerator. A straight line with a correlation coefficient of just over 70 per cent is achieved. Earlier test work was geared more to operating at a flux density ratio of 1 to let the system settle. However, the regenerator flux density ratio extending to 3.5, which well covers the operating conditions of a steelworks, was operated giving maximum thermal efficiencies of upto 80 per cent.

7.6.2 Sealing:

An air leakage of 0 to 35 per cent is recorded depending on pressure difference between the two gas streams (Fig.7.80). By increasing the spring pressure on the sealing shoe, leakage can be reduced but this tended to increase the torque on the motor and increase wear. The basic design of sealing does seem to hold as it can be seen that in spite of temperature gradient across the matrix surface at high temperautres, temperature has little effect on leakage. However, intensive cooling and heating cycles over a period of time do seem to effect the matrix assembly creating some unevenness of surface and thus increased leakage.

Seal depth measurements, across shoe face, at a number of points for both the top and bottom seal shoes are recorded. By recording these measurements at the start and at regular intervals during the test work, shoe wear has been calculated and presented in Appendix G. The results thus obtained are graphically summarised in Fig. 7.88.

It can be seen that wear rates are less on the top peripheral shoes than for the bottom ones. This may be due to a higher temperature gradient at the top matrix surface making the seal assembly at times act more as a proximity seal rather than the designed rubbing seal. The results also show that the wear across the radial shoes for both the top and bottom assemblies is low. Based on proximity seal arrangement such high leakage rates recorded can be explained. However, for commercial units, with the increased size of matrix the percentage leakage should decrease.

The results show that the rate of wear recedes with time span. This is beacuse after a period of operation the shoe surface becomes polished giving a minimum of coefficient of friction and thus reduced wear.

The sealing system is designed to give a preferential wear on the seal shoes so that at the annual inspection of the unit all seal shoes can be replaced. The actual life of the seal shoe is very much dependent on the rate of wear and its original thickness. Based on the wear rates measured it is concluded that the existing designed shoe will provide a shoe life of upto two years. This is consistent with our theoretically predicted shoe life (Appendix D).

In addition, to improve matrix life, high alumina segment rings have been instituted on the matrix surface.

7.7 DISCUSSION

7.7.1 Material Selection:

Results have clearly demonstrated that silliminite fired at around 1280°C can provide effective material for the production of both matrix and seal shoes, which will withstand gas temperatures of upto 1300°C. An extrusion technique for manufacturing matrix still offers the best route as it uses both the existing manufacturing know-how and a fine-mesh product.

The many problems experienced during trials were related more to the manufacturers not being able to supply the goods of the designed quality rather than to the materials itself. In our earlier search for the manufacturer for the matrix, only one (Morgan Refractories) had responded positively who stated that they had experience of manufacturing blocks to the tolerances required. Even then, a compromise had to be accepted to reduce the alumina content of the silliminite to 45 per cent rather than the 60-65 per cent originally designed. During the trials, a matrix unit had to be scrapped at delivery beacuse of its poor construction standards. There were long delays between the original dates of delivery and the actual delivery dates. Our experience clearly demonstrates that for the final production unit strict control of manufacturing standards will need to be adhered to if a successful matrix is to be produced. To reduce costs, the ceramic industry still needs experience to be able to manufacture deeper blocks.

Earlier seal shoes were made from a higher alumina silliminite using compression techniques. These shoes were considered unacceptable beacuse

i) due to the coarser material used, any chipping created rough surface on the shoe and thus higher friction and wear.

ii) with a higher alumina content in the shoe, the wear damage was more prevelant on the matrix rather than the shoe.

Following this earlier expereince, the shoes were always manufactured using 45E silliminite and extruded. The shoes are fired to 1180°C as opposed to 1280°C for the matrix to achieve a preferential wear of the shoe. As the results show, a designed shoe life of upto two years can be achieved although the intention on a commercial unit will always be to change them at the time of annual inspection.

An extensive search to establish the best ceramic cement for the job, which will withstand temperatures of upto 1400°C without breaking, is still needed. In our case, Moncon Super 80' operated quite successfully.

7.7.2 Matrix Design:

In our rotary regenerator, to accommodate the thermal stresses imposed by the constant heating and cooling cycle of the system the matrix was designed as a multi-block unit. The results conclude that the system operated successfully.

The major problems in the design were experienced in the binding of blocks. When drying, the silliminite blocks tend to warp. To mate the surfaces large amounts of machining is required. To provide a smooth surface finish both top and bottom faces of the matrix are also machined. In large industrial size units these costs can be prohibitively high.

To allow a free movement of blocks, a ceramic paper is instituted between the blocks to cover any leakage. Earlier concept of V shape cut for the support bar was rectified with a rectangular shape to stop the blocks slipping. The gap between the basket and matrix surface was filled with ceramic wool to hold the blocks tightly yet allowing it to expand at higher temperatures.

The earlier problems of matrix wear experienced were eliminated by using high alumina wear rings on both top and bottom matrix periphery. The rings thus installed also removed the over-hang of the seal shoes of the hexagonal seal assembly thus minimising damage to both the shoes and the matrix. An acceptable compatability in thermal expansion for both silliminite matrix material and high alumina segment ring was confirmed in the laboratory before the rings were mounted.

7.7.3 Seal Design

After an extensive review the rubbing seal concept was accepted to be the best. Even so effective sealing of the unit did not materialise. High radial and peripheral leakages were experienced during the test work.

A: Poor Engineering Design:

In the design, the concept of shoe ride over any obstructions is propagated by a single long thin mild steel spindle of 0.5 mm diameter attached to the middle of the seal-shoe to provide movement. Experience showed that any obstruction tended to put a high torque on the long length of spindle, thus bending it. This in turn restricted the return movement of the shoe to its original place and thus gave increased leakages.

To improve the system a long seal concept was developed by instituting a guide plate through the set of seal-shoe assemblies to make it act as a single shoe. This provides even pressure on the shoe which should help it return to its original position. Unfortunately it was found that the use of a thin strip tended to warp with heat, producing uneven surface contact and the thicker strip tended to restrict the free movement of shoe assembly making it a non-viable proposition. In addition, long ceramic shoes tended to break at high temperatures due to high thermal stresses.

It was also seen that after a period of operating the mild steel guide rails tended to collect a powdery-oxide which restricted the free vertical movement of the shoe-assembly and hence gave increased leakages.

Therefore it can be concluded that although the concept of seal design is good, engineering design needs to be improved to provide the free movement of shoe-assembly. Because of the design and time constraints it was not feasible to modify the system on the pilot plant but should be considered when designing the full production unit.

B: Hexagonal Seal Design

Hexagonal design provides an improved free area and hence improved thermal performance when compared with a circular seal assembly. However it was noted that the design tended to put maximum strain on the end shoes of the hexagonal assembly which tended either to chip or left a gap at the edges through uneven mating surfaces of the shoe thus creating high leakages.

The replacement of a single damaged shoe from the assembly was also not easy.

The system thus needs improving even at a cost of thermal penalty .

C: Secondary Sealing:

Secondary sand sealing was provided to minimise the peripheral leakage. This,through ineffective operation tended to blow sand into surrounding parts e.g. water trough and bearings. Replacement of sand with heavier B.F. slag still gave poor results . In the final trials the secondary seals were filled with ceramic-fibre which provided an improved, though not adequate sealing.

The secondary sealing system thus needs further investigation.

7.7.4 Engineering Design

The basic design of the regenerator operated successfully. However time consuming problems were experienced in the following areas:

A. Burner: The burner lighting procedures, at the earliest stages proved troublesome. It was also established that due to inadequate mixing and fuel controls a maximum gas temperature of only 1350^{o}C can be achieved in the rig. This in itself posed no problems as the temperature range to be tested was to be a maximum of 1300^{o}C and any higher temperatures would damage the bellows in the hot gas duct.

B. Cooling: Inadequate and inbalanced water cooling systems, at times stopped the smooth running of the rig. In addition, problems associated with high temperature environment could have been avoided by minimising the use of plastic tubing.

C. Instrumentation: Because of various operating delays the instrument recalibrations at times got neglected. This at times posed serious consequences e.g. the matrix was subjected to temperatures in excess of 1300^{o}C when the set conditions were to be 1100^{o}C, due to an incorrect pyrometer reading.

Those general problems related to the pilot plant have now been rectified.

7.7.5 Computer Model

From the comparison of predicted and pilot plant results it can be concluded that the computer model can be used with confidence for the future design of regenerators.

7.8 MARKET SURVEY

7.8.1 Introduction

An initial investigation of the potential market for the CRR has highlighted four areas where the Ceramic Rotary Regenerator (CRR) could be usefully employed:[251]

1. Steel Industry - public sector (BSC)
2. Steel Industry - private sector
3. Non-ferrous Metals Industry
4. Glass Industry

Figure 7.89 shows comparative thermal output ranges available in the three different heat recovery devices. The major points to be considered when interpreting the graph are:

i) the metallic rotary regenerator has a low thermal capacity when compared with the ceramic rotary regenerator.

ii) the thermal output limit of the static recuperator is smaller than the CRR and can only be used as a clean gas system.

iii) the self cleaning tendency of the CRR will give a wider market potential.

Bolland[252] in his market survey reported that if all potential users were to install CRR heat recovery systems the extra savings in UK would amount to 30 x 10^6 GJ/year Projecting this over ten years, the savings would be 300 x 10^6 GJ. Assuming a CRR market penetration of 25% over this ten years period the extra savings would be 75 x 10^6 GJ, equivalent to 7.6 x 10^6 GJ per year.

In order to penetrate these markets the CRR must be competetive with established waste heat recovery systems such as metallic recuperators, ceramic recuperators, recuperative burners and regenerative burners. Therefore, an estimate of the capital and operating costs for a typical installation has been included in this survey.

7.8.2 Steel Industry - public sector (B.S.C.)

The high temperatures available in the waste gases from soaking pits and reheating furnaces can be used to recover large quantities of heat using regenerative and recuperative heat recovery devices. The CRR is one such device which has a number of advantages which could influence the market:

(a) It can recover the potential energy in the high temperature waste gases without the need for dilution.

(b) it is highly efficient, and

(c) it is more compact than the ompetition for the same duty.

The potential savings have been demonstrated in 7.2.3.

When a potential site within B.S.C. came available, it was decided to cost a scheme for the installation of the CRR and compare this with its competitors.

The project involved the recovery of heat from the medium temperature (ca. 900°C) waste gases of a billet reheating furnace. The recovered heat would preheat the combustion air to 450°C maximum, a temperature limited by the proposed use of existing ductwork and burners. It was decided that for the CRR proposal, the ceramic matrix depth was not set at 800 mm as originally envisaged but 400 mm which was the limit of present design taking account of the ceramic development. A formal proposal was submitted to a BSC works who evaluated the scheme together with two alternative scheme .

At the inception of the development stage it was recognised both by the B S C and BW.D, that to obtain a market penetration it will be necessary to offer a sufficiently attractive economic package (i.e. subsidised) to each individual purchaser until the system is established. This will be on top of the demonstration and/or regional grants that may be available to the customer.

The capital costs of the waste heat recovery schemes considered showed the CRR (BW.D Budget)cost of £179,000 (£400,000 less 49% EEC Development grant and £25,000 BW-D contribution), compared to conventional convection-type recuperator capital investment of £130,000.

The main difference in the two schemes is the position of the recuperative unit in relation to the furnace. Otherwise to meet same air preheats the duct and service costs were expected to be similar. Therefore it was difficult to explain the BW-D budget costs of £400,000 against the comparative conventional recuperator installation costs of £130,000.

The payback periods based on above capital costs for the schemes are:

CRR (with EEC grant)	-	4 years
(without EEC grant)	-	9 years
Conventional recuperator	-	3 years

As a consequence of this study, it is clear that there is no market for the CRR in medium temperature applications where a metallic recuperator could be used unless substantial cost reductions can be obtained in the capital cost of the project. With the present depressed state of the industry, where investments are low, it is difficult to justify substantial discounting to penetrate the market.

With regard to high temperature applications of the CRR at soaking pits and high productivity reheating furnaces, the generation of air preheat temperatures of over 550°C will involve complete replacement of the ducts and burners. This is likely to be costly and the fuel savings achieved are unlikely to meet the B S C payback criteria at the present BW-D budget costs.

The only market for the CRR may be for new furnaces requiring high air preheat temperatures. There are very few new projects proceeding and there will be great reluctance to install an unproven waste heat recovery system on a new furnace.

7.8.3 Steel Industry - private sector

The market outlook for the public sector of the steel industry is reflected in the private sector. Heat recovery schemes producing medium preheat air temperatures are widely adopted on the larger furnaces in this industry and although the private sector is quite willing to accept new technology, it will be difficult to make the case for installing the CRR over conventional devices.

On smaller furnaces within the private sector, waste heat recovery is not as widespread and it may be possible to justify the installation of a small CRR on payback terms. But if the element of competition is introduced at the present budget costs, conventional recuperators will be difficult to match.

7.8.4 Non-ferrous Metals Industry

The waste gases from this industry are considered to be low to medium temperature and as such the market for the CRR in competition with conventional heat recovery devices, will be negligible.

7.8.5 Glass Industry

Three separate areas of application can be considered for the use of the CRR within the glass industry.

A. Pot melting furnaces are not expected to produce an appreciable market due to their mode of operation and the adequate performance of the conventional heat recovery devices.

B. Tank melting furnaces provide a very difficult and arduous application for the CRR. The waste gases from glass tanks are laden with molten particulate matter which solidify on contact with a cooler surface i.e. the regenerator. Conventional regenerators partially solve the problem by careful selection of refractories and gas passage dimensions.

This application was thought to be a second stage development of the CRR and would rely on the successful completion of the prototype using the existing refractories within the matrix. Separate development will be required to develop the matrix block production methods for the special glass regenerator refractories.

C. The forehearths of the glass furnaces develop high temperature, and relatively clean, waste gases which could be suitable for heat recovery with the CRR. The progress of market development of the CRR for use on forehearths will be severely hampered by competition from the newly-developed recuperative - and regenerative type burners which are aptly suited for this application.

7.9 CASE STUDY: APPLICATION OF CERAMIC ROTARY REGENERATOR IN STEEL INDUSTRY.

At this stage in the project is it prudent to review the economic situation with regards to the potential installation of the CRR in steelworks. For this, four sites were studied namely, Scunthorpe soaking pits, Lackenby soaking pits and Lackenby slab reheating furnaces No.2 and 3. The costing for this review has been taken from a recent report by M/S. Stone and Wilshaw.[253] and updated to meet our requirements in this study and summarised in Table 7.14.

1. Scunthorpe Soaking Pits:

Information for the calculation was taken from No.5 East soaking pit which utilises a blast furnace gas/coke oven gas mixture to heat the pit through a single burner. On exit to the furnace, the waste gases are diluted with cold air to reduce the temperature to 1000°C before passing through a two stage recuperator. The first stage preheats the combustion air to approximately 400°C and the second stage preheats the gaseous fuel to approximately 300°C.

The heating cycle of the ingots in the soaking pits is split into two periods where the maximum heat is inputted and the heat transfer is the greatest, and the soaking period, where the fuel rate is gradually reduced to allow the heat in the ingot to slowly "soak" through. This latter period has been represented by average conditions.

Cold or hot ingot charging influences the cycle time and ultimately the potential savings. Cold ingot charging is represented by 10 hours in the heating period and 4 hours in the soaking period. Hot ingot charging is represented by 4-5 hours in the heating period and 4 hours in the soaking period.

The scheme utilises the gas recuperator to further increase the fuel savings but pays the price of refurbishing the recuperator for further use.

2. Lackenby Soaking Pits

The 36 soaking pits at Lackenby supply hot ingots to No.1 and No.2 primary mills for the production of slabs and beams; 24 pits for No.2 mill and 12 pits for No.1. Information for the calculations was obtained from the heating period of a soaking pit from No.2 mill. Cold charging is represented by 12 hours in the heating period followed by 4 hours in soak. Correspondingly hot charging takes 4-5 hours in heating followed by 4 hours in soak.

The pit is oil-fired and combustion air is preheated to around 300°C by a recuperator in the waste gas flue. Again dilution air controls the recuperator entry temperature to 1000°C.

Current operating practice at Lackenby dictates a charging pattern of 30% cold/70% hot ingots with each pit having a 24 week campaign. 22 x 21 shift-weeks with 2 weeks off for slagging means an annual rate of 7400 hours/year.

Although there was no detailed investigation into an installation at Lackenby (unlike Scunthorpe) most of the costs for the complete replacement scheme at Scunthorpe are applicable to Lackenby.

Since little information is available for the soaking period except for the average fuel rate of 150 l/hr, the fuel savings in the soaking period is taken as 40% (probably an underestimate).

3. Lackenby No.2 Slab Reheating Furnace

No.2 furnace at Lackenby is a 5-zone conventional, pusher-type, slab reheating furnace with the burners in all 5 zones being supplied with oil and preheated combustion air. While the recuperators supply combustion air at 350^{o}C, the waste gas entry temperature is controlled to about 950^{o}C by the use of water sprays. The hot combustion air is controlled in the hot air header and then distributed throughout the furnace. The information matches the full fuel flow rates at 80% activity levels.

To accommodate the large volume of waste gases produced, 8 CRR's are proposed and for comparison two rotor depths 0.8m and 1.6m are used when predicting the savings accrued. The cost of installation at Lackenby No.2 mill is assumed to be the same as predicted for Ravenscraig.[253] It is recognised that although the existing air ducts and burners designed for 550^{o}C will be able to cope with air preheats generated by o.8m diameter rotor, additional costs in ducting will need to be borne for the 1.6m diamter unit. It should also be noted that since costings for the 3 meter unit with increased rotor depth are not available, the cost of the unit has been calculated as directly proportional to the rotor depth.

4. Lackenby No.3 Slab Reheating Furnace

No. 3 furnace at Lackenby is a 5-zone OZRF design, pusher-type slab reheating furnace utilising preheated combustion air in the oil burners in the 4 main zones. The novel design of this furnace allows the majority of the heat to to be transferred to the slabs over the first 30% of the furnace. With no preheating zone to allow the waste gases to give up some of their heat, the gases leave the furnace centre at 1250-1300°C. Four two-pass recuperators are installed above the furnace to recoup some of the heat lost in those gases. Again dilution air is used to protect the recuperator from excess temperature. The fuel flow rates relate to an average level of activity of 80%.

It is proposed that 10 CRR's are used to accommodate the large volume of waste gases produced. Two rotor depths, 0.8m and 1.6m are used in the calculation for comparison of the fuel savings produced. All costs have been taken from a detailed scheme from Ravenscraig.

Again increasing the rotor depth directly increases the cost of the unit.

SAVINGS

Assuming the radiation losses are negligible and fuel/air ratio together with waste gas inlet temperature for the recuperator/CRR design remain the same; using the formula, fuel savings for all the cases considered is calculated and a summary of the results is presented in Table 7.14.

$$\text{Fuel Savings (\%)}\ S = 100 \times \frac{1 - \left\{ cv + n(1+x)ha - \left[n(1+x)+1 \right] hg \right\}}{cv + n(1+x)ha - \left[n(1+x)+1 \right] hg}$$

The results show that payback periods between two to five years can be achieved.

When analysing the results, the following points must be considered:

1. It must be remembered that the costing for Lackenby reheating furnaces are based on that for Ravenscraig and the layout of Lackenby is mugh tighter than that of the Scottish Division. This could mean cost penalties.

2. The combustion air mains at Lackenby reheat furnaces are rated to take air at 550^{o}C. For preheats slightly greater than this, it may be possible to increase the duct lining to accommodate the temperature and therefore eliminate the need for a new hot air distribution system and consequently reduce the cost of installation. But it must be remembered that with this the ducting supports must be able to take the weight increase and that the velocity and consequently the pressure drop will increase in the mains.

3. No consideration has been given to the effect of increased pressure drop through the proposed waste gas system and also the effect of the reduced draught due to the increased density of the existing waste gases caused by the lower temperature.

4. Preheating the combustion air increases the flame temperature if all other conditions are constant. Experience on soaking pits has shown that difficulties occur when the flame temperature is increased namely melting of the ingot tops. This is remedied by either increasing the air/fuel ratio which decreases the fuel saving or by slagging more

often to prevent the ingot tops coming in contact with the flame. Therefore, the effect of increasing the flame temperature in soaking pits even further must be looked at more closely.

Further savings could accrue by increasing the flame temperature on reheating furnaces as the heat transfer will increase significantly due to the greater temperature differential.

5. In the research so far, 0.8 meters has been given as the depth of the suggested 3 meter diameter production unit. The above calculations have also used an increased rotor depth to show potential savings if the future refractory research continues in this vein.

Detailed costings have been made on the 0.8 m rotor depth unit and projected costings of an increased rotor depth unit have been made on the basis of the cost being proportional to the rotor depth.

An example of the cost breakdown for the installation is given below:

1. Soaking Pit (rotor depth 0.8meter)

	£
Rotor assembly	90,000
Hood refractory linings	10,000
Rotor refractory blocks	21,000
New Ducting	103,000
Electrics & Instrumentation	10,000
New Burner	50,000

Expansion joints & duct valves	34,000
Steel supports	11,000
Civil supports	8,000
Water and air to regenerator	2,000
Profit	32,000
	£376,000

2. Reheating Furnace (rotor depth 0.8 meters)

Rotor assemblies	717,000
Hood refractory linings	80,000
Rotor refractory blocks	170,000
New ducting	1,200,000
Bellows	50,000
Steel supports	400,000
Electrics & Instrumentation	40,000
Duct valves	48,000
Civil supports	74,000
New Burners	165,000
Water & air to regenerator	16,000
Profit	326,000
	£3,286,000

7.10 RESEARCH FUNDING

Market Survey (251) had clearly demonstrated that through an increased waste heat recovery from the high temperature gases leaving the furnaces, successful development of the regenerator could offer an average market potential of 7.5×10^6 GJ per year energy savings over a ten year period. The economics of the fuel usage to be gained from the unit are of particular interest to B.S.C., because of the direct relevence to the very great quantities of fuel consumed by its many soaking pits and reheating furnaces. BW-D a major UK contractor recognising the wider applications of the regenerator e.g. non-ferrous and glass industry, was keen to exploit the novel developed unit worldwide. With the technical know-how of the B.S.C. and engineering and marketing expertise of BW-D, a collaborative development of the project offered a good incentive.

The areas of risk in the pilot plant development were mainly concerned with the selection of matrix material which will be able to withstand the thermal loadings of a regenerative system whilst at the same time remaining within acceptable tolerances and distortion limits in order to avoid matrix/ seal degredation with consequent loss of unit efficiency. A further risk area for concern was the possibility of fouling and blockages, though this can mainly be overcome by selecting the required hole size and flux densities of media. The efficiency of the regenerator depends upon the adequacy of gas seals and in our rubbing seal concept clearly wear rates and attainable tolerances are major factors in seal efficiency.

For the production unit it was intended that, with the exception of ceramic material, all manufacture, assembly and installation will be carried out by the Babcock Group. Thus the materials and machining technique required will be readily available and will not pose any problems. The ceramics material may require advanced production techniques in order to obtain blocks of the size, accuracy and configuration required. These areas were to form a major part of the BW-D feasibility exercise to be carried out during the project together with the manpower assistance in the day to day running of the project.

Both the B S C and BW-D operating under considerable financial constraints felt that the work could not be progressed without the financial assistance of the government. Therefore a grant application with detailed submission was made to the Department of Energy in 1980. The government recognising the potential benefits of the development to the national energy conservation programme authorised a grant of £197,000 split in two phases i.e. (Phase 1 £140,000 and Phase 2 £57,000) in October 1981, with an over-rider that Phase 2 was only to be started at the conclusion of Phase 1.

This report concludes the work for Phase 1 as originally submitted.

7.11 CONCLUSIONS

The chapter clearly demonstrates the need for developing a ceramic rotary regenerator to recover potential heat from the high temperature waste gases avaialble in the ferrous and non-ferrous industry.

The major problem of material selection to meet the demands imposed by high temperature gases has been successfully completed in the BSC described ceramic rotary regenerator (CRR). The units basic concept of heat transfer by storage in and recovery from a porous ceramic matrix which is rotated on its vertical axis across the flow paths of hot gases and cold combustion air provides a compact, more efficient heat recuperation system. The rubbing seal concept, though proved successful for wear fails to provide acceptable leakage rates. This is not to say that the basic idea is poor but the engineering design needs to be modified. Even so air preheats of upto 750^{o}C for waste gas inlet temperatures of 1200^{o}C are achieved with a maximum unit efficency of up to 80 per cent.

The market potential of the system was fully realised by the industry and the development work has been proceeding in collaboration with Babcock Woodall-Duckham. At these times of escalating energy costs the Government fully appreciate the need for such a development work. Recognising the financial constraints of the industry, a R & D grant was awarded by the Department of Energy to develop the project.

The development of a production unit can only be achieved with the help of ceramic industry who unfortunately have little experience in this field. The scheme developed for the application of a CRR unit on a reheating furnace showed that the BW-D Budget costs submitted for the scheme (not unit) were unacceptably high when compared with the cost of a similar duty recuperator system. Therefore it can be concluded that if market penetration is to be achieved these costs will need to be pruned considerably.

TABLE 7.1 ENERGY REJECTION IN SOAKING PIT WASTE GASES

Based on steel heated in soaking pits

Works	Steel processed K tonne/week	GJ/h	MJ/tonne Steel	TJ/annum
Llanwern	41.3	100.3	408	809
Port Talbot	53.4	138.0	434	1113
Revenscraig	29.3	40.5	232	323
Scunthorpe	64.7	231	600	1863
South Teesside	60.8	157	434	1266
Consett	21.4	50	393	403
Corby	13.62	51	629	411
Shotton	24.5	79	542	637
Sheffield Division	60.3	136	307	870

TABLE 7.2 ENERGY REJECTION IN REHEAT FURNACE WASTE GASES

Based of Steel Geated in Reheating Furnaces

Works	Steel processed K tonne/week	GJ/h	MJ/tonne steel	TJ/annum
Llanwern	36.2	163	756	1314
Port Talbot	47.0	275	982	2218
Ravenscraig	22.9	41	302	332
Scunthorpe	49.0	305	1044	2459
South Teesside	64.8	234	607	1887
Consett	7.6	30	663	242
Shotton	21.6	148	1151	1193
Sheffield Division	19.7	101	700	648

TABLE 7.3 ENERGY REJECTION IN REHEATER SKID COOLING

Based on Steel Heated in Reheating Furnaces

Works	Steel produced K tonne/week	GJ/h	MJ/tonne steel	TJ/annum
Llanwern	36.2	74.3	345	599
Port Talbot	47.0	136	485	1094
Ravenscraig*	22.9	8.9	65	71
Scunthorpe	49.0	21.9	75	177
South Teesside	64.8	125	324	1008
Consett	7.6	28	442	226
Shotton	21.6	148	1151	1193
Sheffield Division	19.7	9	61	56

* After steam credit in waste heat boilers.

TABLE 7.5

CERAMIC MATERIALS REJECTED

TABLE 7.6

ALUMINIUM-SILLIMINITES & THEIR PROPERTIES

PROPERTIES OF SOME SILICON CARBIDES AND NITRIDES

Material	Density (kgm^{-3})	Strength MNm^{-2})	Thermal Expansion Coefficient	Melting Point (^{o}C)	Oxidation resistance
Refel Silicon Carbide	3100	514	4.3	2700	exceptional
Hot pressed Silicon Carbide	3300	814	4.4	"	Generally Poor
Reaction Bonded Si_3N_4	2500	200	3	1900	
Hot pressed Si_3N_4	3300	1034	2.6	"	
Sialons	similar	similar	3	"	10 times better than Si_3N_4
Sillimanite	2400	69		1800	absolute

PROPERTIES OF SOME SILICON CARBIDES AND NITRIDES

TABLE 7.7

SEAL BLOCK SET TEST CONDITIONS

Test No.	Seal Block Set	End Float mm	Actuation Air Flow (kg/s) x 10^{-5}	Test Temp ^{o}C
M1	A(1-10)	0	0	28
M2	A(1-10)	0	varied	22
K1	A(1-9, 11)	0.37	0	25
K2	A(1-9, 11)	0.37	4.1	26
K3	A(1-9, 11)	0.37	6.2	26
K4	B(1-11)	0	0	21
K5	B(1-11)	0	4.1	22
K6a	B(1-10, 16)	0.20	0	22
K7a	B(1-10, 16)	0.20	4.1	22
K8a	B(1-10, 17)	0.40	0	23
K9a	B(1-10, 17)	0.40	4.1	23
K10	A(1-9, 11)	0.37	4.1	22.5

SEAL BLOCK SET TEST CONDITIONS

TABLE 7.8

EQUIVALENT COLD SEAL PERFORMANCE ON A CEL
REGENERATOR DESIGNED FOR 3 kg/s AIR FLOW RATE

Test No.	Seal Actuation Pressure Diff. N/m^2	Seal Actuation Flow (kg/s/m)	Seal Assembly End Float (mm/m)	Leakage Loss (% of total air flow)
M1	0	-	Nil	0.25
K1	0	-	1.5	0.49
K2	385	1.63×10^{-4}	1.5	0.41 0.49*
K3	548	2.48×10^{-4}	1.5	0.34 0.46*
K4	0	-	Nil	1.43
K5	240	1.69×10^{-4}	Nil	1.34 1.42*
K6a	0	-	0.80	0.42
K7a	323	1.65×10^{-4}	0.80	0.34 0.41*
K8a	0	-	1.6	0.40
K9a	312	1.65×10^{-4}	1.6	0.31 0.39*
K10	419	1.65×10^{-4}	1.5	0.33 0.41*

* Indicates actuation flow as leakage loss

EQUIVALENT COLD SEAL PERFORMANCE IN A
REGENERATOR DESIGNED FOR 3 KG S AIR FLOW RATE

TABLE 7.9

RESULTS OF LEAKAGE AGAINST PRESSURE DROP FOR PROXIMITY SEAL TESTS

Total Air Flowrate = 2000 l/min

Seal Width mm	Seal Gap $\times 10^{-3}$ mm	Leakage Rate @ P - mmWG l/min			
		2.54	7.62	15.24	30.40
120	25	-	250	400	750
	50	200	500	760	1250
	75	375	750	1175	1750
	100	550	1050	1150	-
	125	825	1550	-	-
90	25	-	250	450	750
	50	200	450	750	1200
	75	325	700	1075	1650
	100	550	1050	1500	-
	125	800	1400	-	-
60	25	-	250	400	725
	50	325	525	850	1300
	75	425	850	1250	1950
	100	600	1150	1650	-
	125	800	1450	-	-

RESULTS OF LEAKAGE AGAINST PRESSURE DROP FOR PROXIMITY SEAL TESTS.

TABLE 7.10

RESULTS OF ROTARY REGENERATION SEAL TESTS

Test No.	Load on Shoe Kg (LBS)	Distance Travelled M	Duration of test (DAYS)	Temperature ^{o}C	Wear on shoe mm	Wear on Matrix mm	Wear Rate of shoe mm/day	Effective wear Rate of shoe on Pilot Plant mm/day
I	1.721 (3.8)	53,486	9	3hrs @ 200^{o} 22hrs @ 450^{o} 7 days 21hrs @ 700^{o}	1.5	2.0	0.167	0.0240
II	1.721 (3.8)	65,981	11	Cycling 400^{o} to 800^{o}	3.25	0.75	0.295	0.0425
III	1.721 (3.8)	71,981	12	Cycling 250^{o} to 550^{o}	3.5	1.3	0.292	0.0420
IV	3.896 (8.6)	17,735	2.96	Cycling 250^{o} to 550^{o}	2.75	1.35	0.929	0.1335
V	0.535 (1.18)	188,951	31.5	Constant 700^{o}	2.8		0.089	0.0138

RESULTS OF ROTARY REGENERATION SEAL TESTS TABLE 7.11

SLIDING ABRASION TESTS

Test No.	Moving Surface	Static Surface	Temp. ^{o}C	Moving Wear Avge. Depth (mm)	Moving Wear Volume Loss (ml)	Static Wear Avge. Depth (mm)	Static Wear Volume Loss (ml)	Length of Test	REMARKS
1	Sillmnt	Sillmnt	Room Temp	0.04	0.10	0.13	0.77	2hrs 5 mins	No grooving evident very little apparent wear.
2	Sillmnt	Sillmnt	1000	0.29	0.77	1.35	8.51	2hrs 30mins	'Dry' wear of both shoe and matrix with slight grooving of both surfaces along matrix.
3	Sillmnt	Sillmnt	1100	0.01	0.02	1.89	10.89	2hrs 30 mins	'Dry' wear producing powder. Chipping of shoe ends.
4	Sillmnt	Sillmnt	1200	+0.10	+0.23	0.16	0.86	1hr 54mins	General shoe deposition with slight grooving of matrix surface.
5	Sillmnt	Sillmnt	1300	0.08	0.19	0.34	1.84	2hrs 18mins	Surface deposits up to 1.16 mm high causing grooving of matrix surface.
6	Sillmnt	Sillmnt	1300	0.21	0.48	0.33	1.90	2hrs 18mins	Repeat of test 5
7	Sillmnt	Sillmnt	1400	+0.15	+0.30	0.10	0.52	2hrs 2mins	Both surfaces partially glazed. Shoe pick up debris from matrix.
8	Sillmnt	Mullite	1150	+0.06	+0.14	0.19	1.13	2hrs 7mins	Grooving of mullite and slight fritting type deposition on sillimanite. Mullite light brown.
9	Sillmnt	Mullite	1200	+0.18	+0.41	0.01	0.74	2hrs 14mins	Repeat of test 6
10	Sillmnt	Mullite	1300	+0.44	+1.02	0.53	2.86	2hrs 20mins	Surface shoe deposits up to 1.99 mm causing grooving of mullite surface.
11	Sillmnt	Mullite	1300	+1.04	+2.44	0.35	2.08	2hrs 7mins	Grooving of mullite caused by large fritting type deposition on silli shoe. 5.21 mm High. Millite off white (Repeat of test 10)
12	Sillmnt	Mullite	1400	+0.60	+1.24	0.35	1.83	2hours	Surface shoe deposits up to 1.32 mm causing wear of mullite.
13	E.S.B Mullite	Sillmnt	1300	0.08	0.16	0.05	0.32	1hr 46mins	Both surfaces appear glazed. Little wear
14	Sillmnt	Silli W.ring	1200	0.06	0.14	+0.13	+0.75	2hrs 14mins	Reversal of test No. 4. Deposition on stator and wear of shoe.
15	Sillmnt	Alumina	1300	0.01	0.23	0.08	0.48	2hrs 5mins	No grooving evident Needle like rolls on both surfaces o.16mm diameter.

SLIDING ABRASION TESTS

TABLE 7.12

Typical Properties of CRR Matrix/Shoe Material

Morgan 45E Sillimanite

Chemical analysis:

Alumina	45.0%
Silica	50.3%
Titania	1.4%
Ferrix Oxide	1.4%
Lime	0.4%
Magnesia	0.2%
Alkalis	1.7%

Physical Properties:

Bulk Density	2.38 g/cc
Apparent Porosity	2.0%
Cold Crushing Strength	17,000 lb/in^2
Maximum Service Temp.	approx. 1300 ^oC
Mean Co-efficient of Thermal Expansion	$6 \times 10^{-6}/^oC$
Thermal Conductivity Mean Temperature 1000 ^oF	11 $BTU/ft^2 hr ^oF/in$
Specific Heat	approx 0.25 $BTU/lb/^oF$

Chemical Properties:

Acid (H_2SO_4) Solubility Test DIN 51 102

1.0-1.5% weight loss

TYPICAL PROPERTIES OF CRR MATRIX/SHOE MATERIAL MORGAN 45E SILLIMANITE — TABLE 7.13

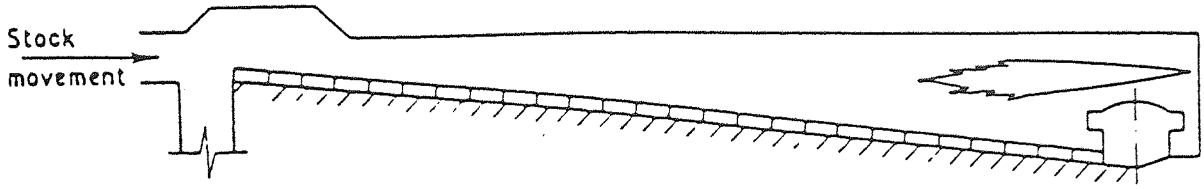
CRR APPLICATIONS IN STEELWORKS: ECONOMIC ASSESSMENT

	Lackenby					Scunthrope Soaking Pits
	Soaking Pits	No.2 Reheat Furnace		No.3 Reheat Furnace		
Fuel: Type	oil	oil		oil		Gas
Density Kg/m^3	0.96	0.96		0.96		1.26
CV MJ/Kg	40.5	40.5		40.5		3.58
Fuel flowrate /h	500	9240		11200		-
Nm^3/h	-	-		-		3820
Excess Air %	19	19		19		17.8
Existing Recuperation						
Air Preheat ^oC	300	350		400		370
Gas Preheat ^oC	-	-		-		320
No. of CRR's	1	8		10		1
Waste Gas Inlet Temp. ^oC	1250	1000		1200		1300
Rotor Depth m	1.0	0.8	1.6	0.8	1.6	1.0
Air Preheat Temp. ^oC	1000	460	630	585	820	940
Waste Gas Outlet Temp. ^oC	440	625	475	720	530	700
Fuel Savings/ tonnes/year	1140	3725	8750	8300	16500	$7.7 \times 10^6 Nm^3$
Capital Expenditure £	39400	3.1M	4.6M	3.8M	5.7M	416000
Financial Savings £/yr	18100	593000	1.36M	1.32M	2.63M	94000
Payback Period year	2.2	5.2	3.4	2.9	2.2	4.4
Fuel Savings %	40*	7	16.1	12.9	25.6	34.5**

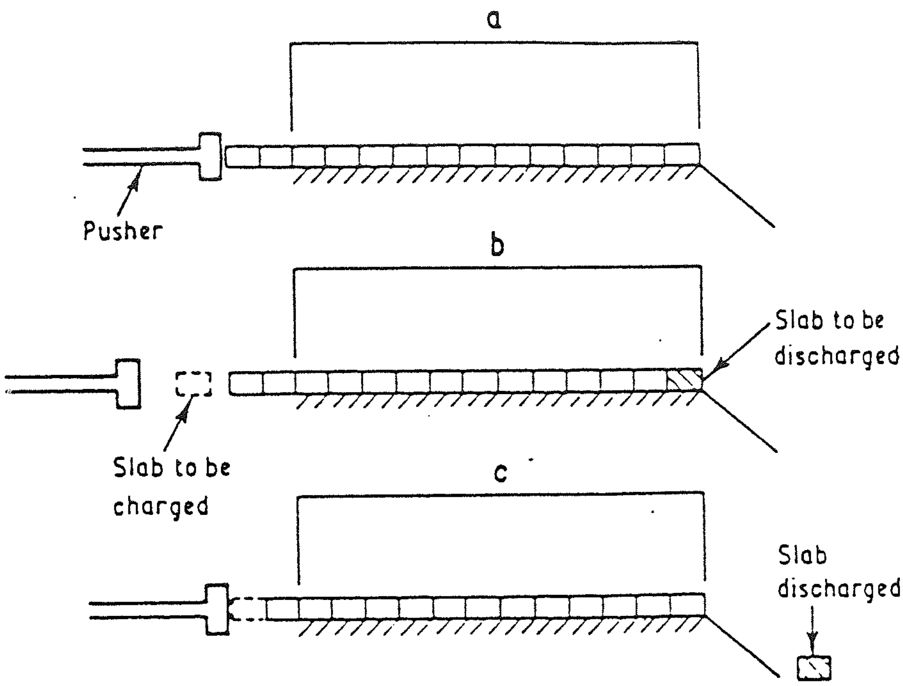
* Savings based on 30% cold/70% hot charging practice
** Savings based on 50% cold/50% hot charging practice

TABLE 7.14

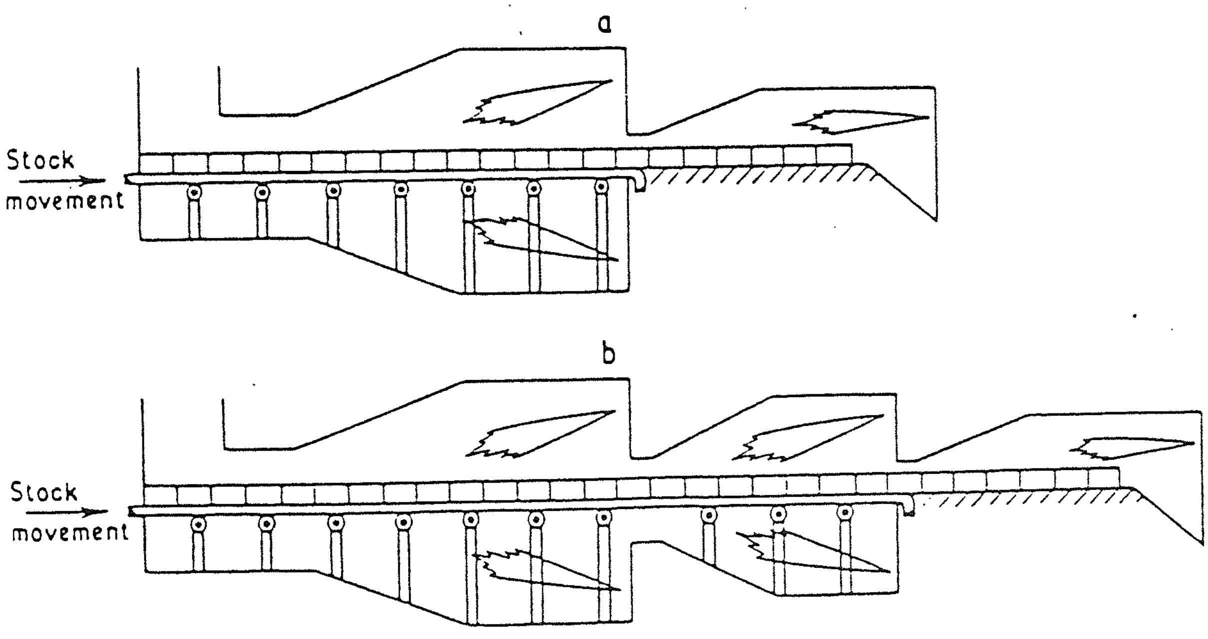
CRR APPLICATIONS IN STEELWORKS: ECONOMIC ASSESSMENT



SINGLE-ZONE REHEATING FURNACE



CHARGING AND DISCHARGING OF A PUSHER FURNACE

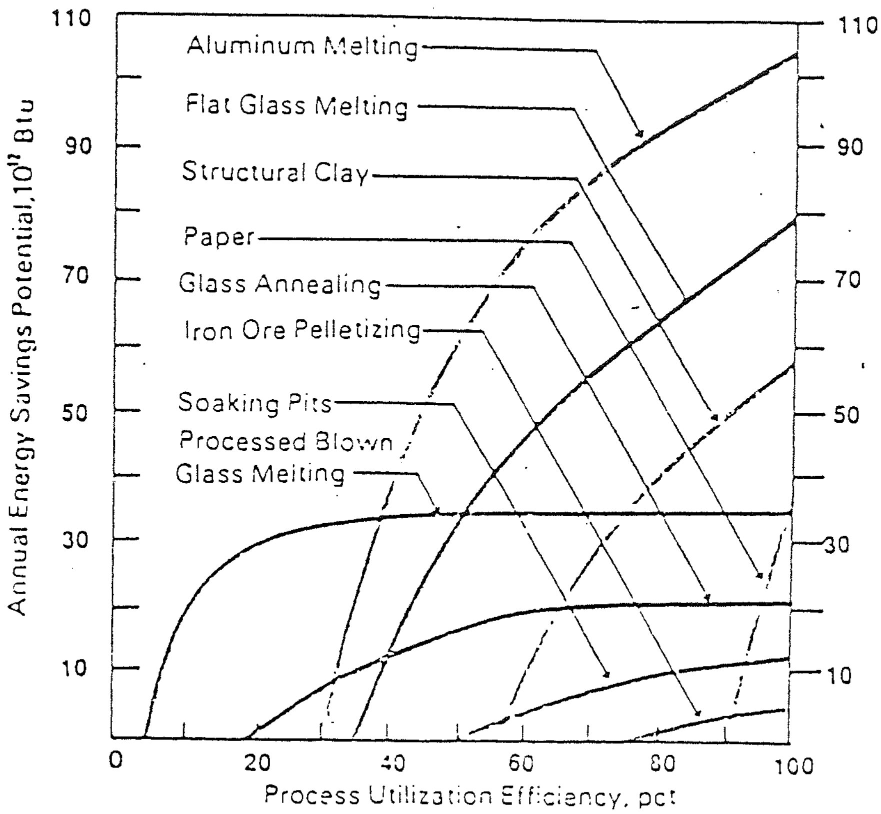


TYPICAL 3-AND 5-ZONE REHEATING FURNACES

TYPICAL REHEATING FURNACES

FIGURE 7.1

Energy Utilization Effeciency Potential



ENERGY UTILISATION EFFICIENCY POTENTIAL

FIGURE 7.2

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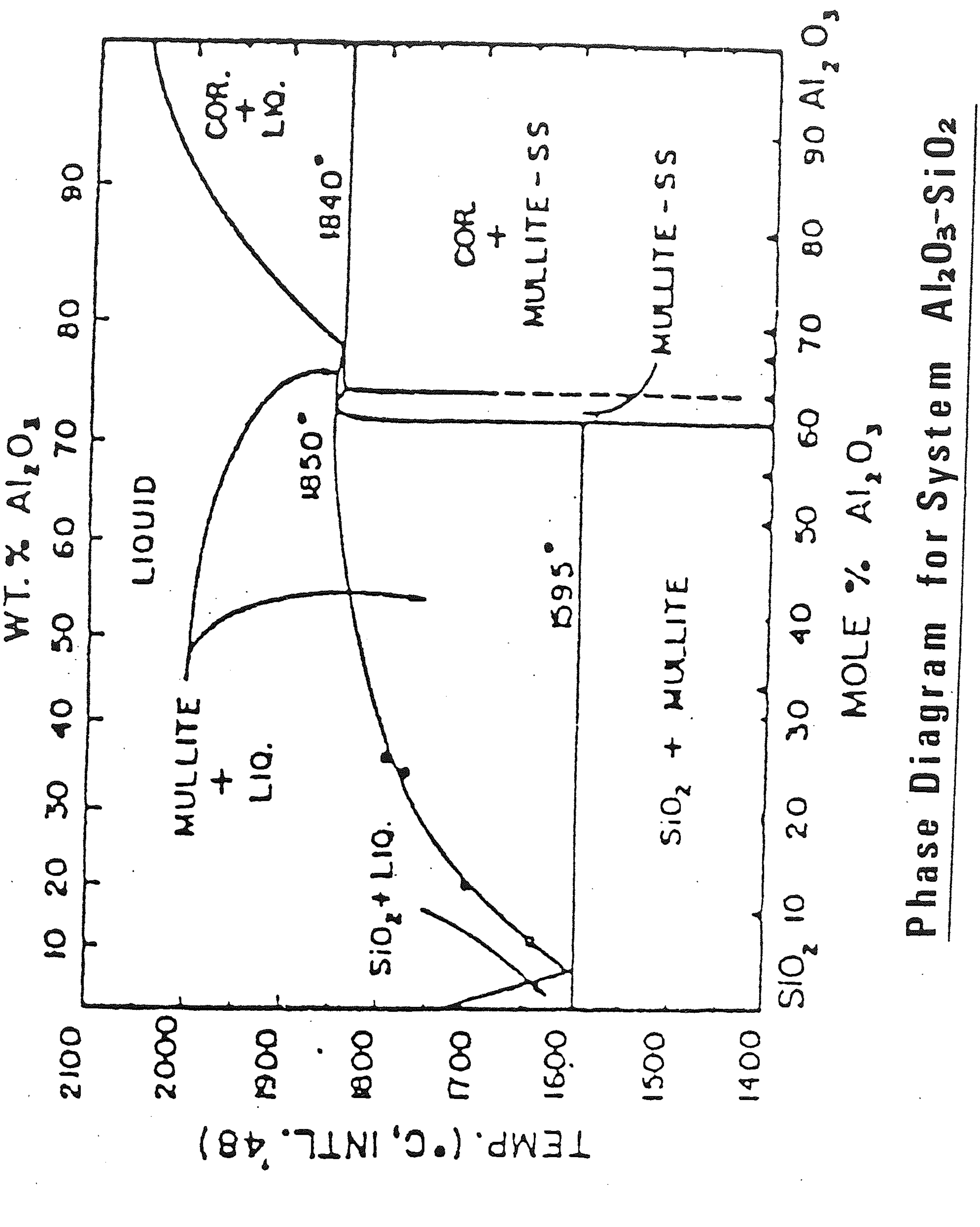
Phase Diagram for System Li_2O-Al_2O_3-SiO_2.

FIGURE 7.3

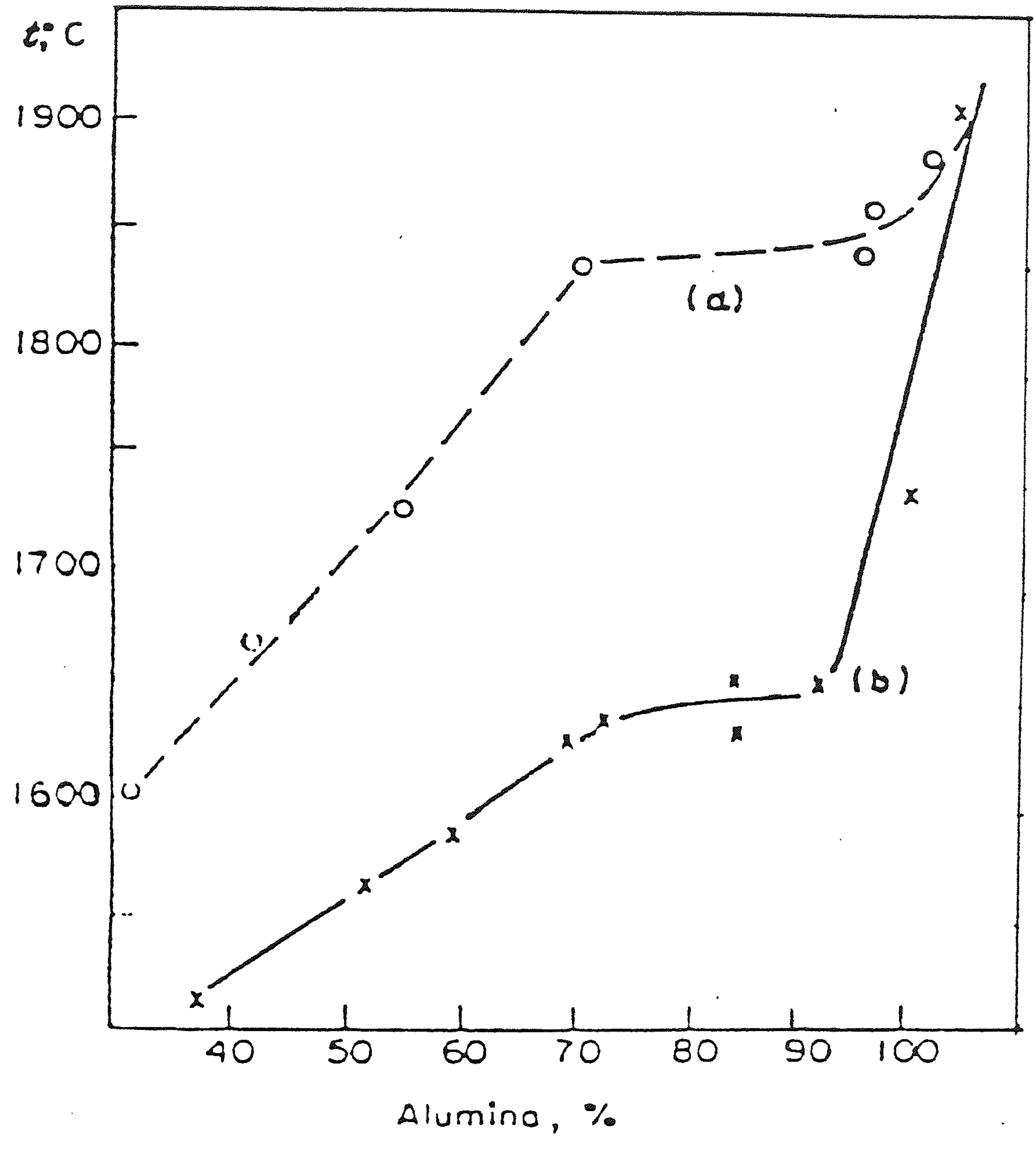
CERAMIC ROTARY REGENERATOR

GURE 7.4



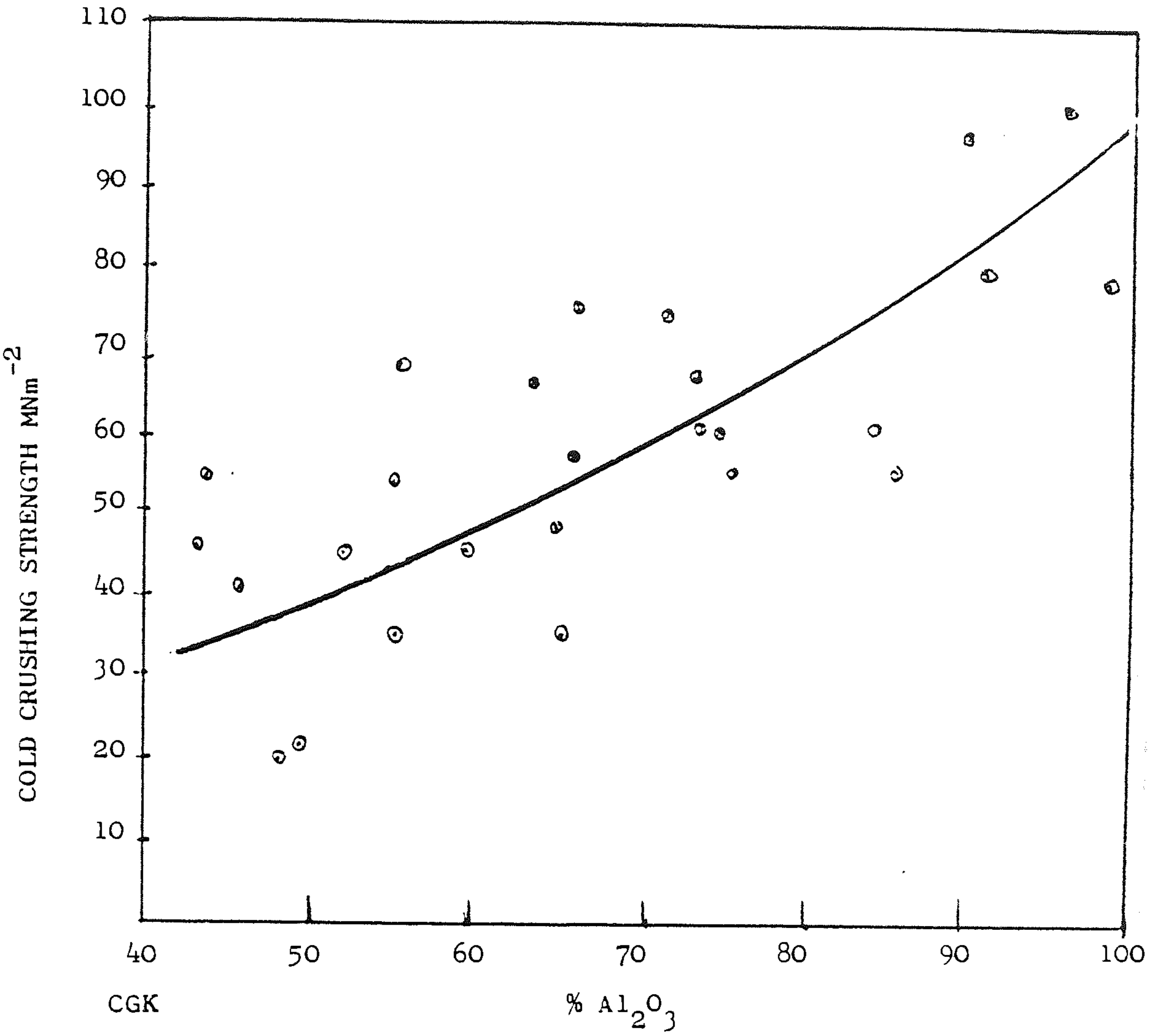
PHASE DIAGRAM FOR SYSTEM AL_2O_3-SiO_2

FIGURE 7.5



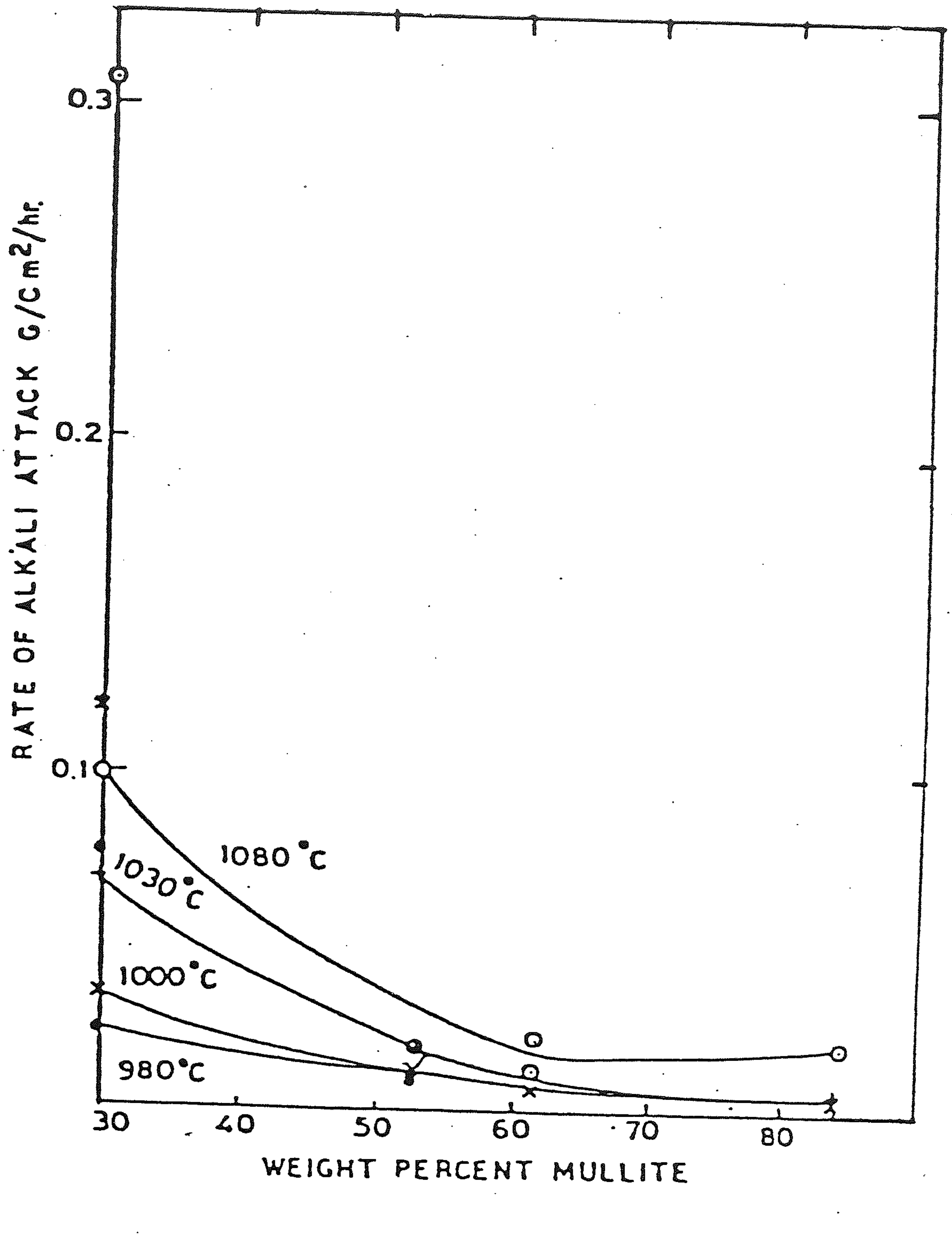
EFFECT OF ALUMINA CONTENT ON CERAMIC REFRACTRINESS

FIGURE 7.6



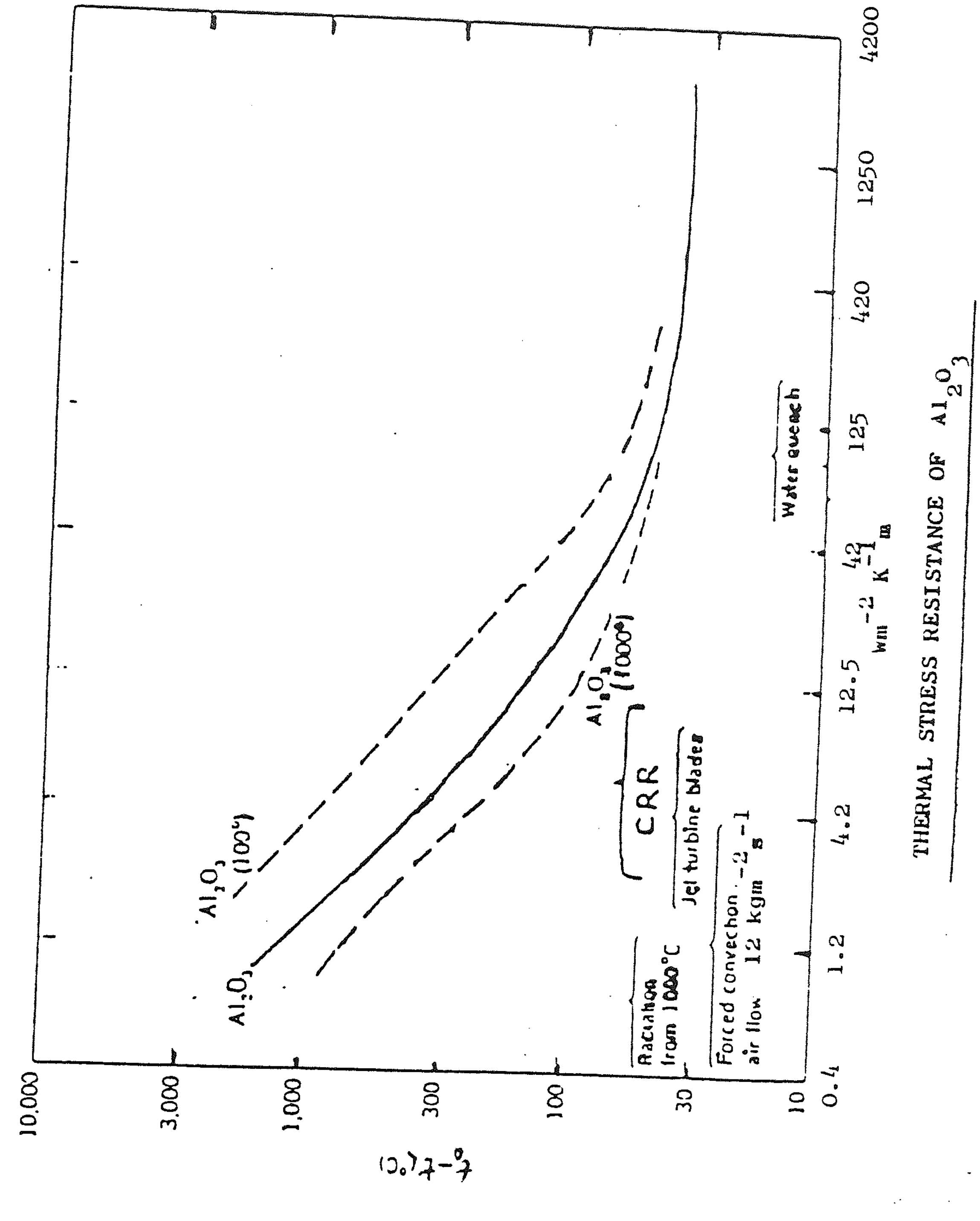
Effect of Al_2O_3 on Refractory Strength

FIGURE 7.7



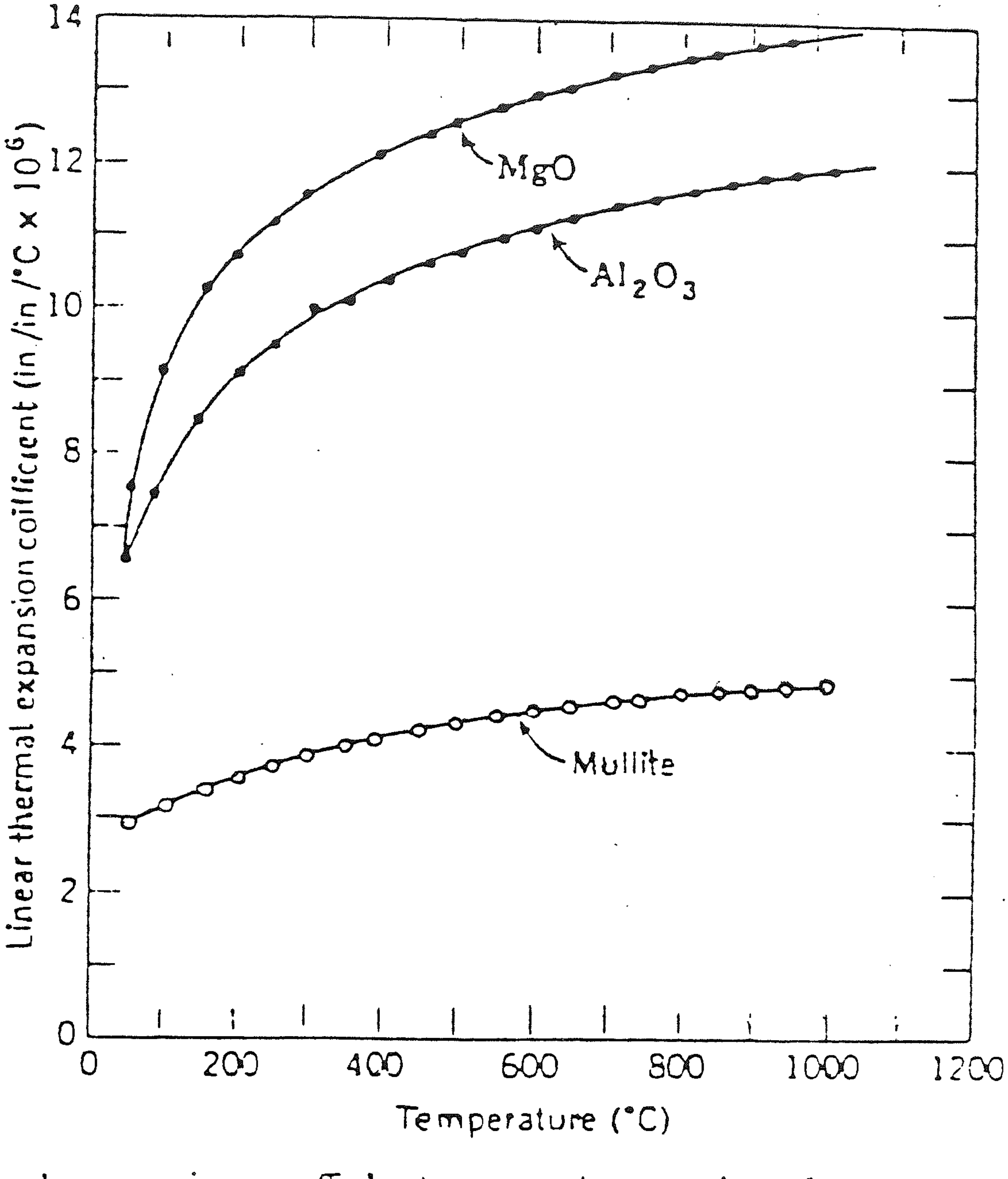
Effect of mullite content on rate of alkali attack at different temperature

FIGURE 7.8



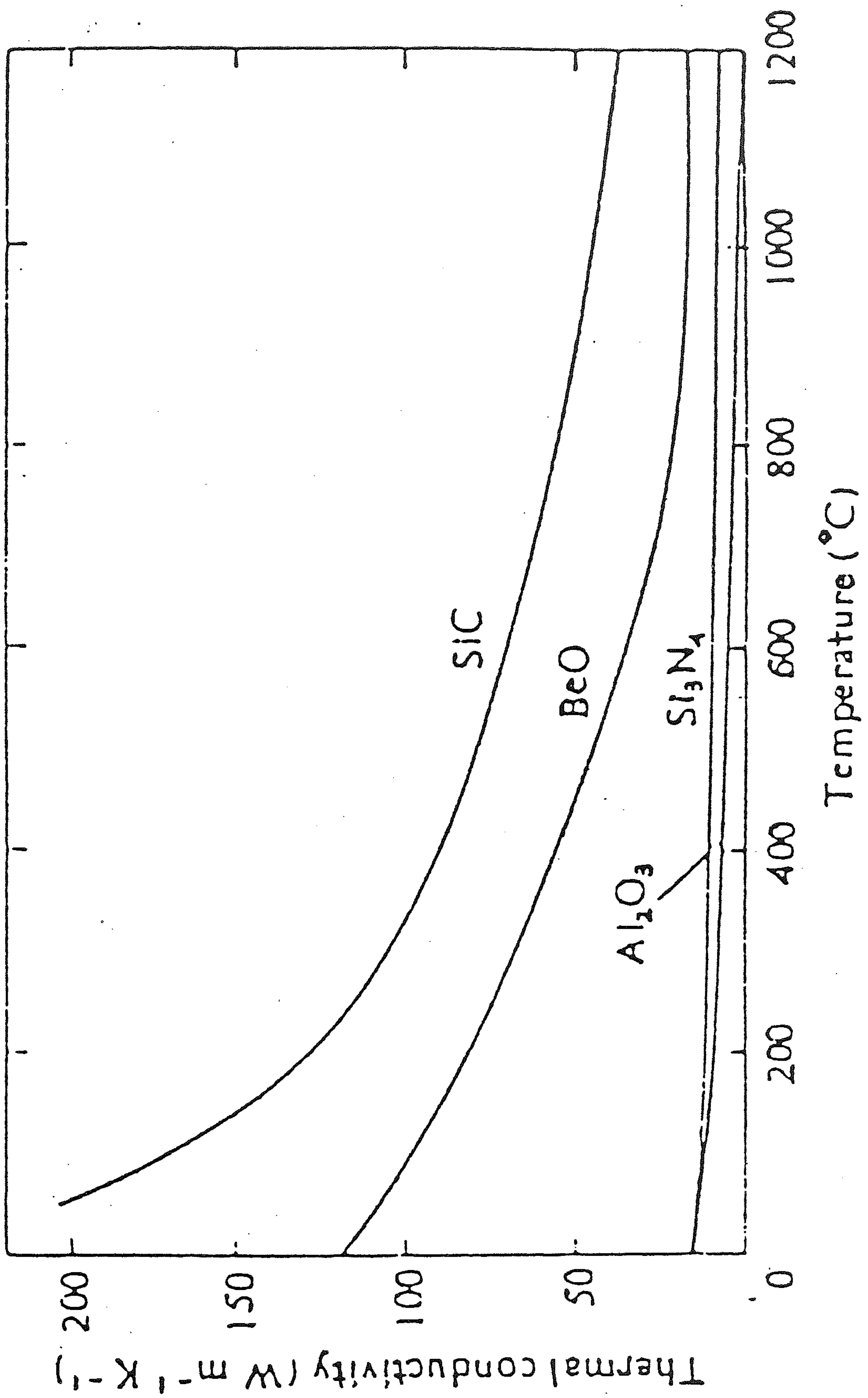
THERMAL STRESS RESISTANCE OF AL_2O_3

FIGURE 7.9



Thermal expansion coefficient versus temperature for some ceramics

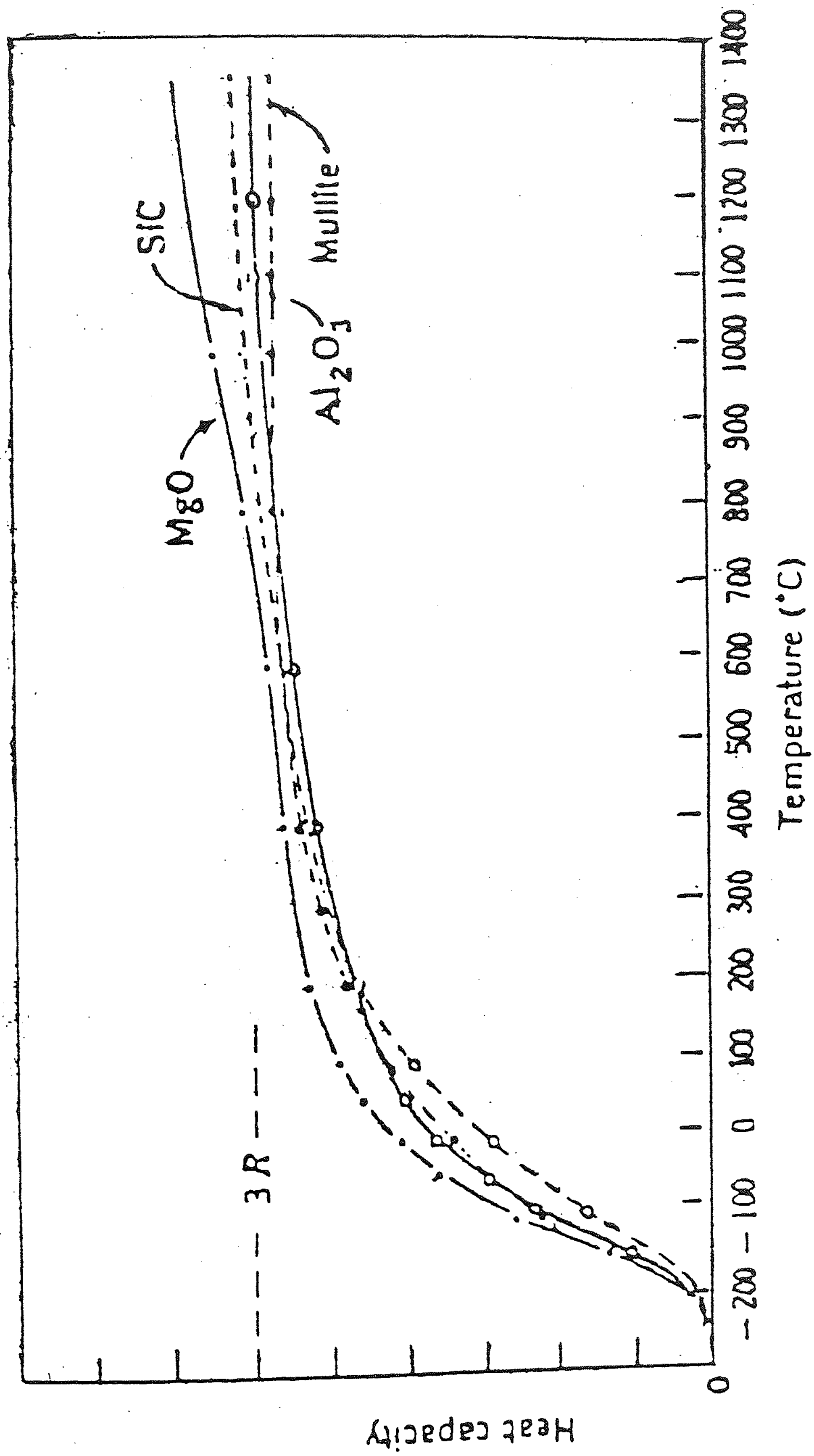
FIGURE 7.10



Temperature dependence of the thermal conductivity of some engineering ceramics

EFFECT OF TEMPERATURE ON THERMAL CONDUCTIVITY

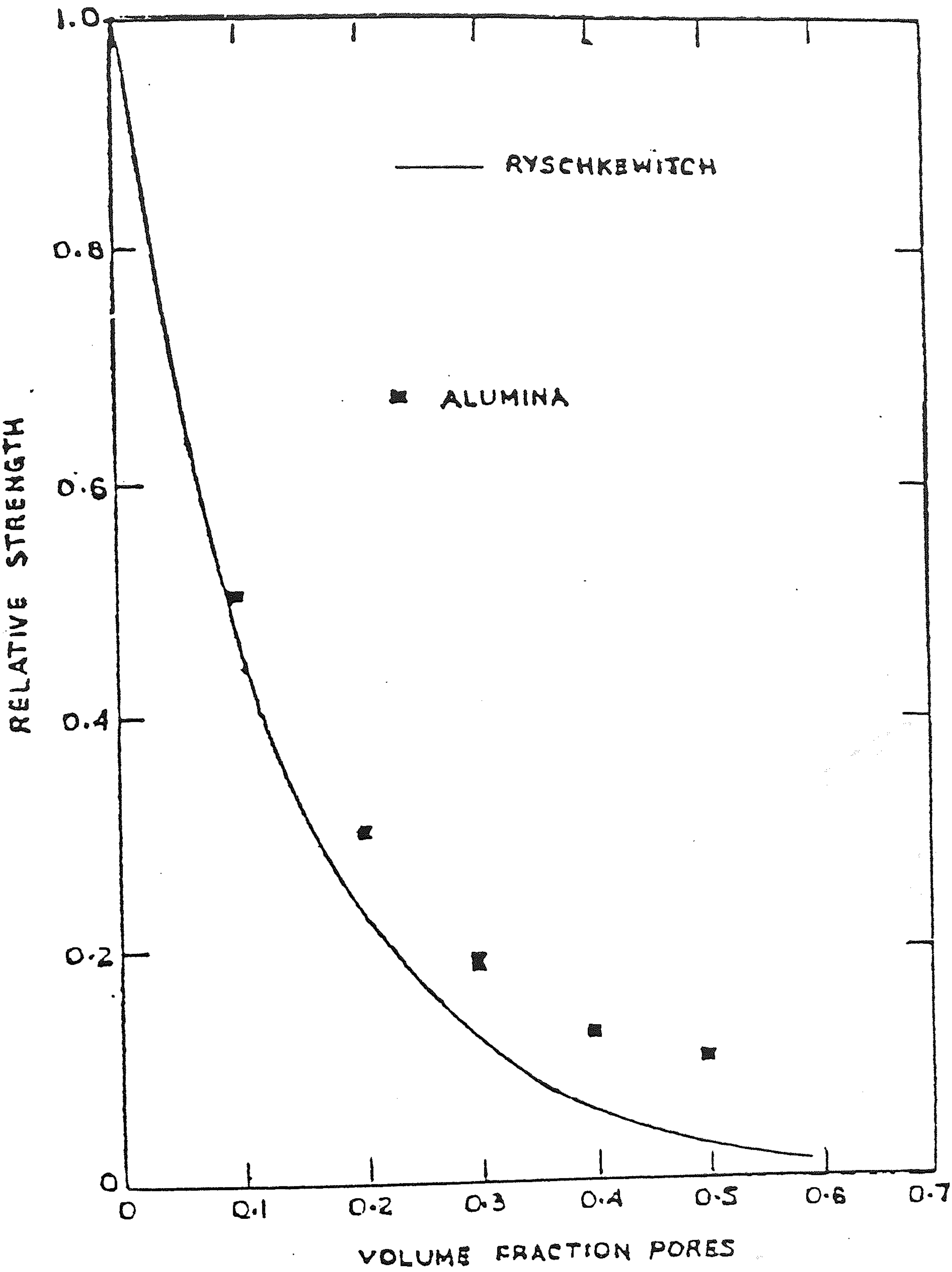
FIGURE 7.11



Heat capacity of some ceramic materials at different temperatures.

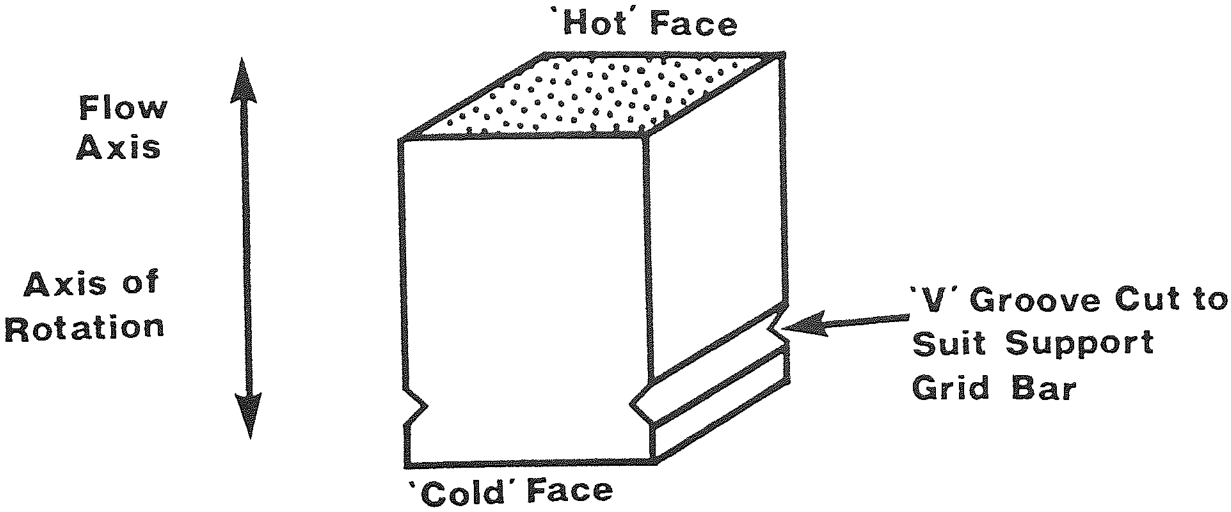
HEAT CAPACITY OF SOME CERAMIC MATERIALS

FIGURE 7.12

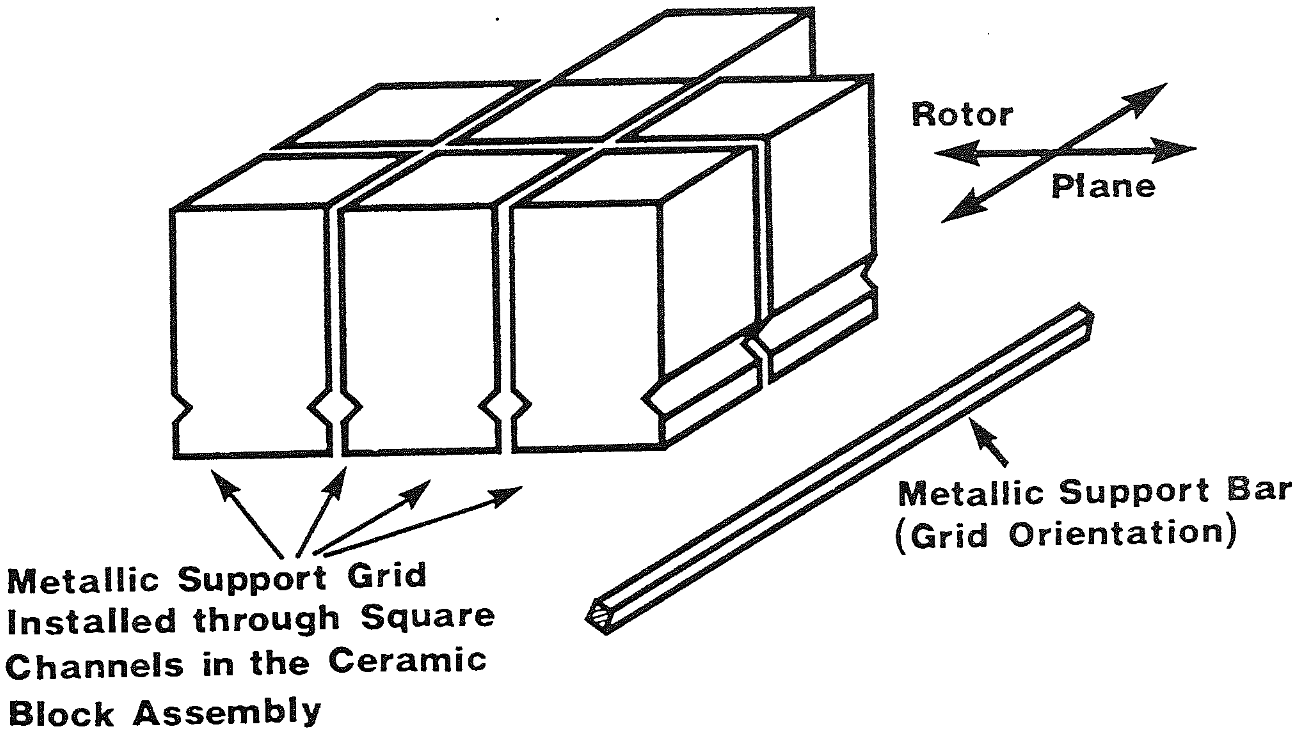


Effect of porosity on strength of Al_2O_3

FIGURE 7.13



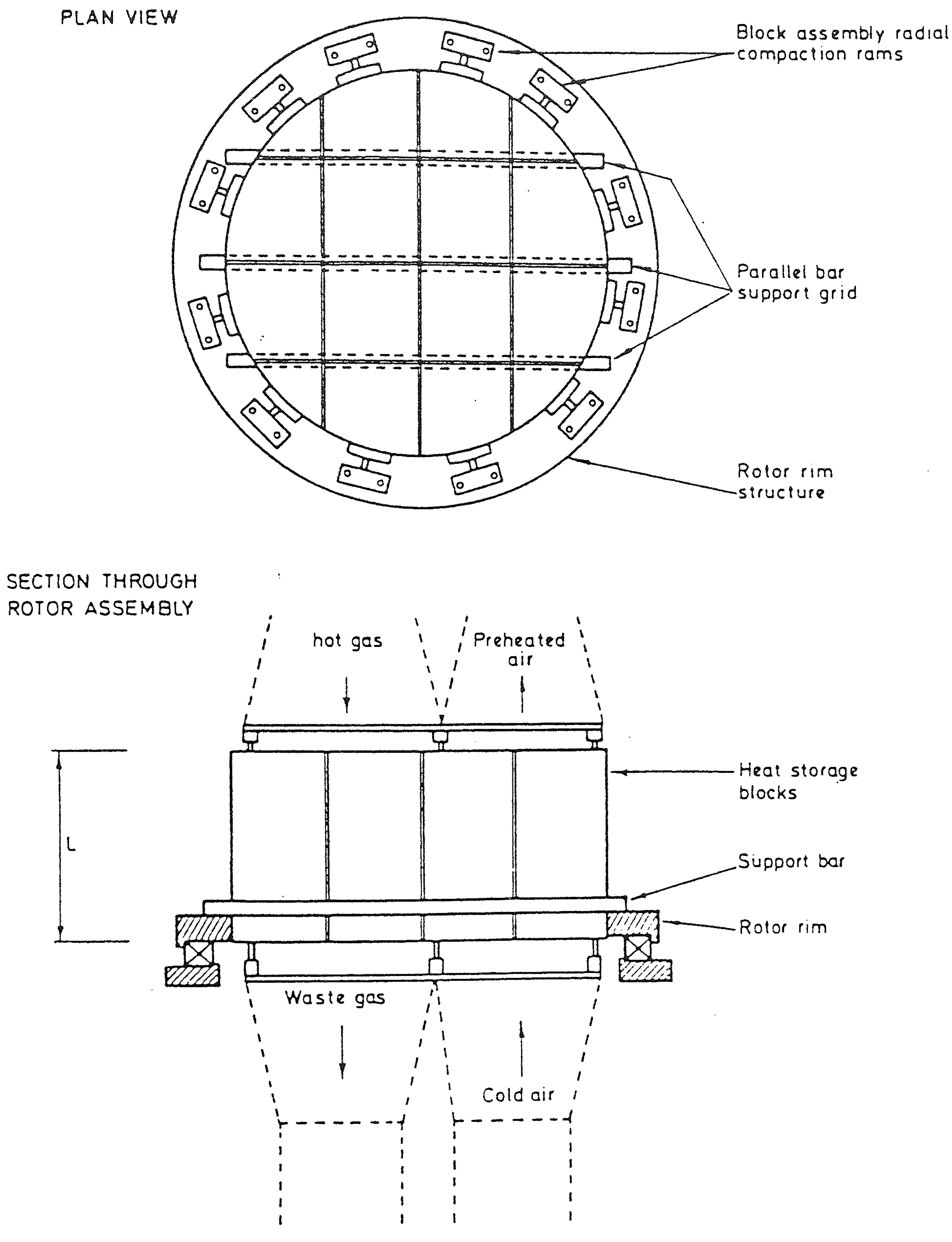
Heat Storage Block Design



Multi-Block Assembly Support Arrangements

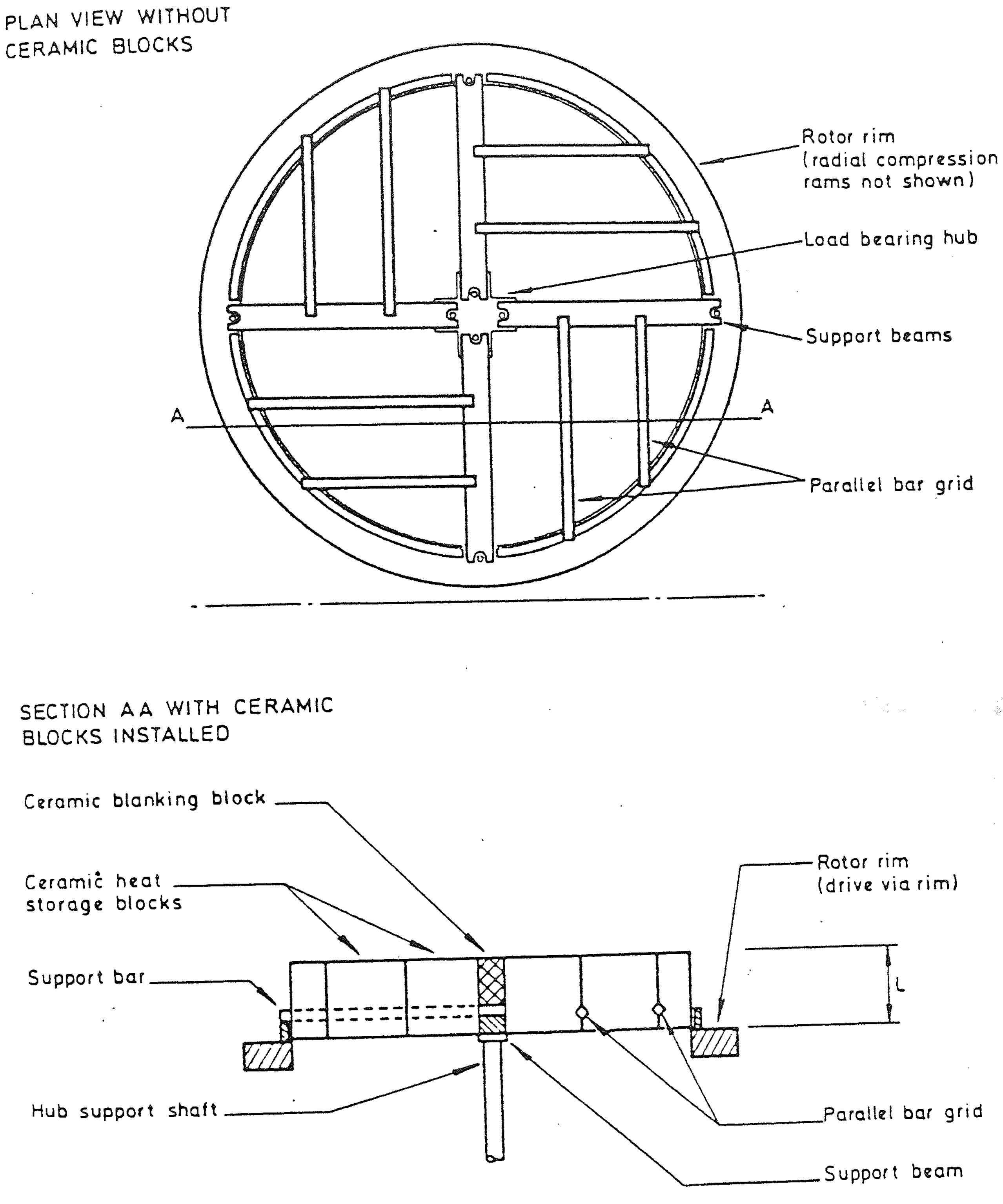
Rotor Matrix Structure

FIGURE 7.14



SMALL AND MEDIUM SIZED ROTORS

FIGURE 7.15

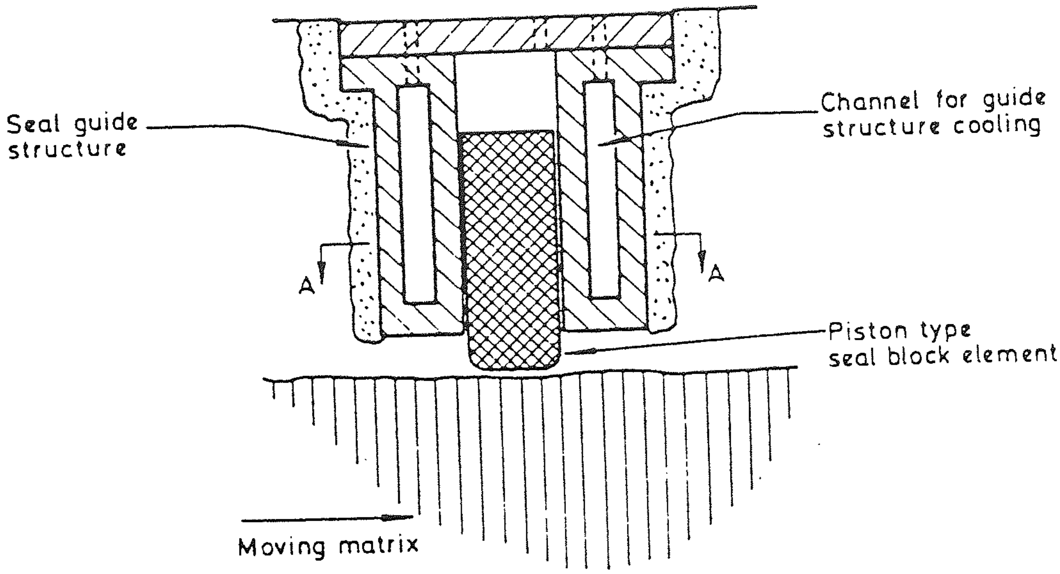


CERAMIC BLOCK SUPPORT STRUCTURE ARRANGEMENT

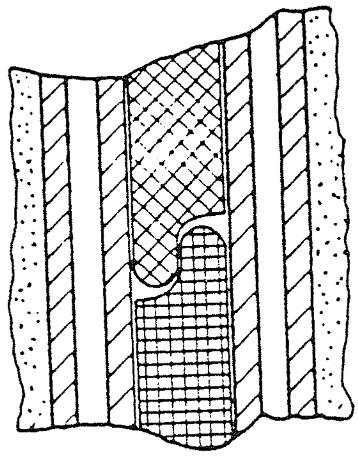
FIGURE 7.16

Sealing System

FIGURE 7.17



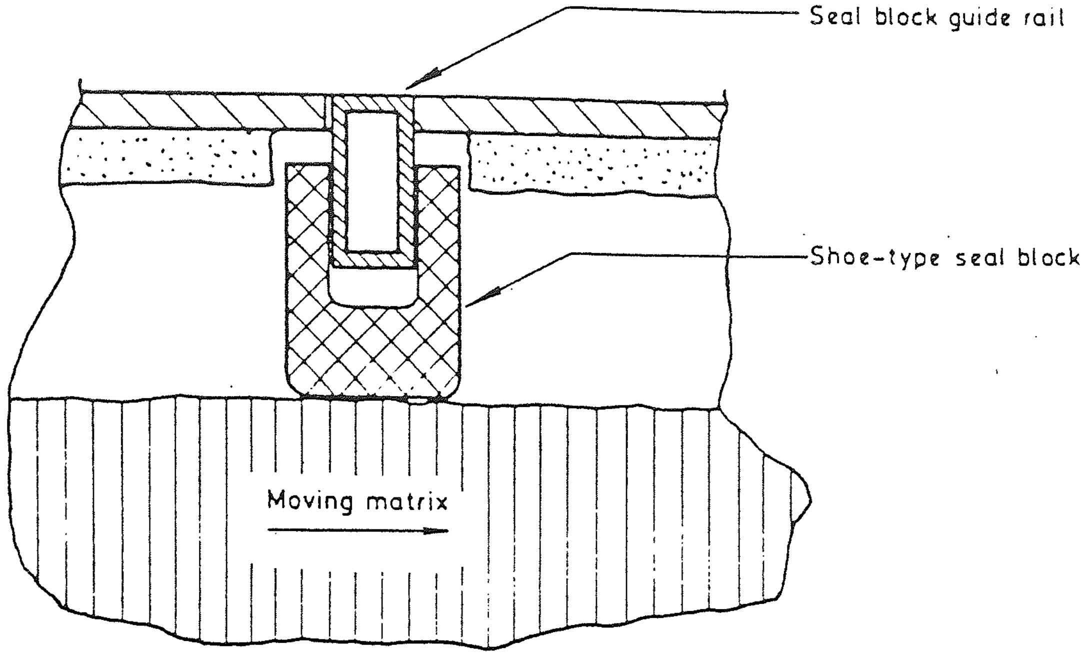
TRANSVERSE SECTION

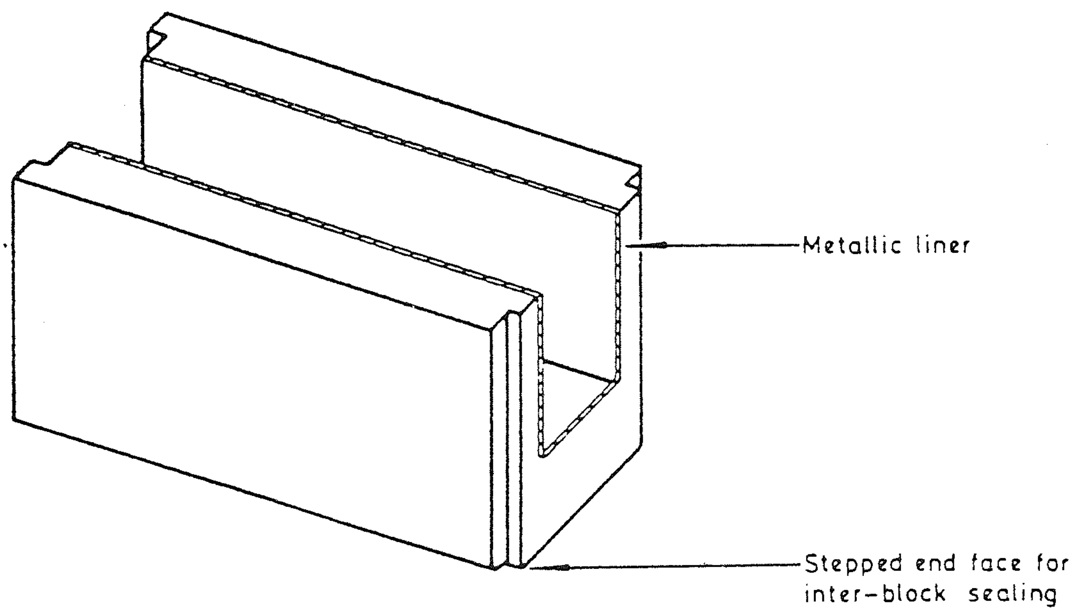


SECTION AA VIEW SHOWING
LAP JOINTING OF THE PISTON BLOCK ASSEMBLY

PISTON TYPE SEAL SYSTEM

FIGURE 7.18

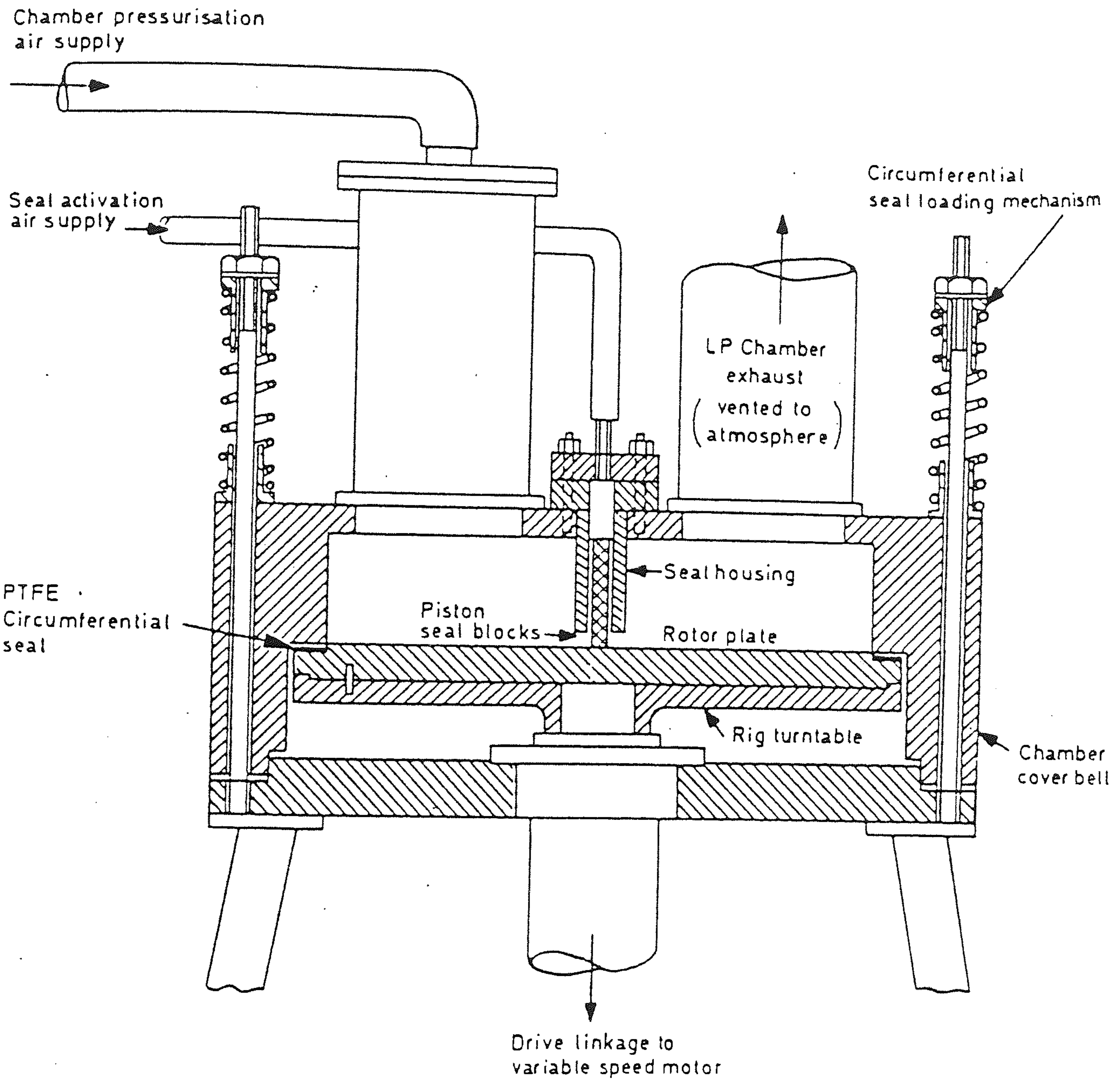




CERAMIC SHOE ELEMENT

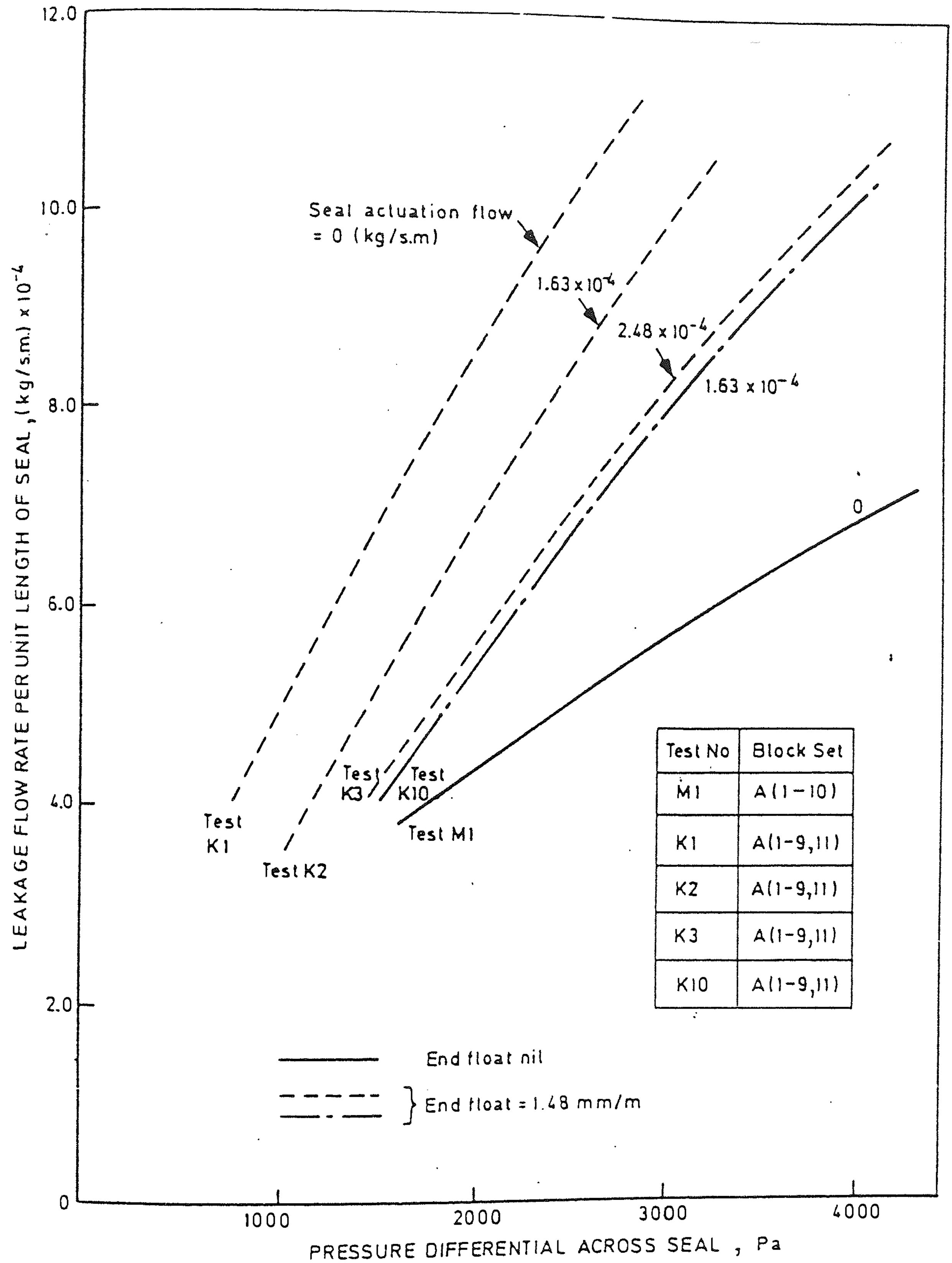
THE SHOE BLOCK - FLEXIBLE SEAL CONCEPT

FIGURE 7.19



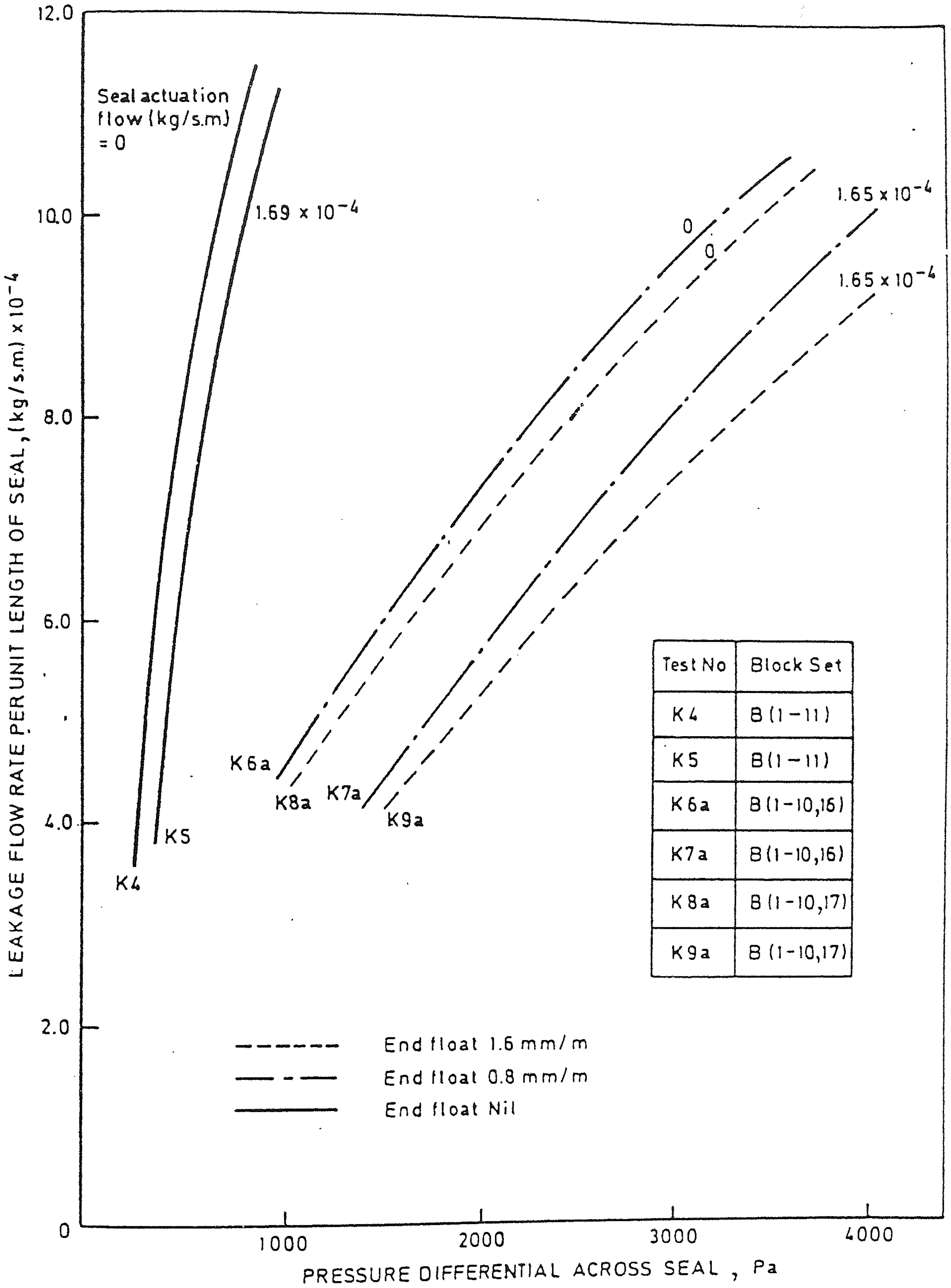
SEAL TEST CHAMBER

FIGURE 7.20



SQUARE BUTT JOINTED SEAL PERFORMANCE

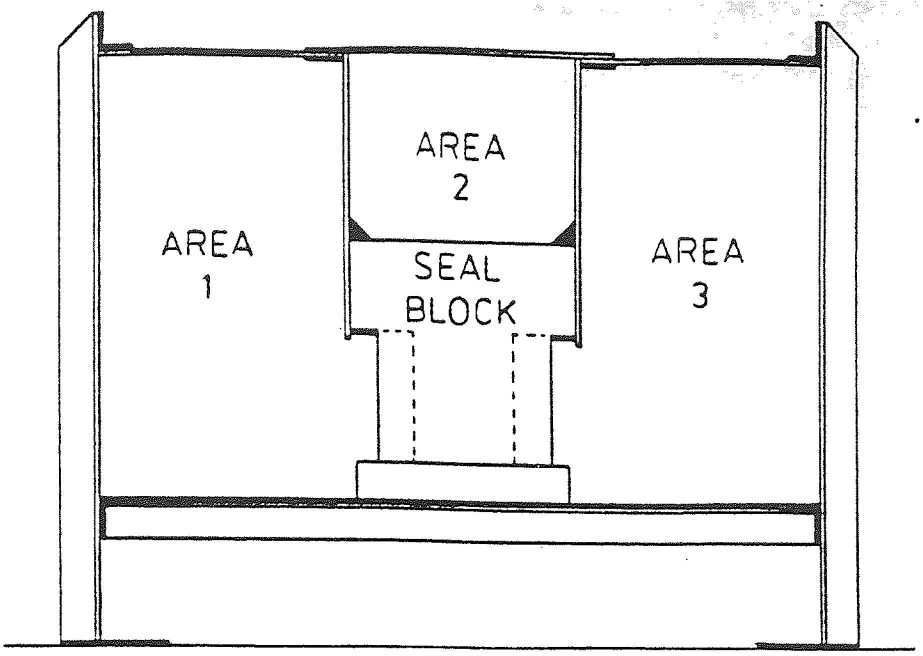
FIGURE 7.21



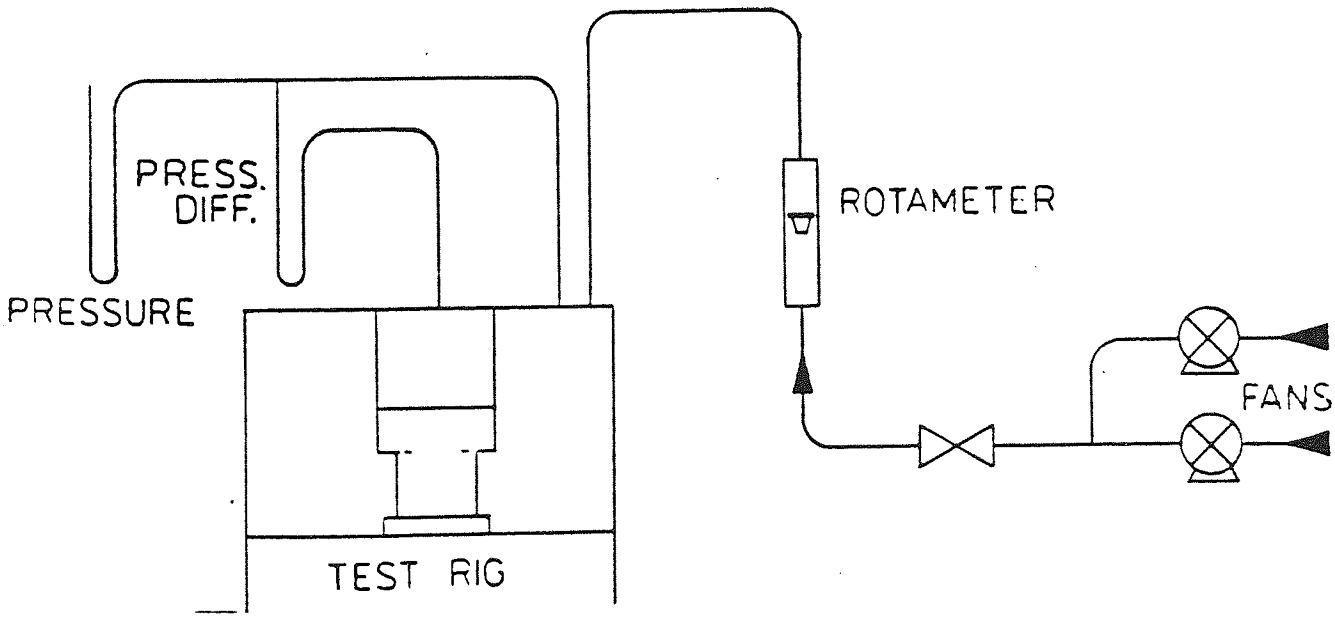
Test No	Block Set
K4	B (1-11)
K5	B (1-11)
K6a	B (1-10,16)
K7a	B (1-10,16)
K8a	B (1-10,17)
K9a	B (1-10,17)

STAGGERED LAP JOINTED SEAL PERFORMANCE

FIGURE 7.22

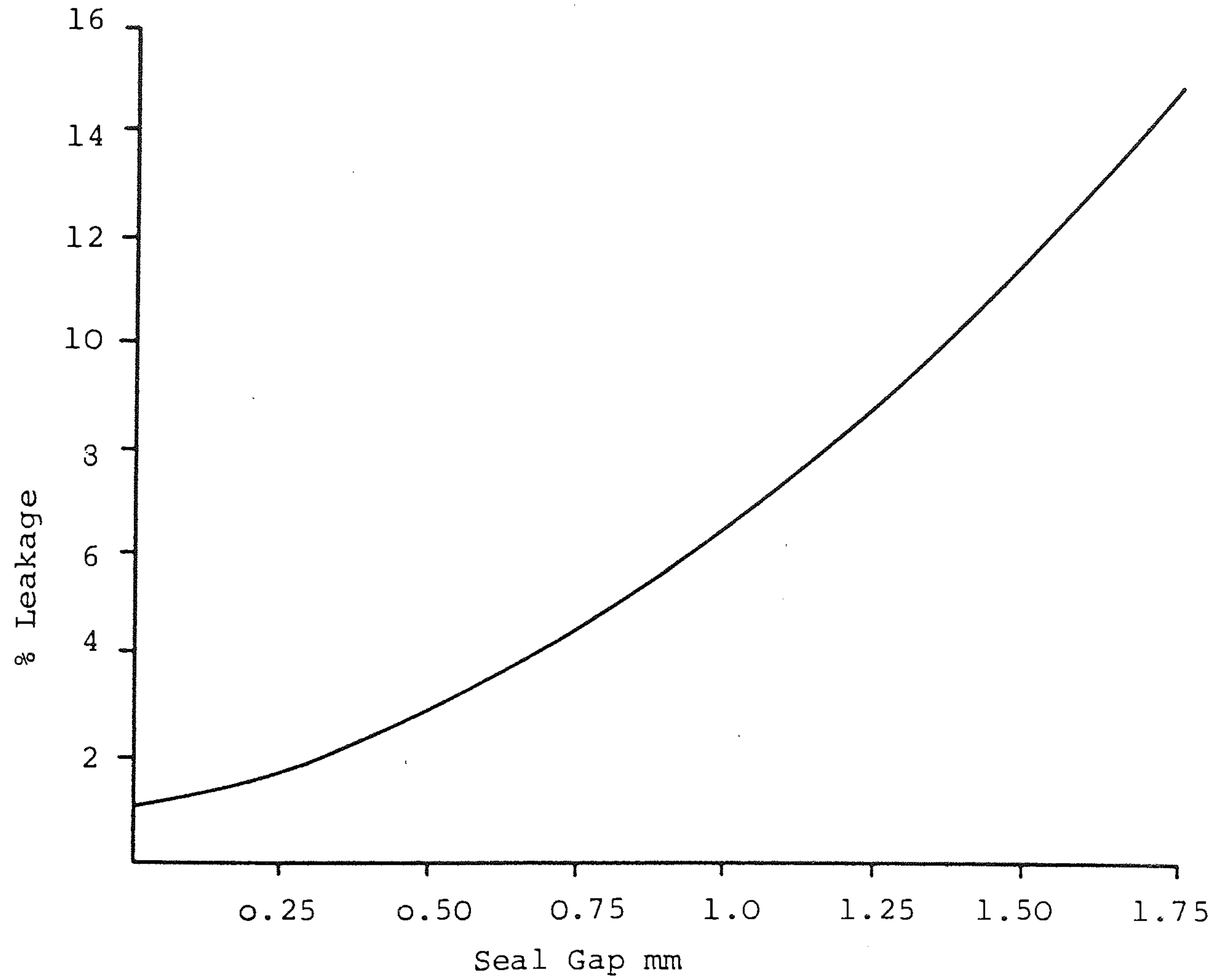


CROSS SECTION THROUGH TEST RIG.



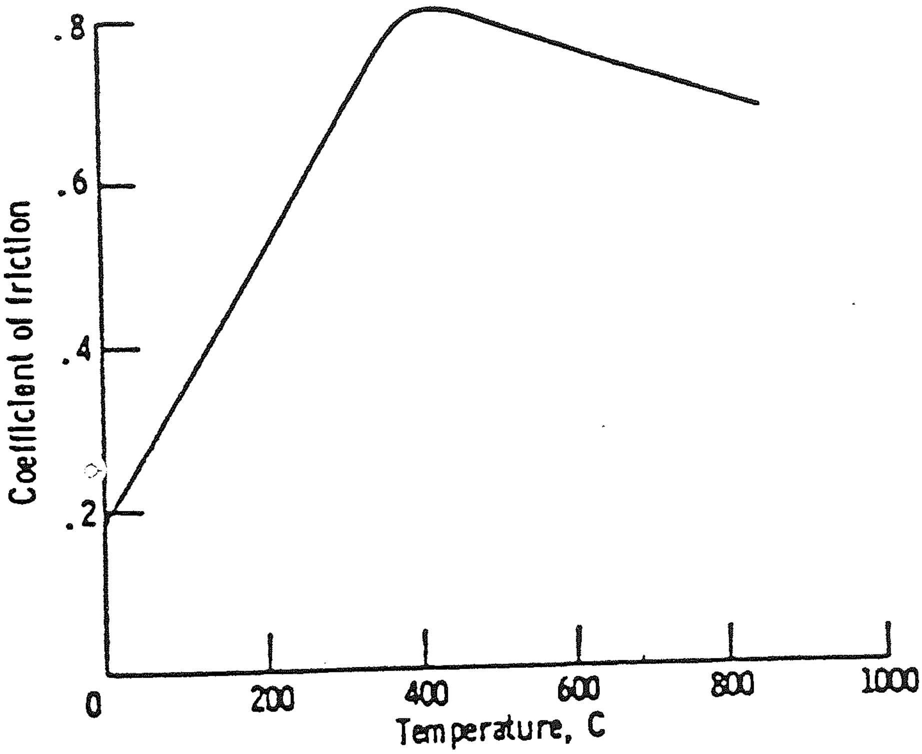
PROXIMITY SEAL
FLOW DIAGRAM

FIGURE 7.23



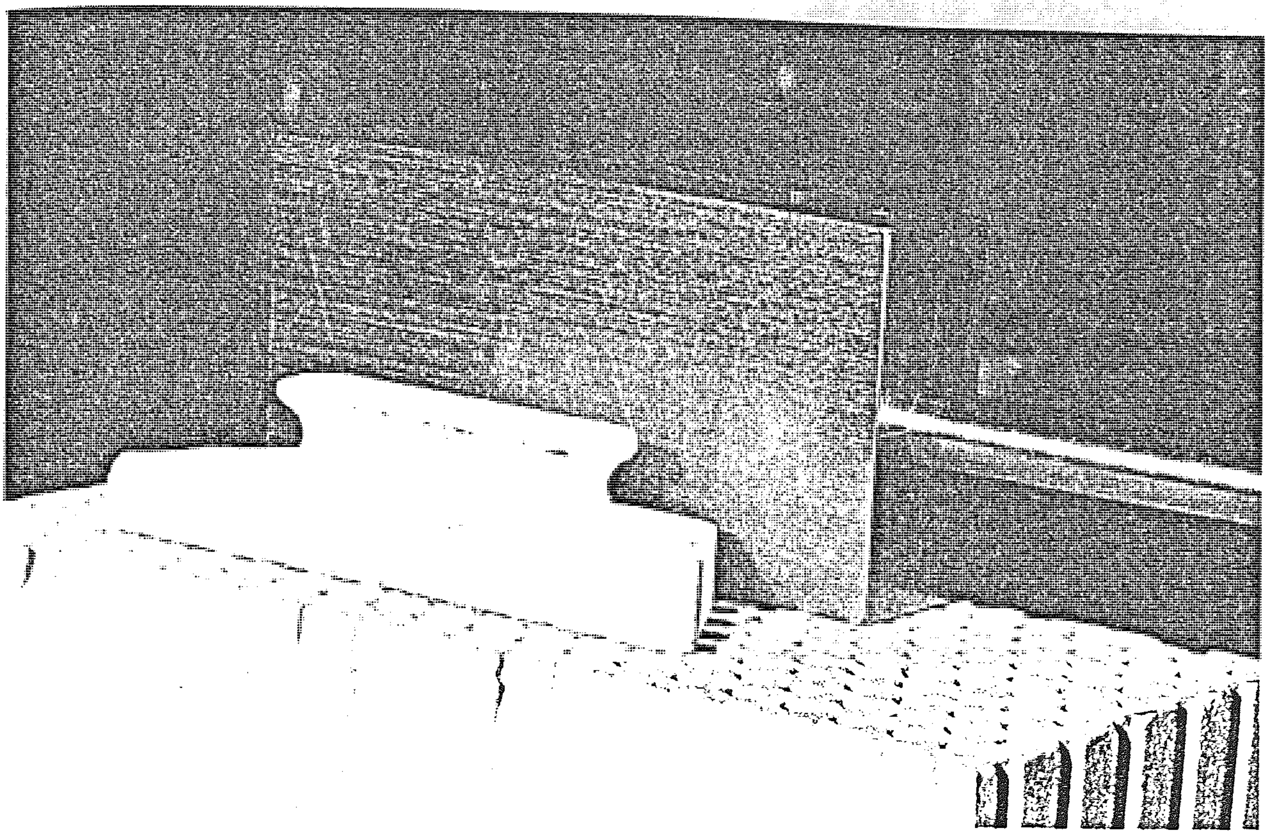
PERCENTAGE LEAKAGE V/S SEAL GAP FOR A 3M CRR UNIT
AT 2.5 KG/SEC AND 4" WG.

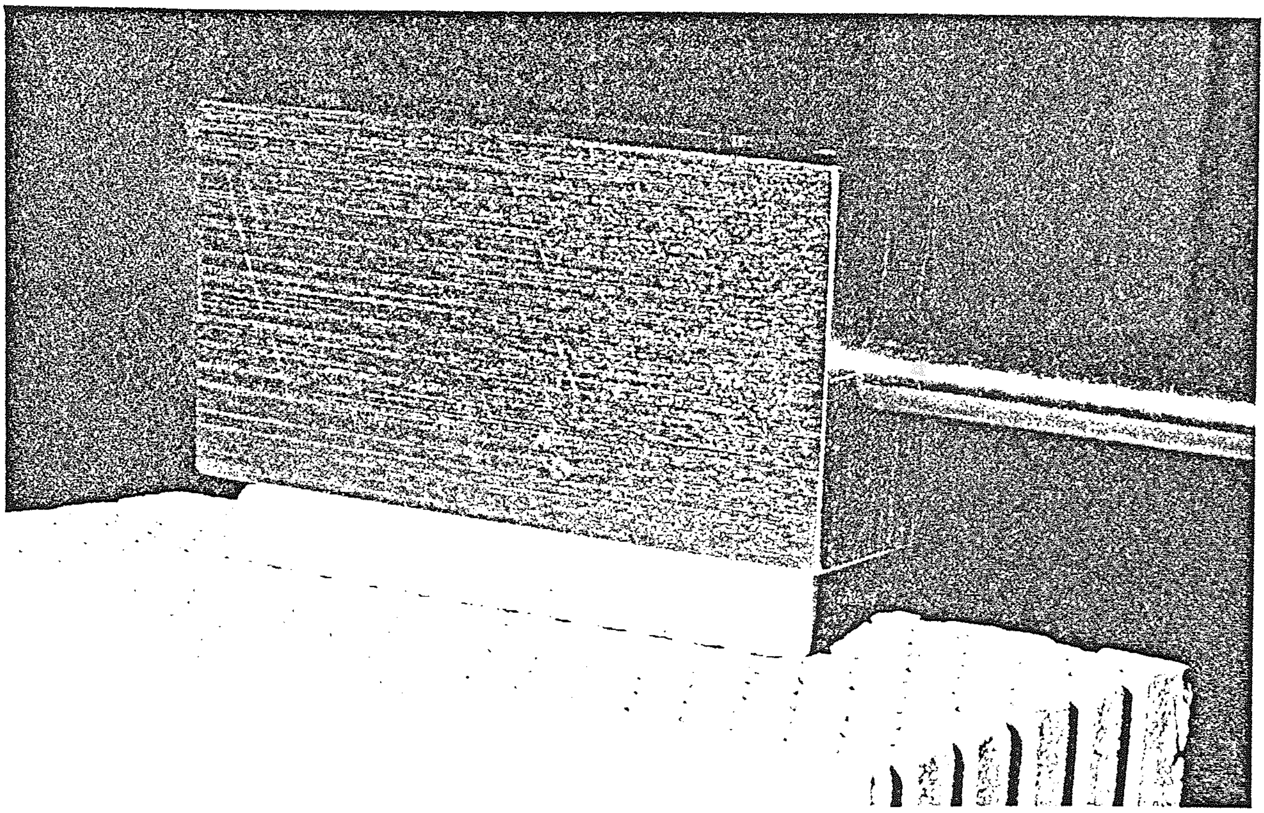
FIGURE 7.24



Effect of temperature on friction coefficient of Al_2O_3.

FIGURE 7.25



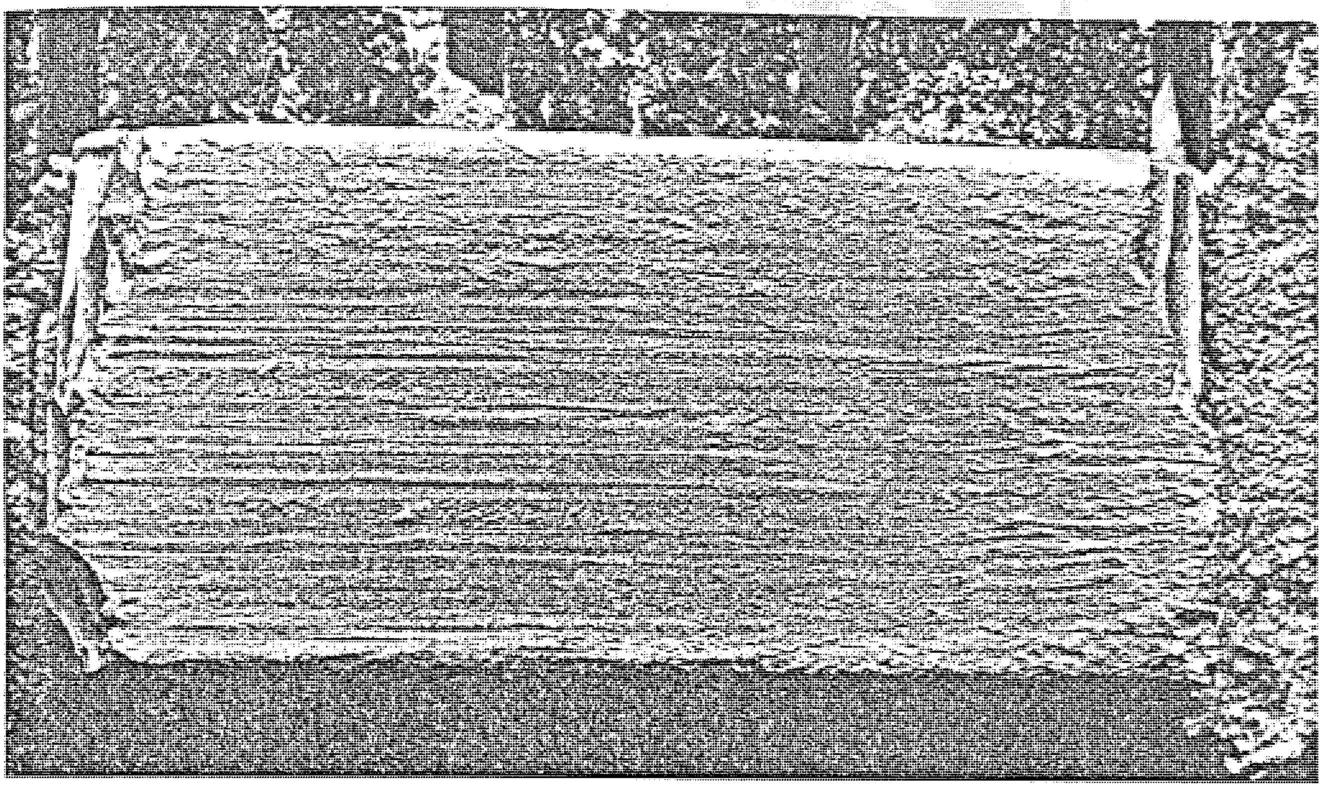


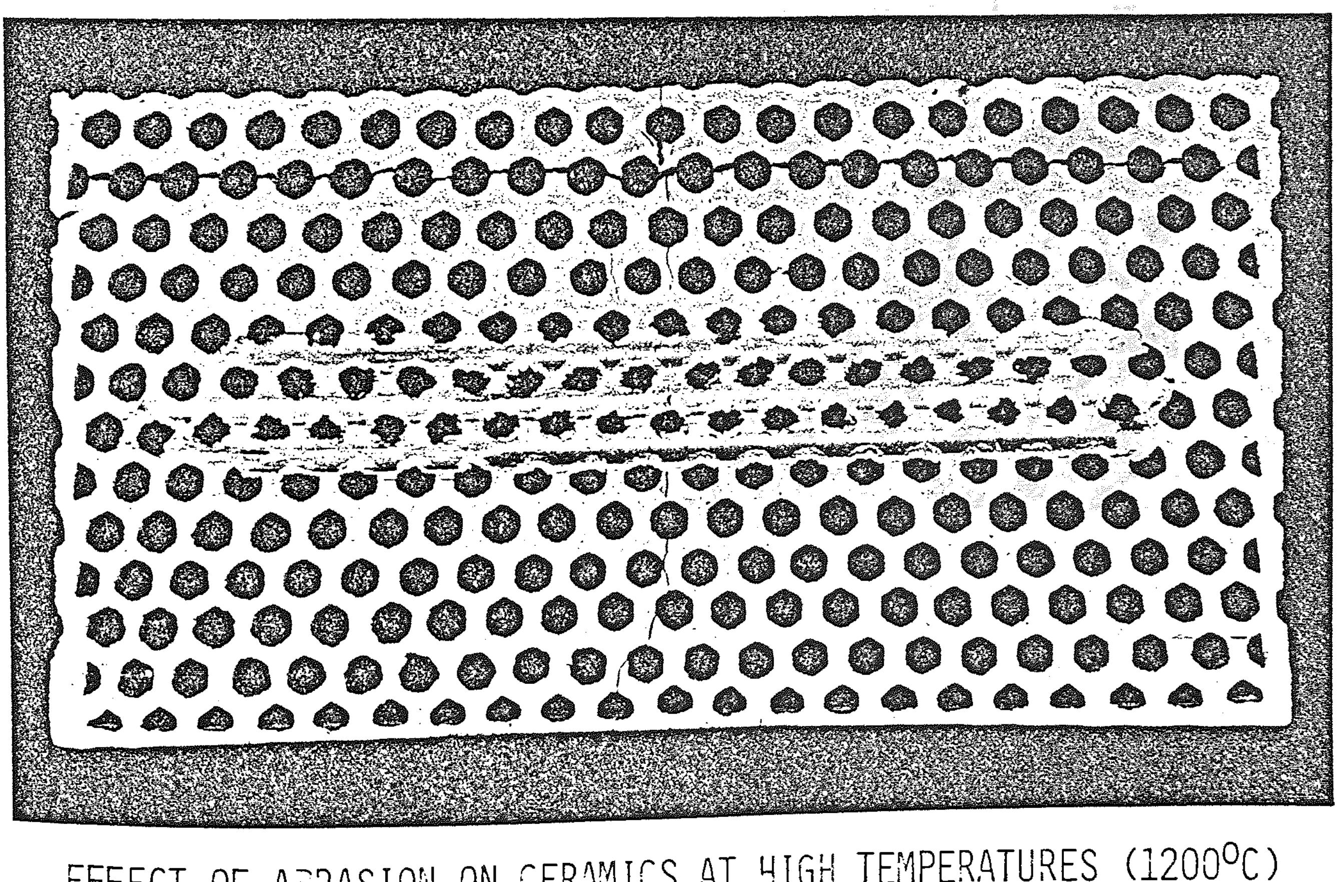
TEST PIECES : CERAMIC WEAR

FIGURE 7.26

Diagram of apparatus to determine abrasion by rubbing.

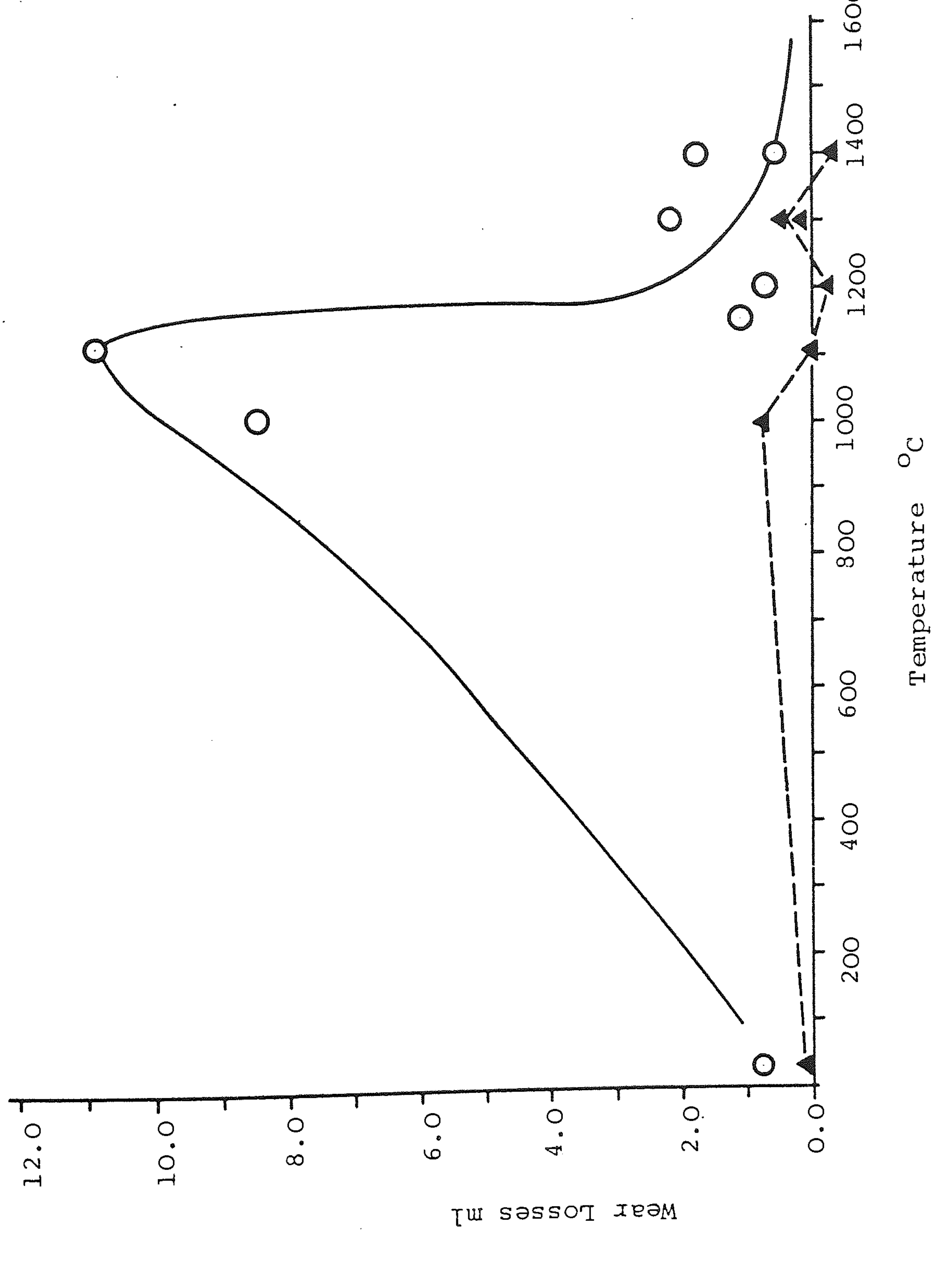
IGURE 7.27





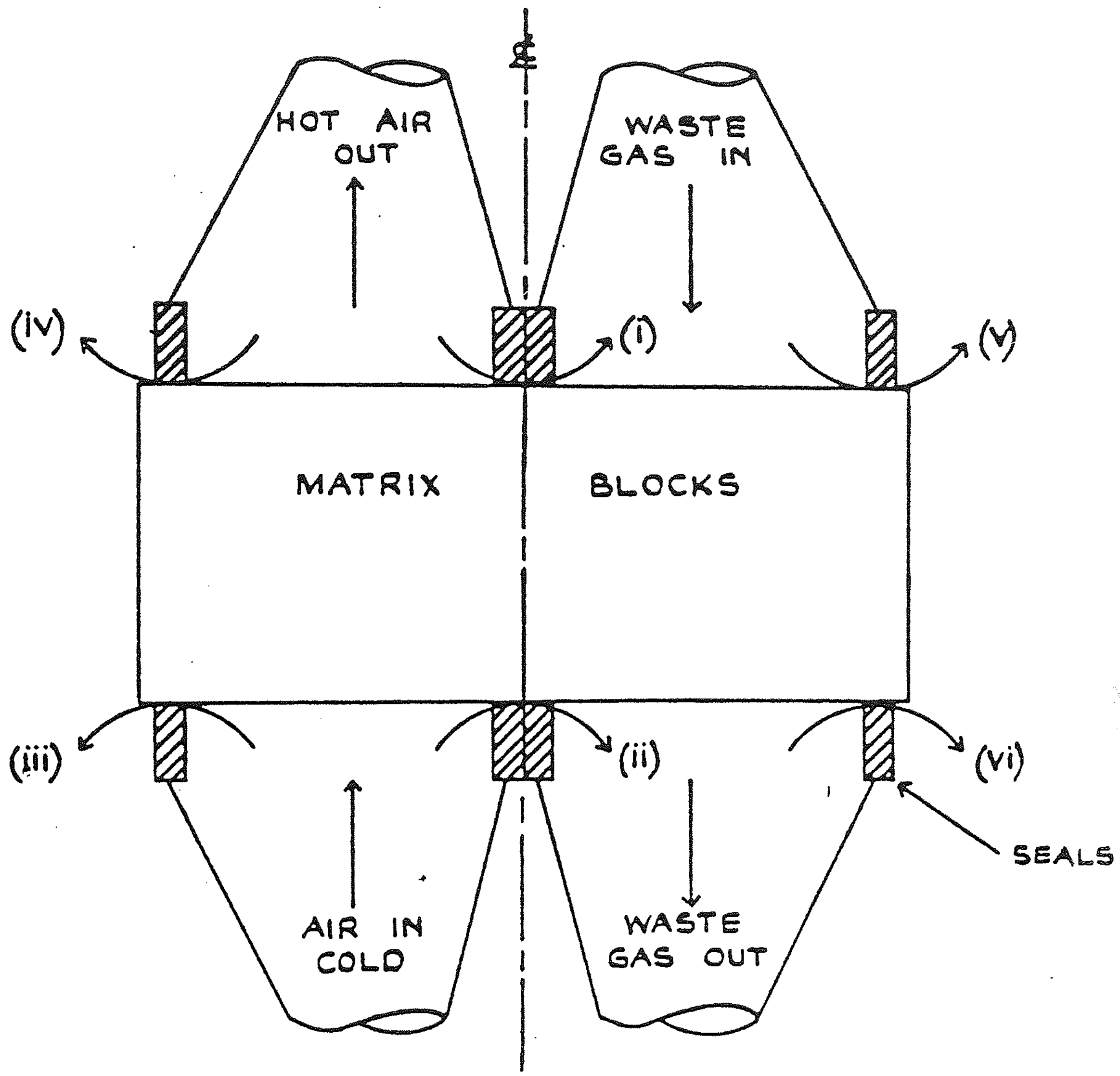
EFFECT OF ABRASION ON CERAMICS AT HIGH TEMPERATURES (1200°C)

FIGURE 7.23



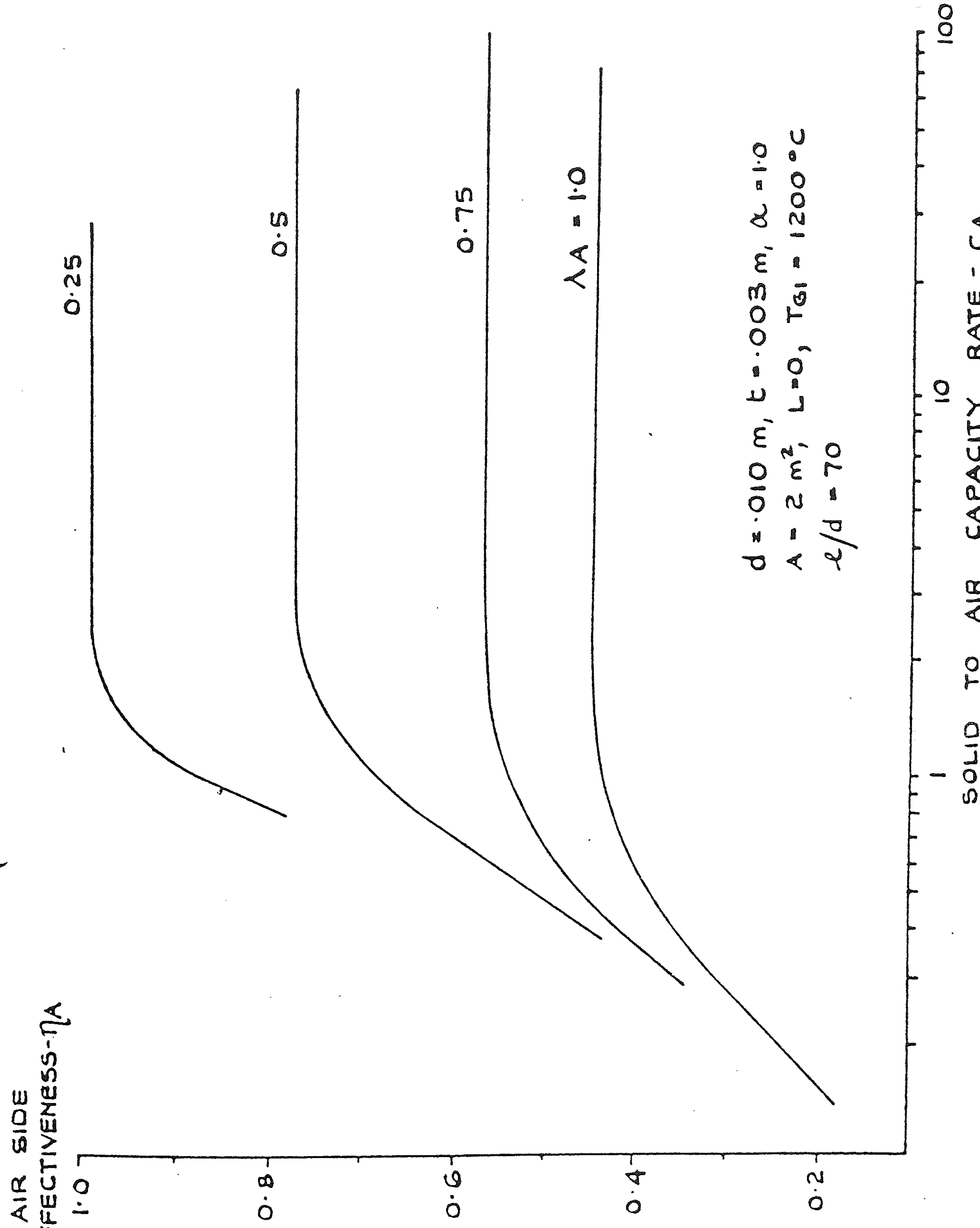
CERAMIC WEAR RELATED TO TEMPERATURE

FIGURE 7.30



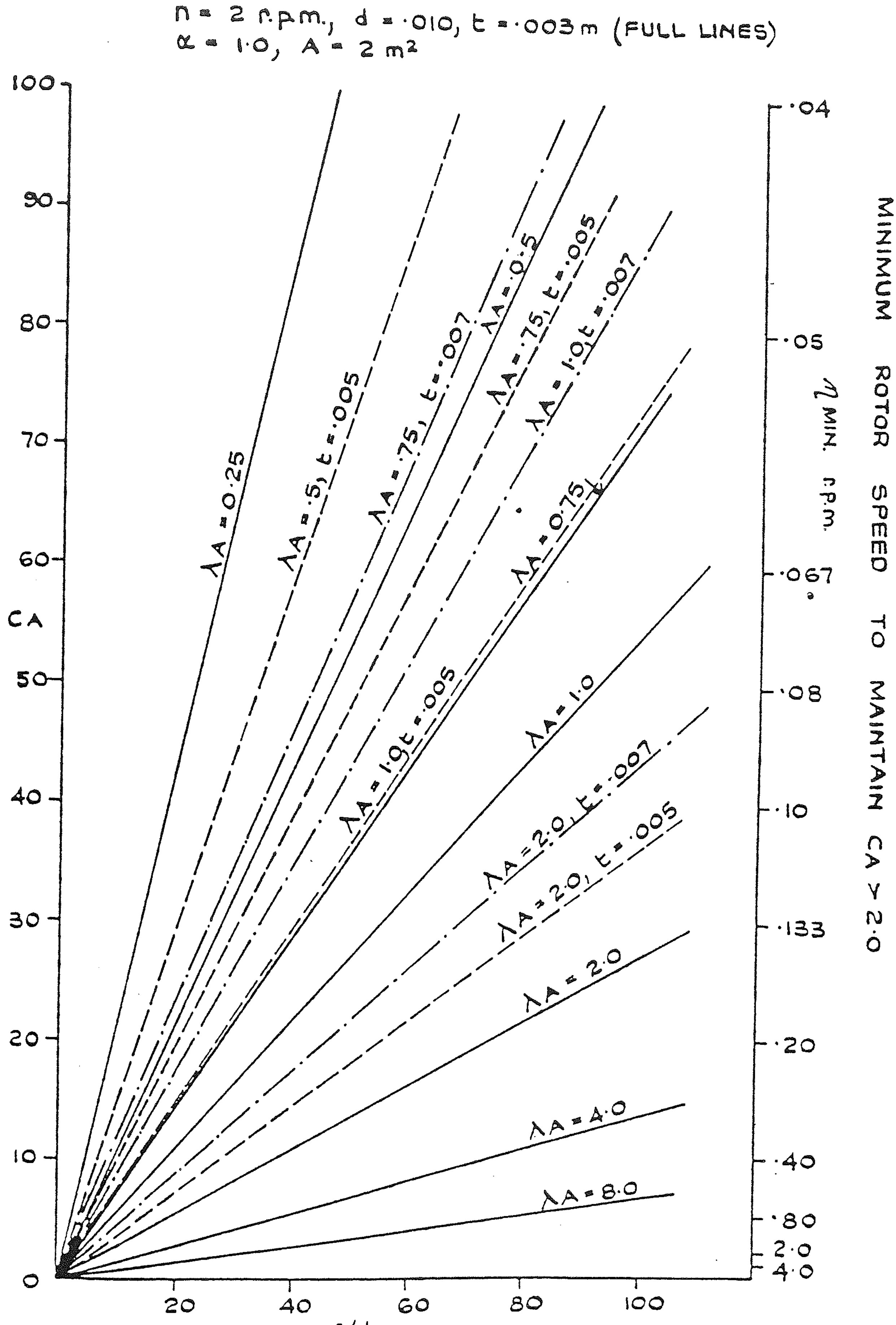
ROTARY REGENERATOR - POTENTIAL AIR AND WASTE GAS LEAKAGE PATHS

FIGURE 7.31



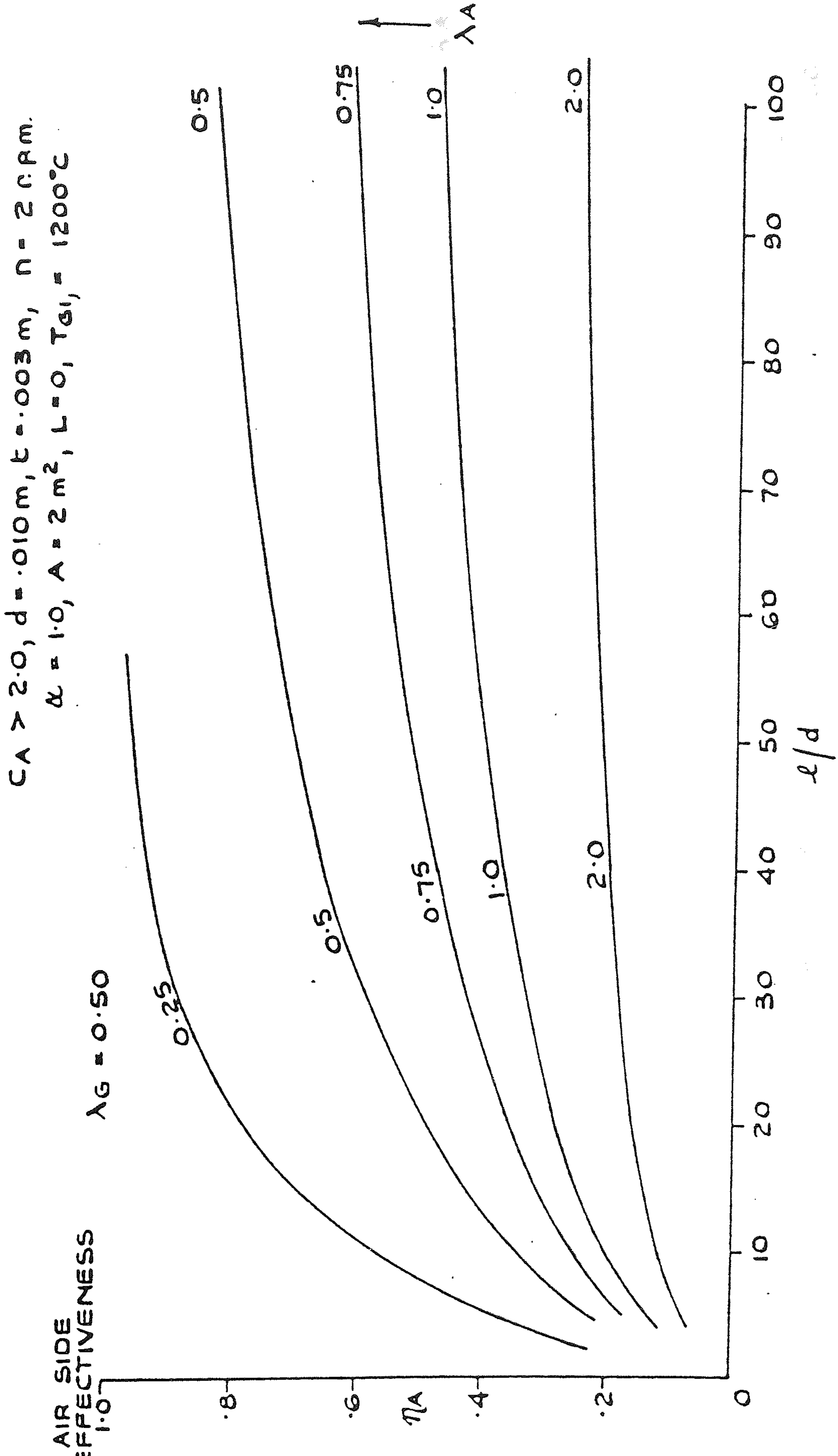
EFFECT OF SOLID TO AIR CAPACITY RATE ON AIR SIDE EFFECTIVENESS

FIGURE 7.32



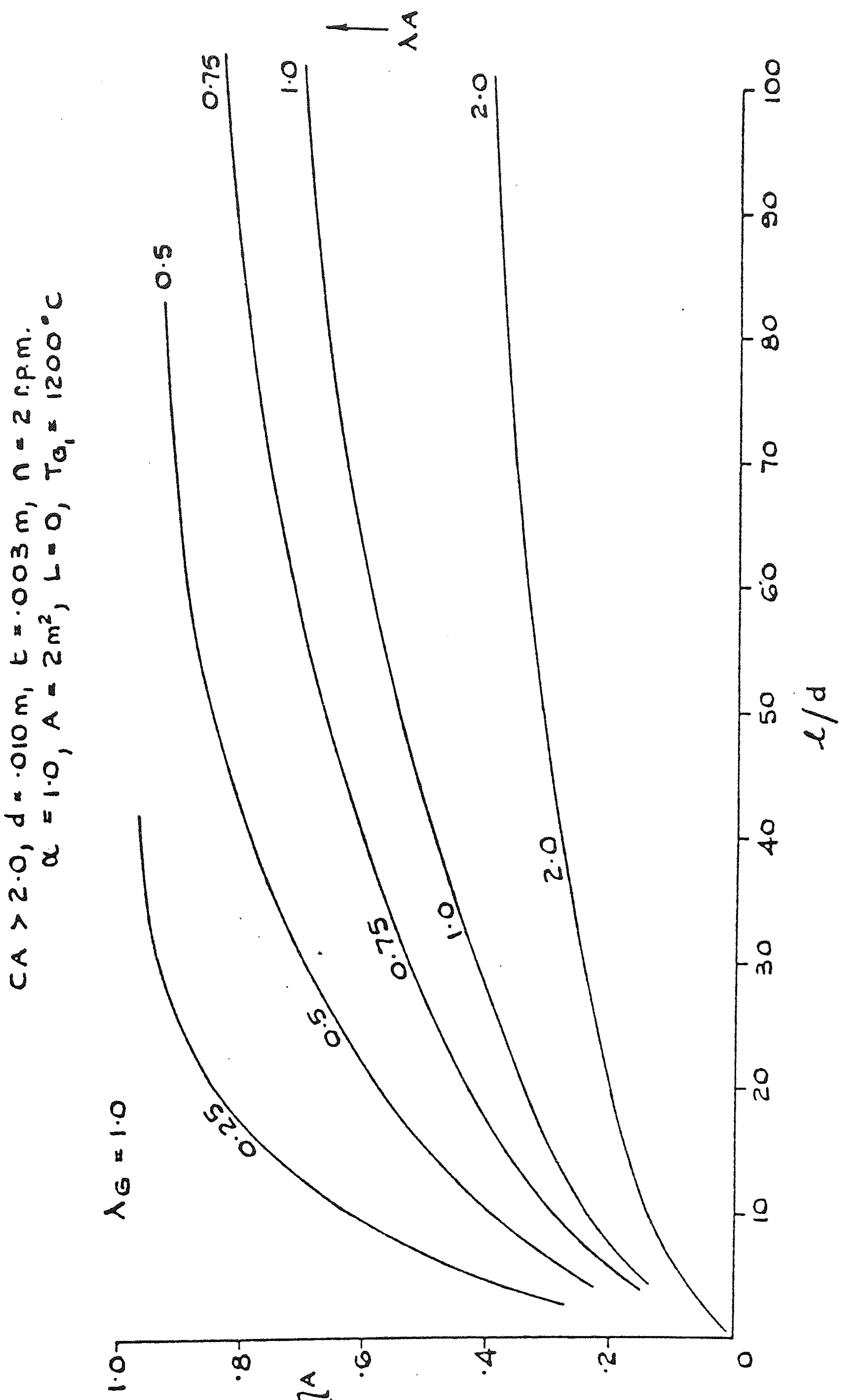
VARIATION OF CA WITH ℓ/d FOR VARIOUS VALUES OF λA AND t, WITH d = ·010 m.

FIGURE 7.33



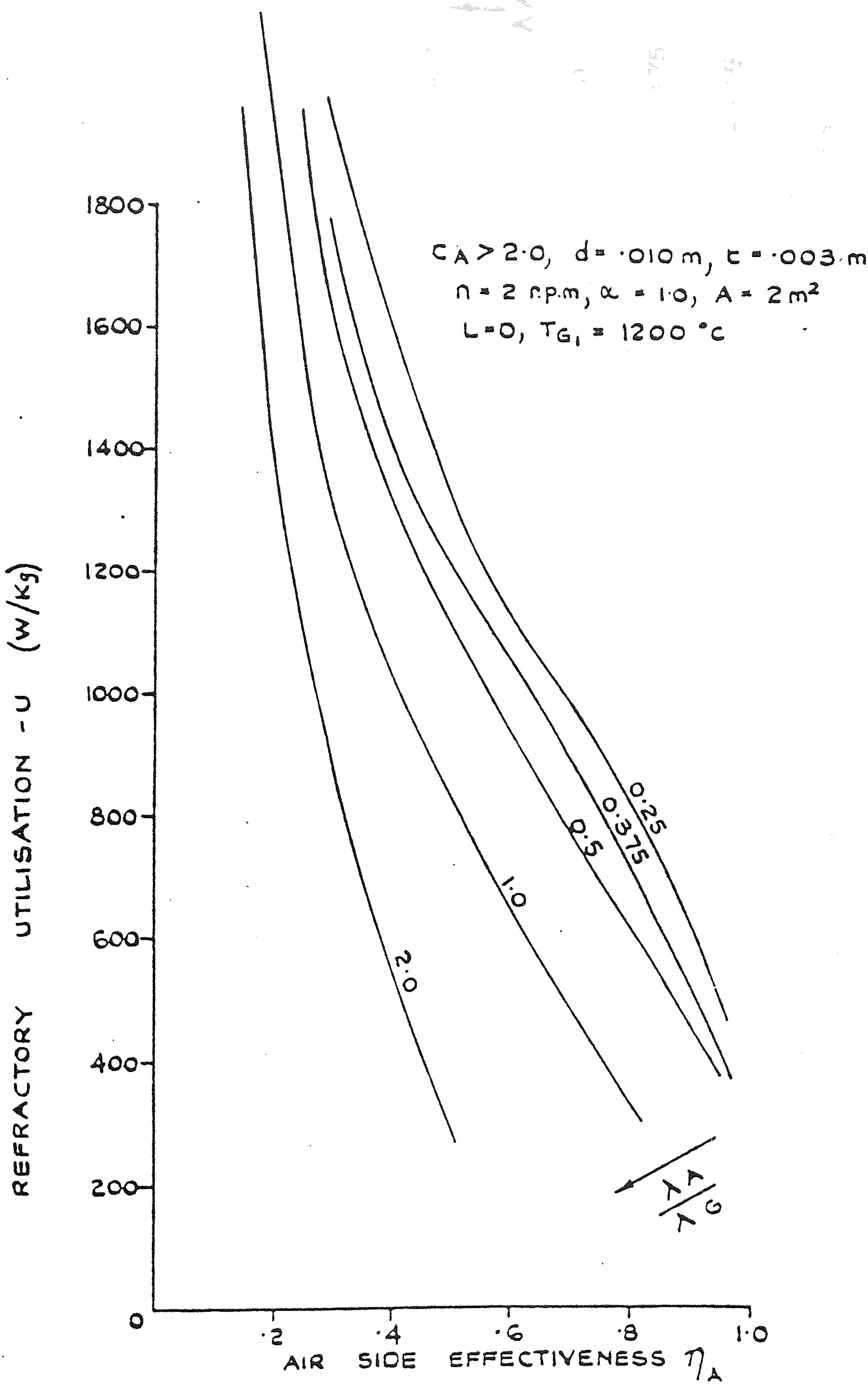
ℓ/d VS η_A FOR VARIOUS VALUES OF λ_A AND $\lambda_G = 0.5$

FIGURE 7.34



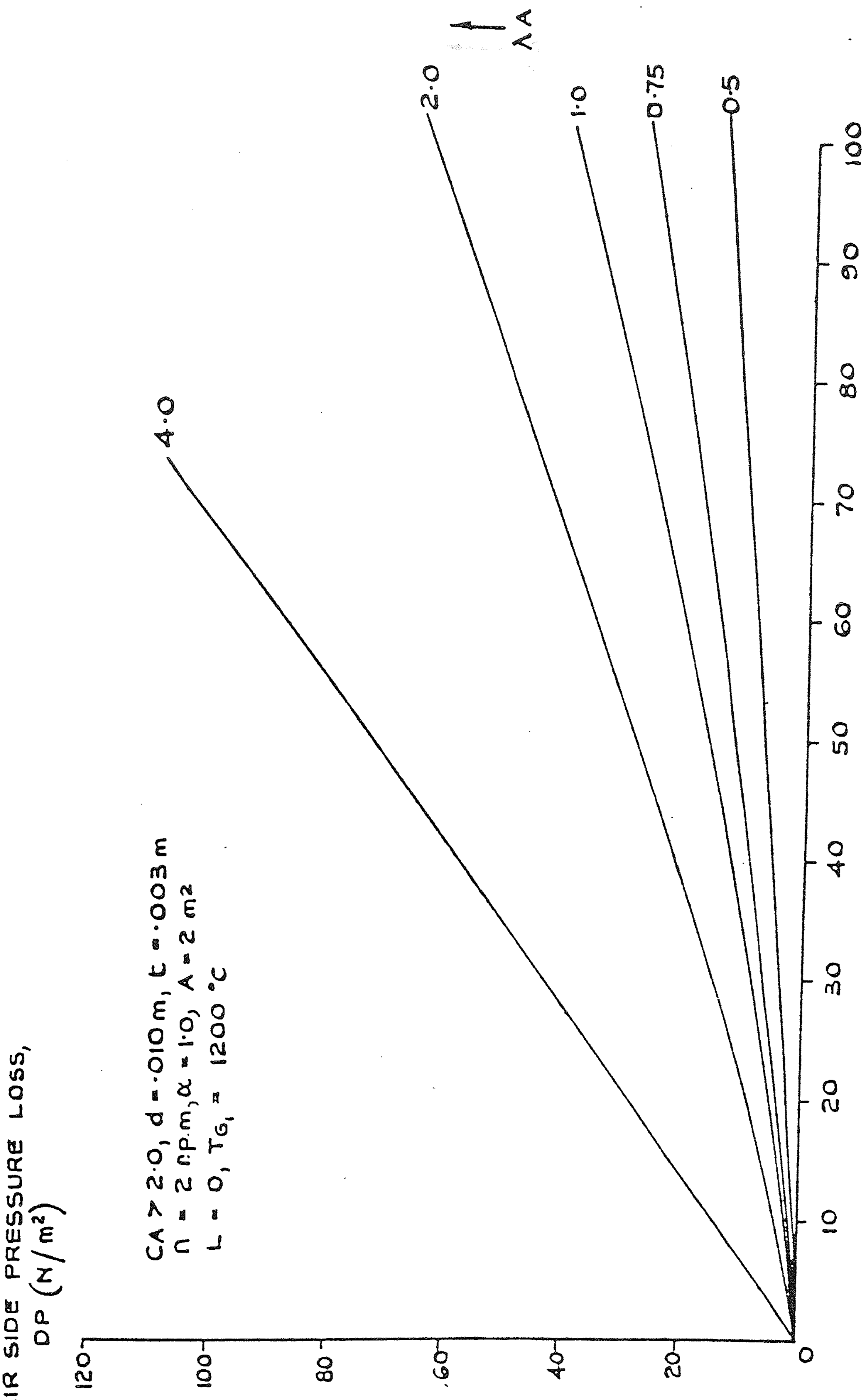
ℓ/d VS η_A FOR VARIOUS VALUES OF λ_A AND $\lambda_G = 1 \cdot 0$

FIGURE 7.35



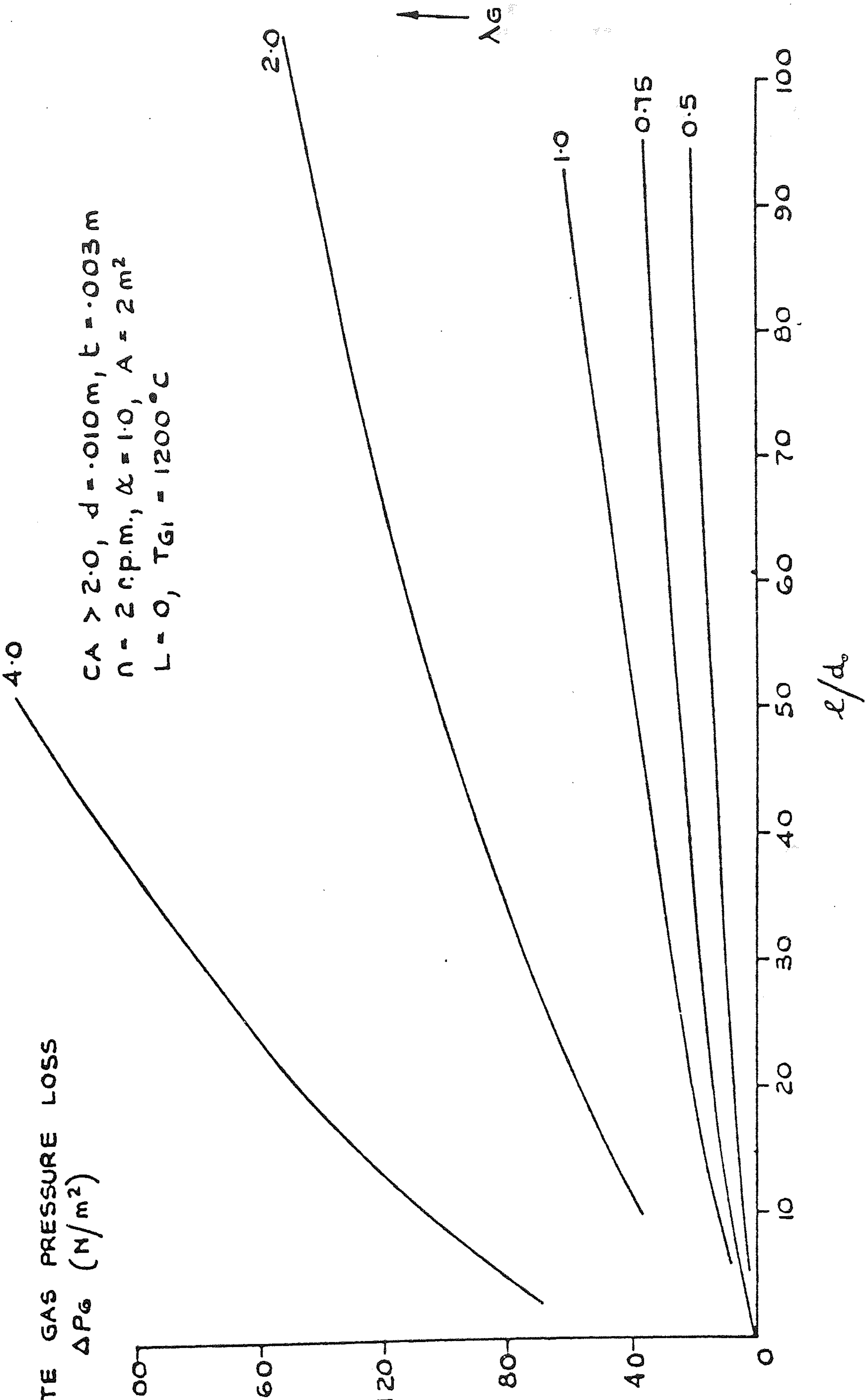
U Vs η_A FOR VARIOUS RATIOS OF λ_A/λ_G

FIGURE 7.36



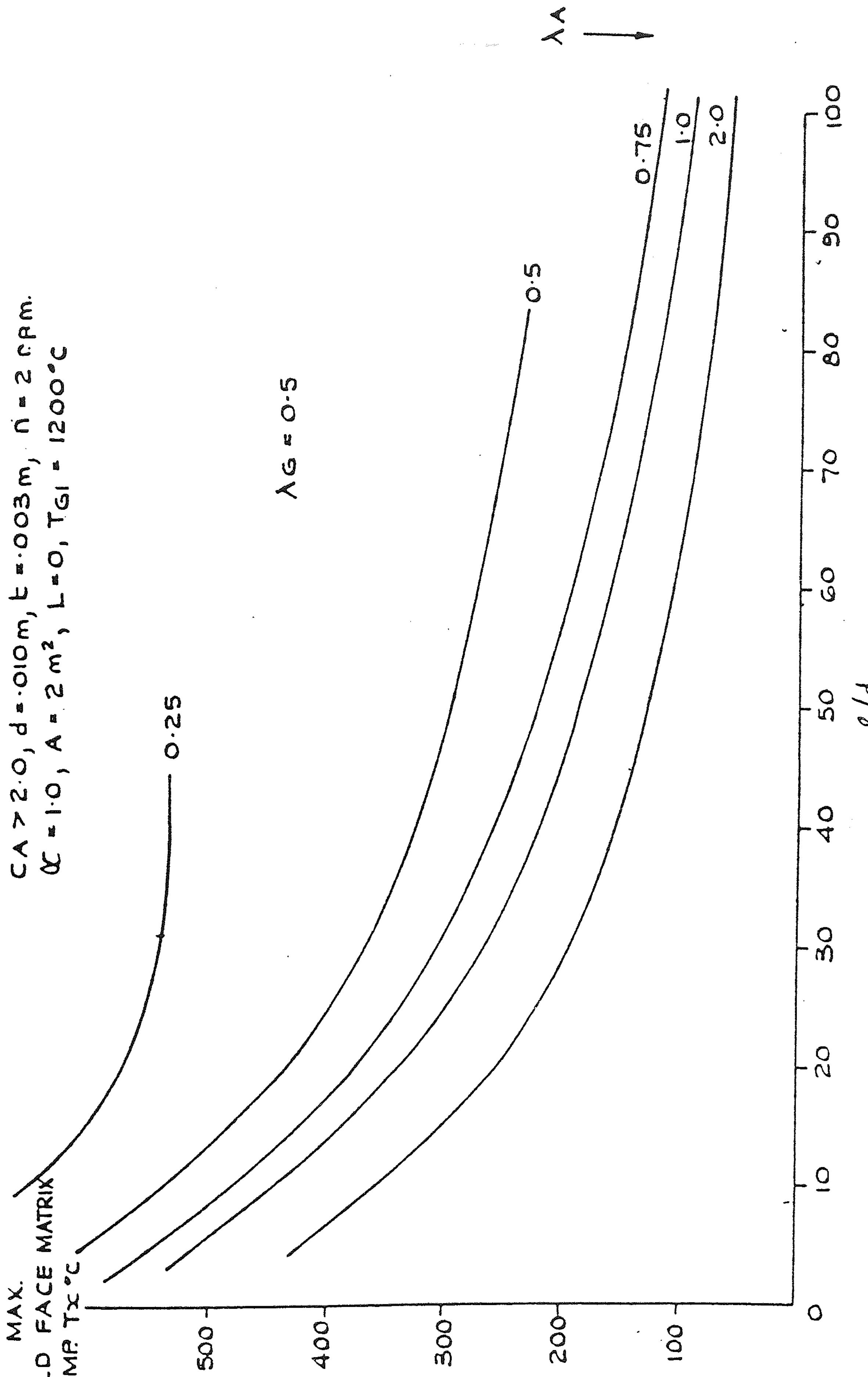
AIR SIDE PRESSURE LOSS VS ℓ/d. FOR VARIOUS λA

FIGURE 7.37



WASTE GAS PRESSURE LOSS Vs ℓ/d FOR VARIOUS λ_G

FIGURE 7.38



Tx VS l/d FOR VARIOUS VALUES OF λ_A AND λ_G = 0·5

FIGURE 7.39

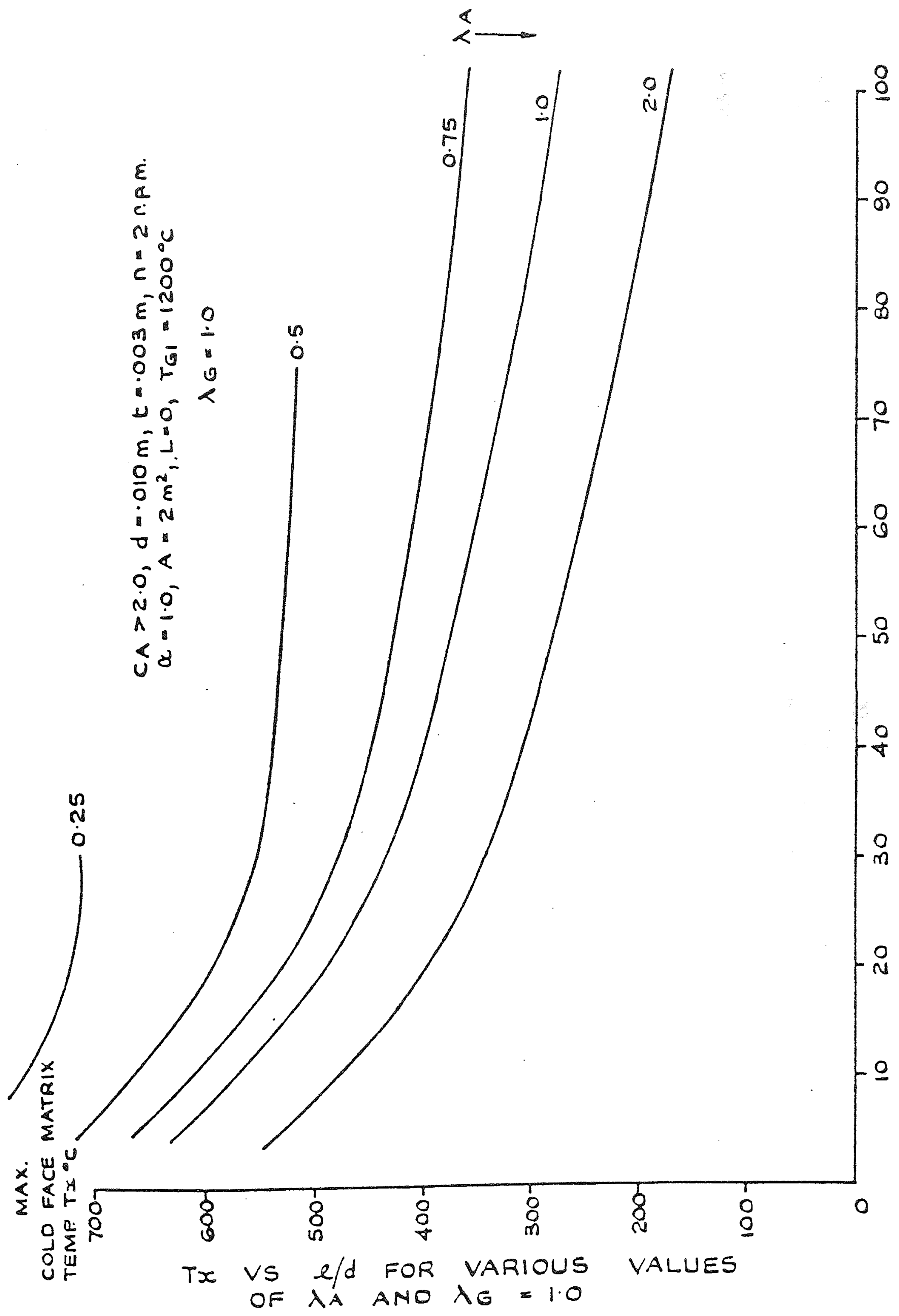
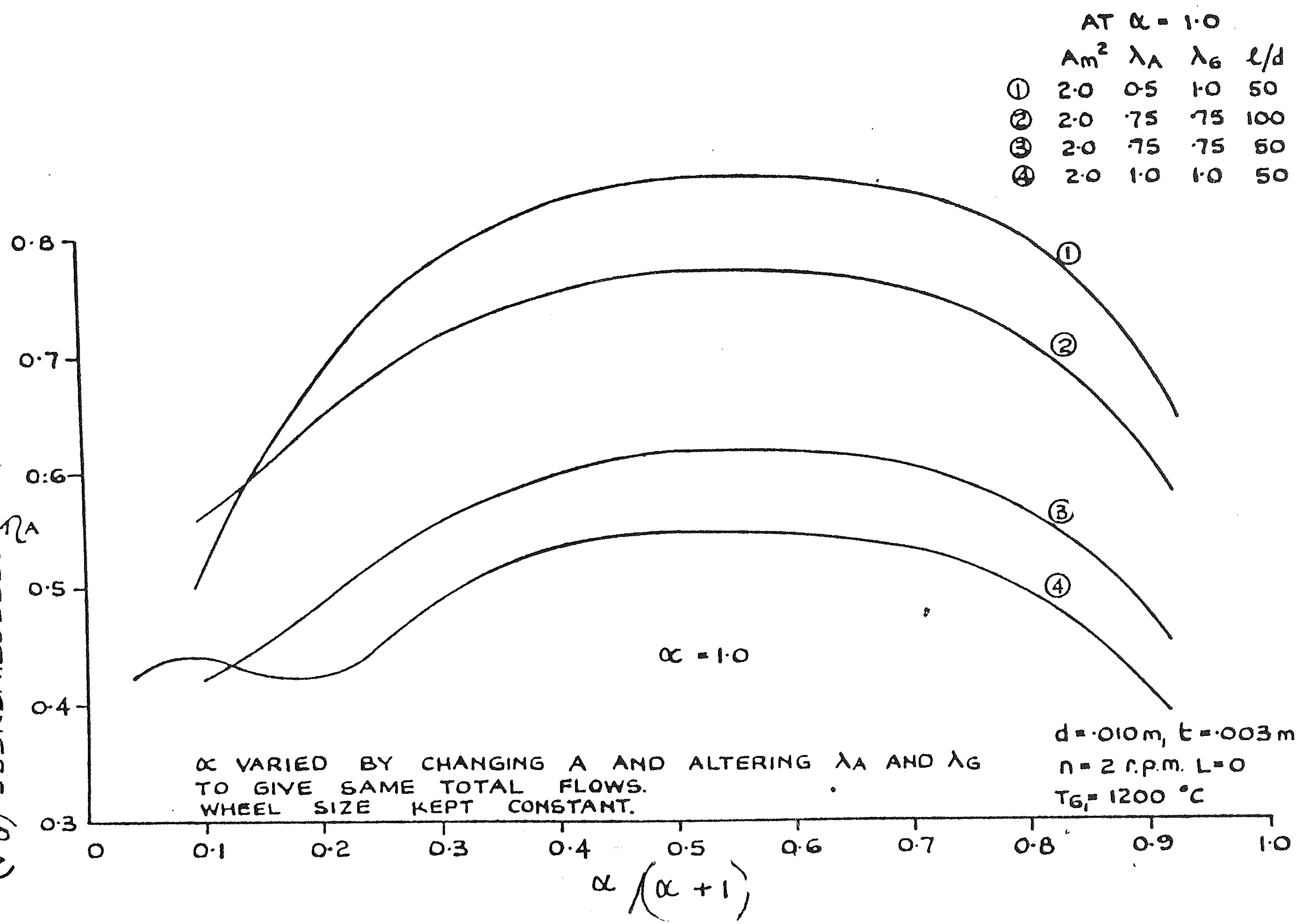
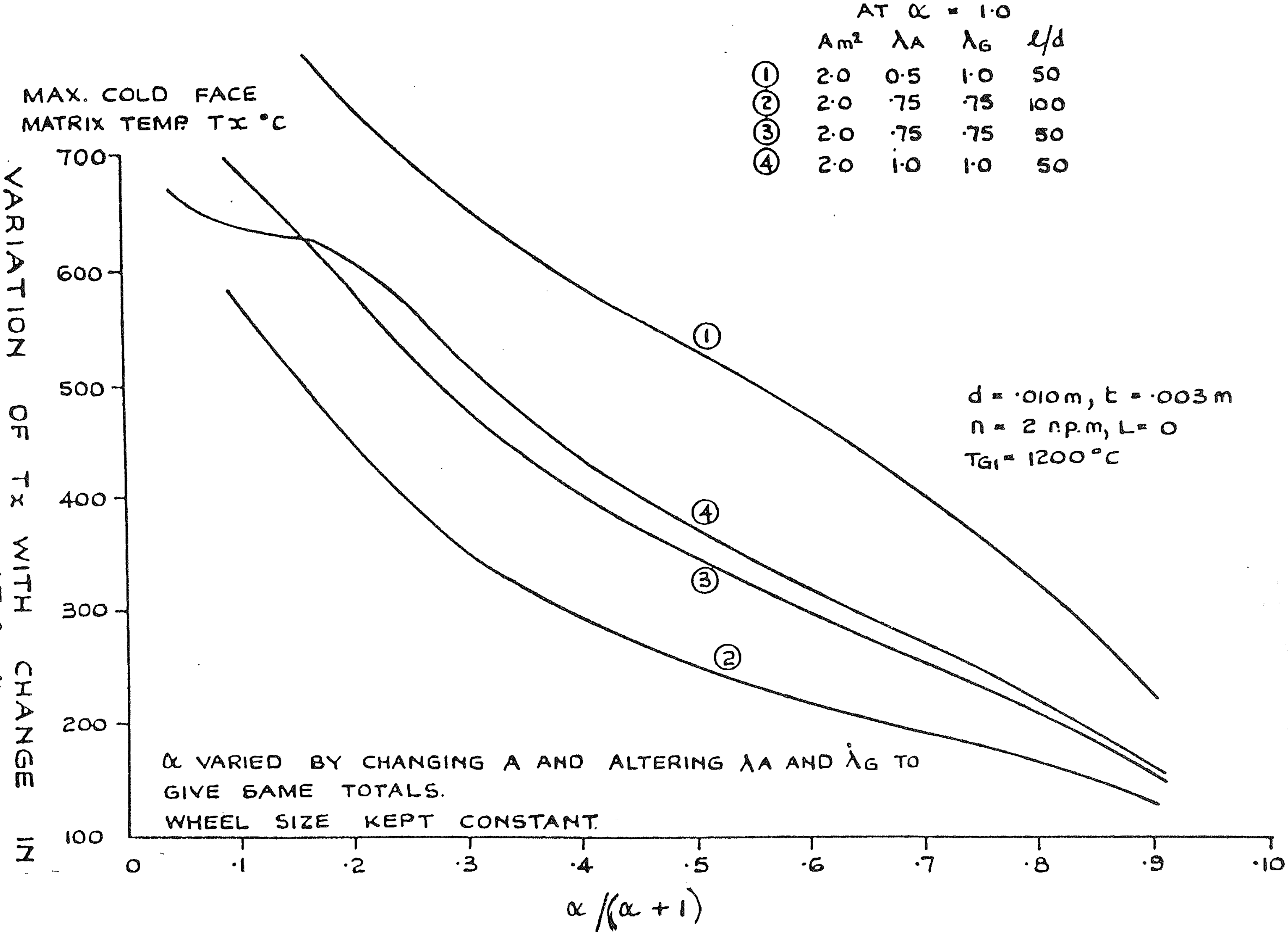

MAX. COLD FACE MATRIX TEMP Tx °C
CA > 2·0, d = ·010 m, t = ·003 m, n = 2 r.p.m.
α = 1·0, A = 2 m², L = 0, TGI = 1200°C
λG = 1·0
λA
0·25
0·5
0·75
1·0
2·0
700
600
500
400
300
200
100
0
10
20
30
40
50
60
70
80
90
100
Tx VS ℓ/d FOR VARIOUS VALUES
OF λA AND λG = 1·0

FIGURE 7.40



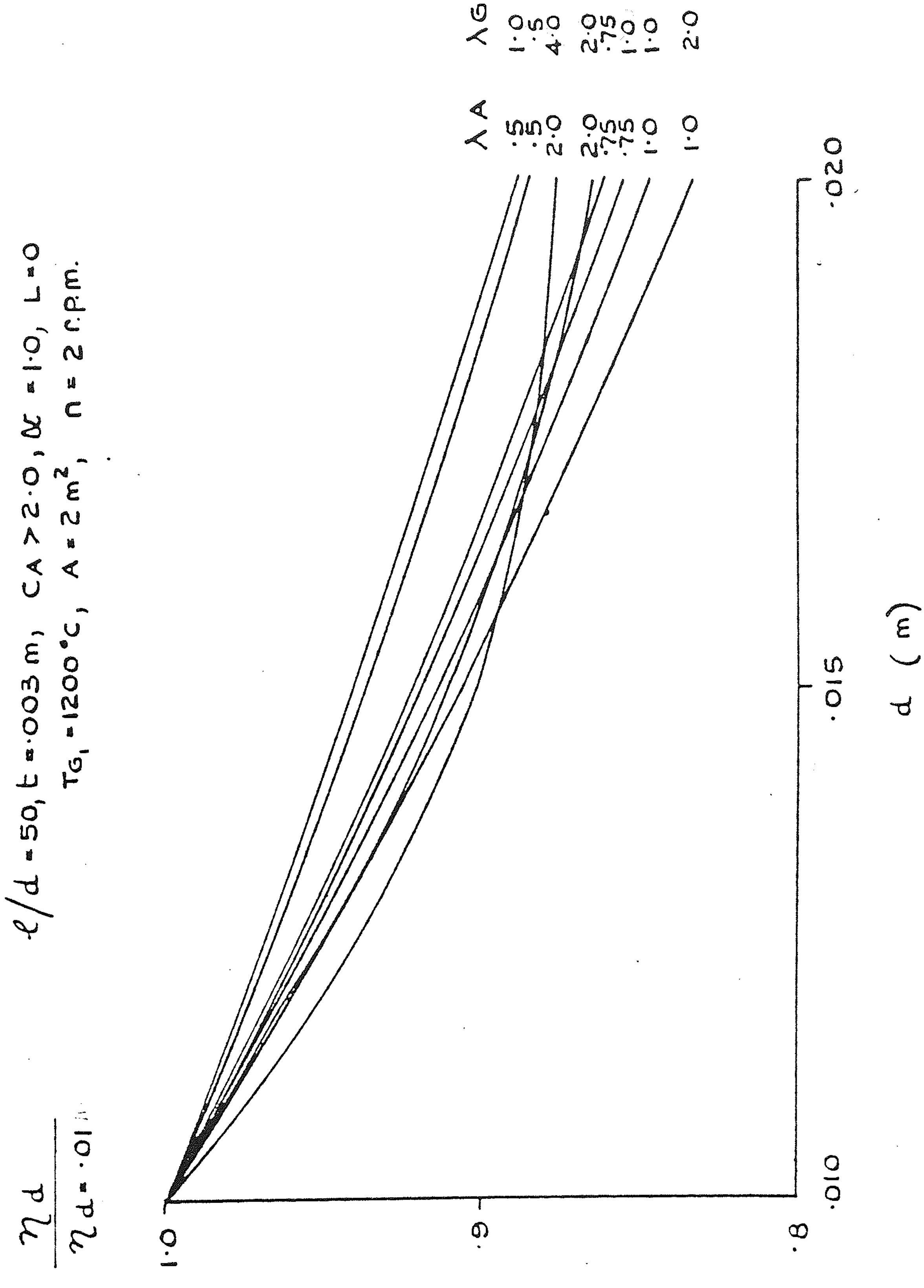
VARIATION OF AIR SIDE EFFECTIVENESS (η_A) WITH CHANGE IN THE PORT AREA RATIO (α

FIGURE 7.41



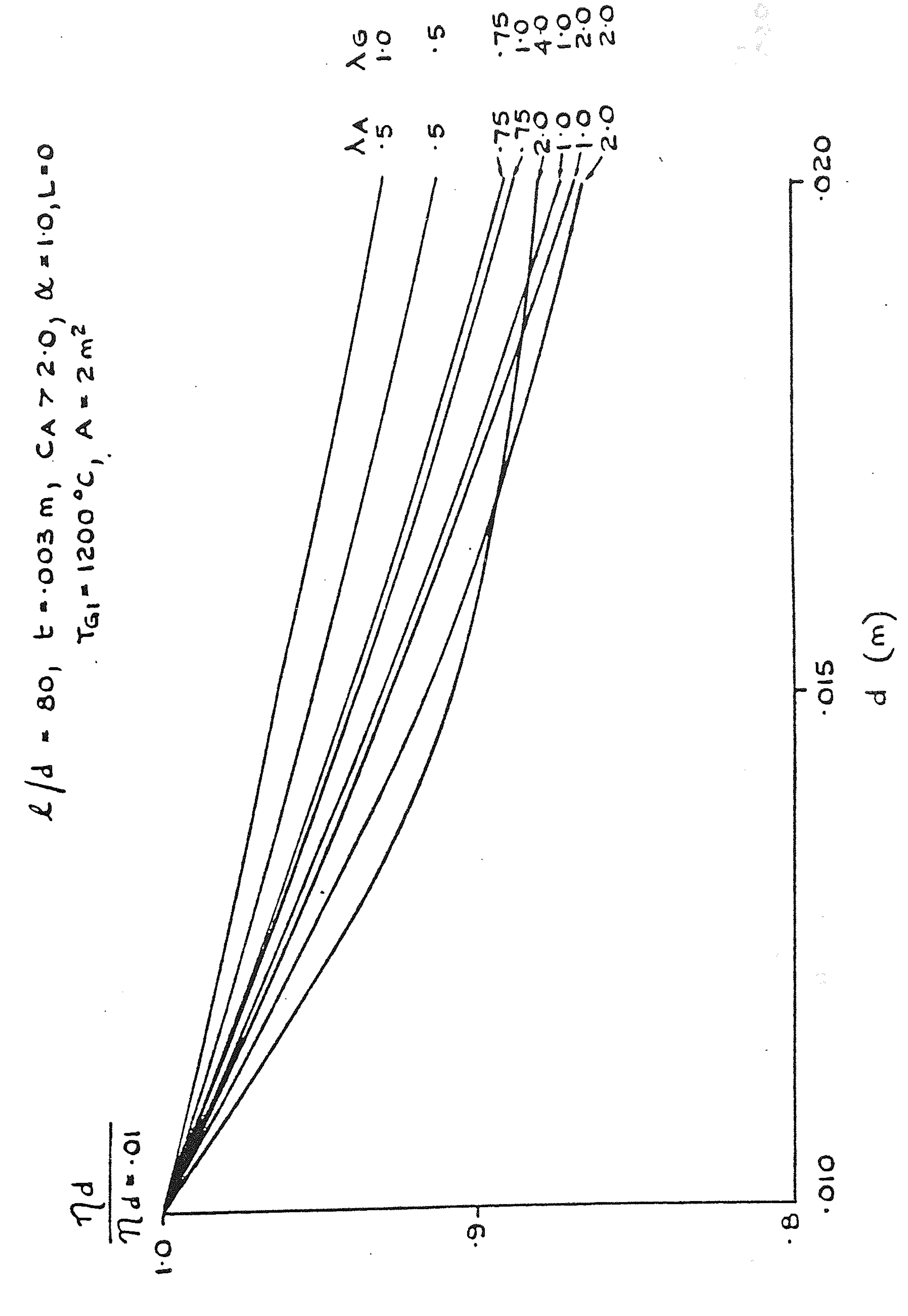
VARIATION OF Tx WITH CHANGE IN THE PORT AREA RATIO α

FIGURE 7.42



EFFECT OF HOLE SIZE ON
AIR SIDE EFFECTIVENESS

FIGURE 7.43



EFFECT OF HOLE SIZE ON
AIR SIDE EFFECTIVENESS

FIGURE 7.44

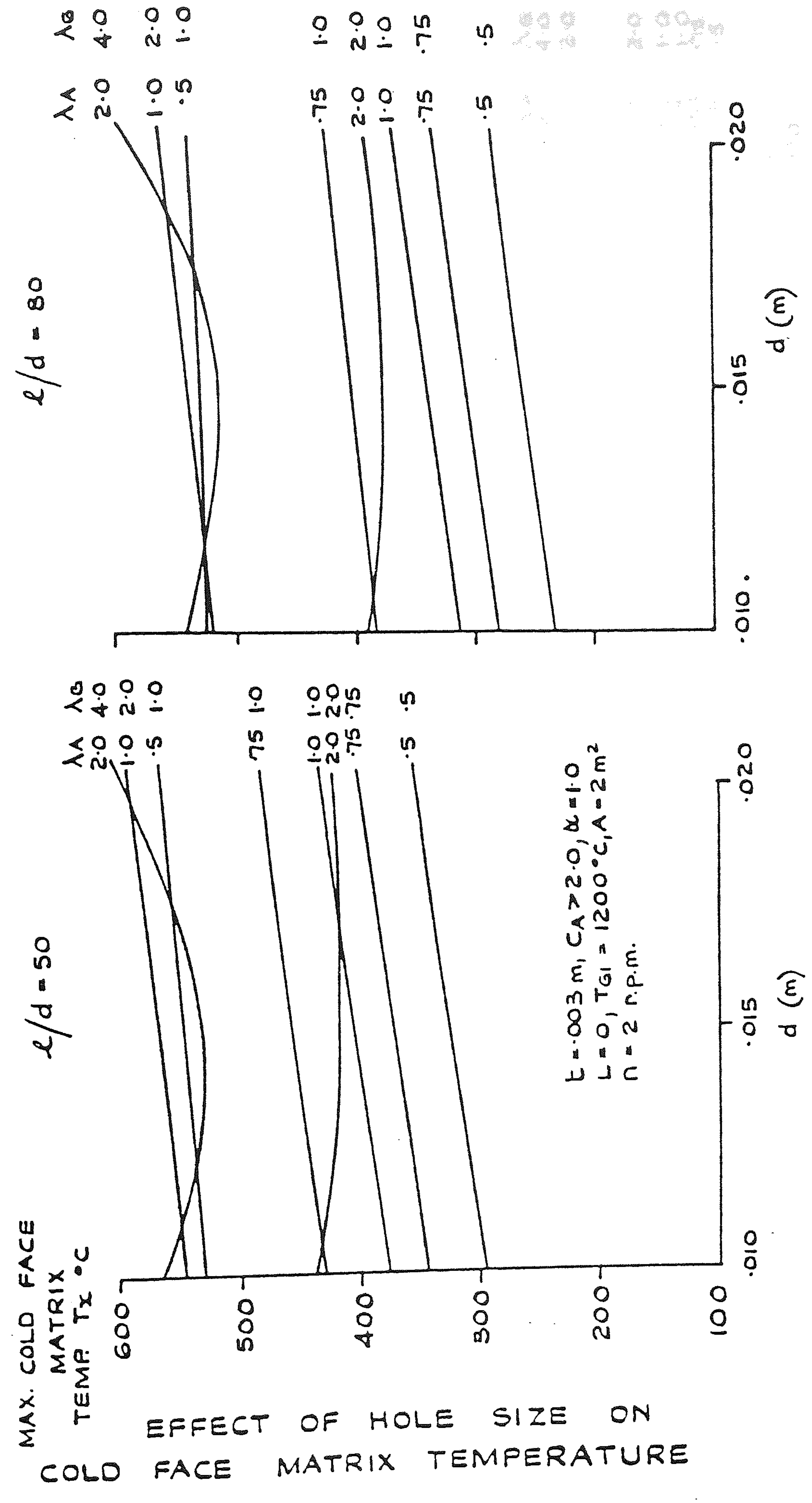
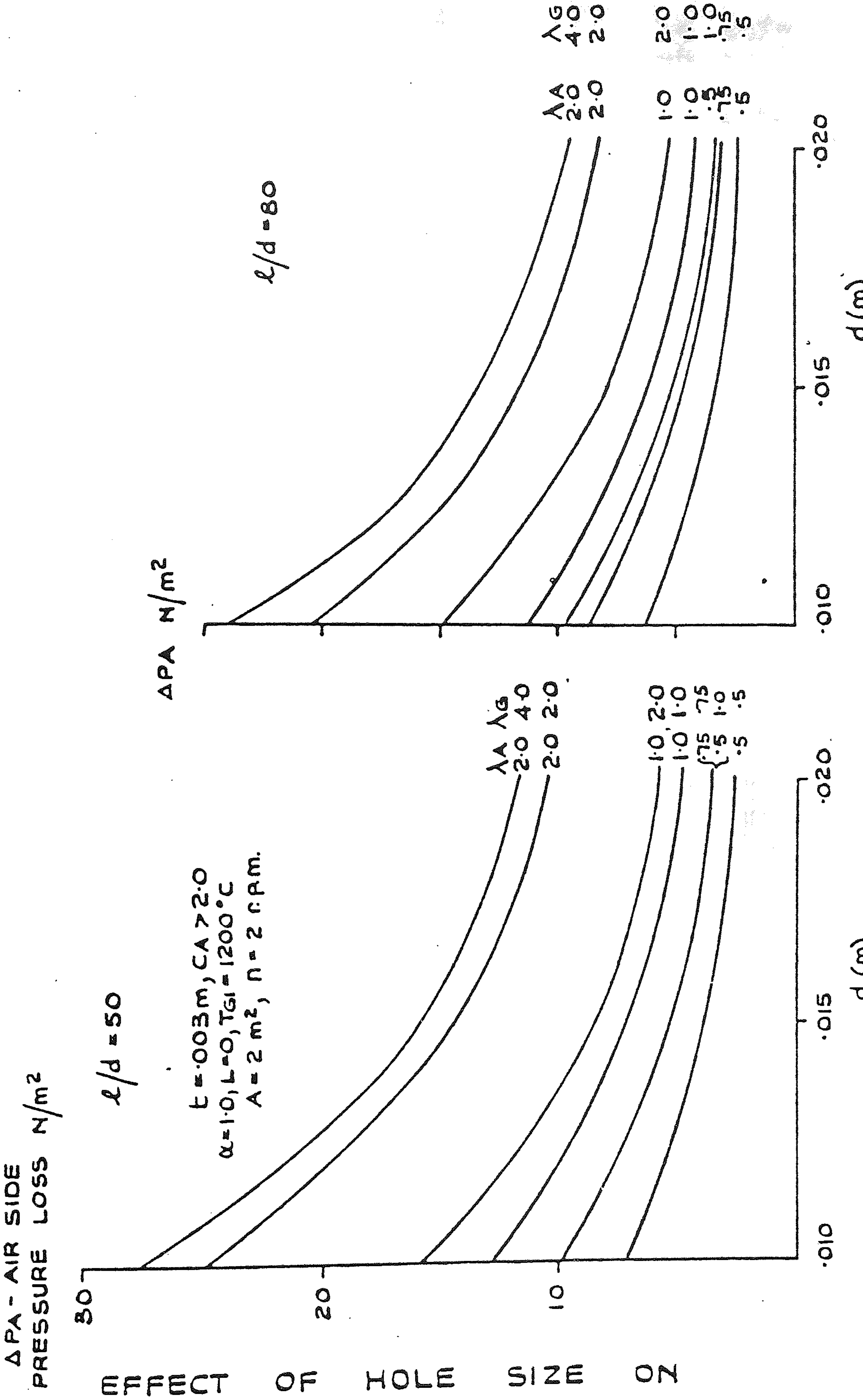
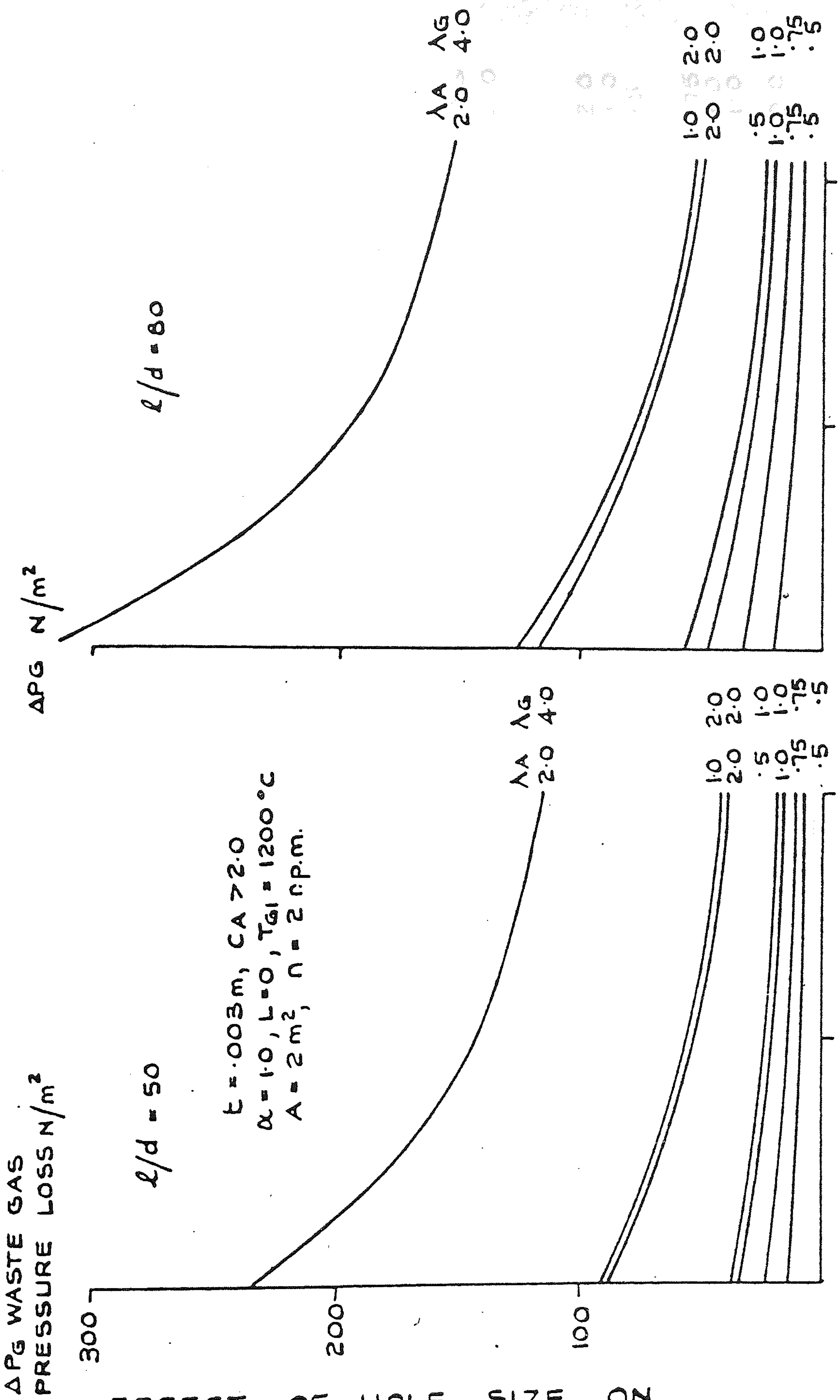

EFFECT OF HOLE SIZE ON COLD FACE MATRIX TEMPERATURE
MAX. COLD FACE MATRIX TEMP. Tx °C
600
500
400
300
200
100
ℓ/d = 50
ℓ/d = 80
λA λG
t = ·003 m, CA > 2·0, α = 1·0
L = 0, TGI = 1200°C, A = 2 m²
n = 2 r.p.m.
·010
·015
·020
d (m)

FIGURE 7.45



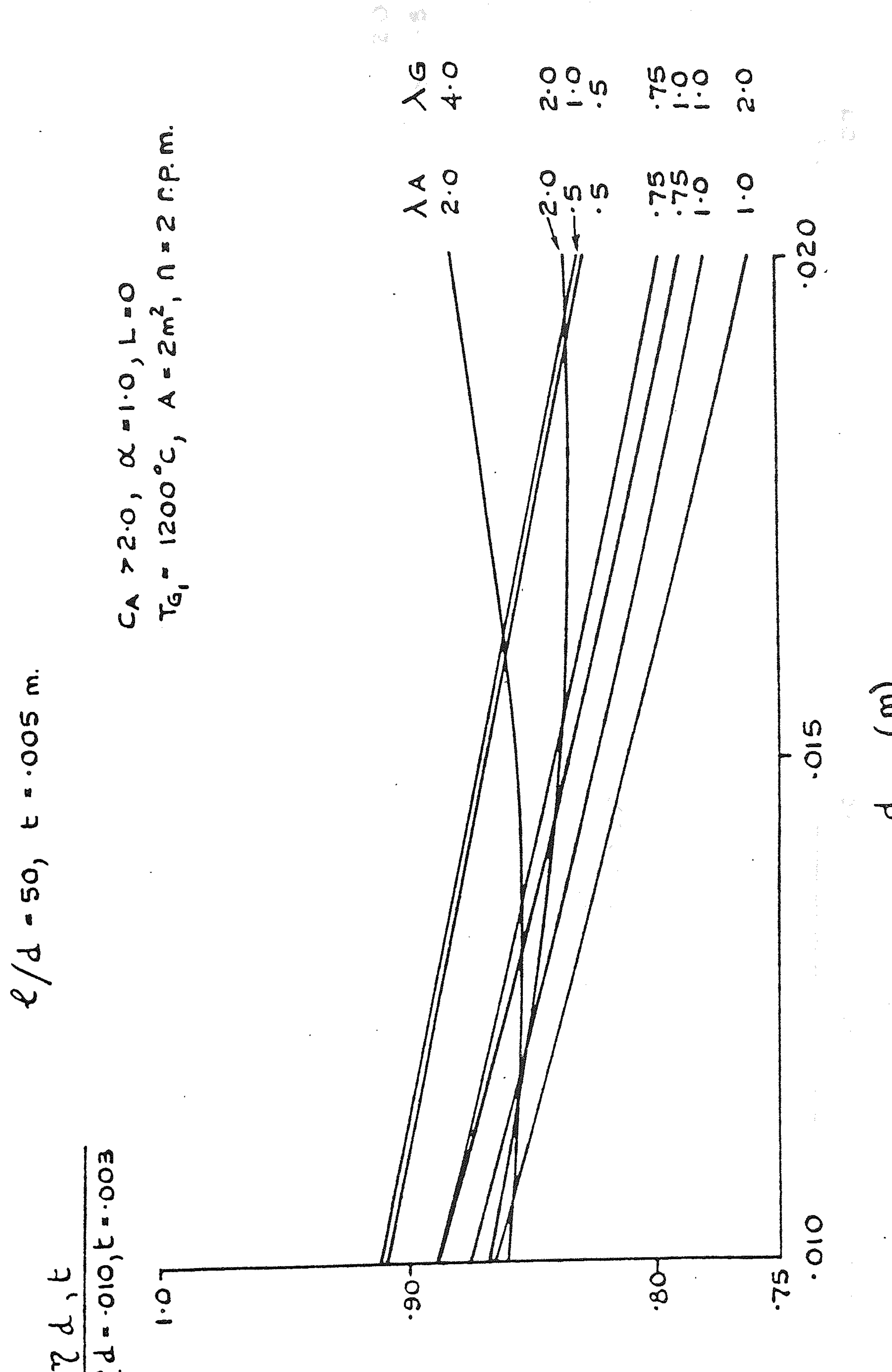
EFFECT OF HOLE SIZE ON AIR SIDE PRESSURE LOSS

FIGURE 7.46



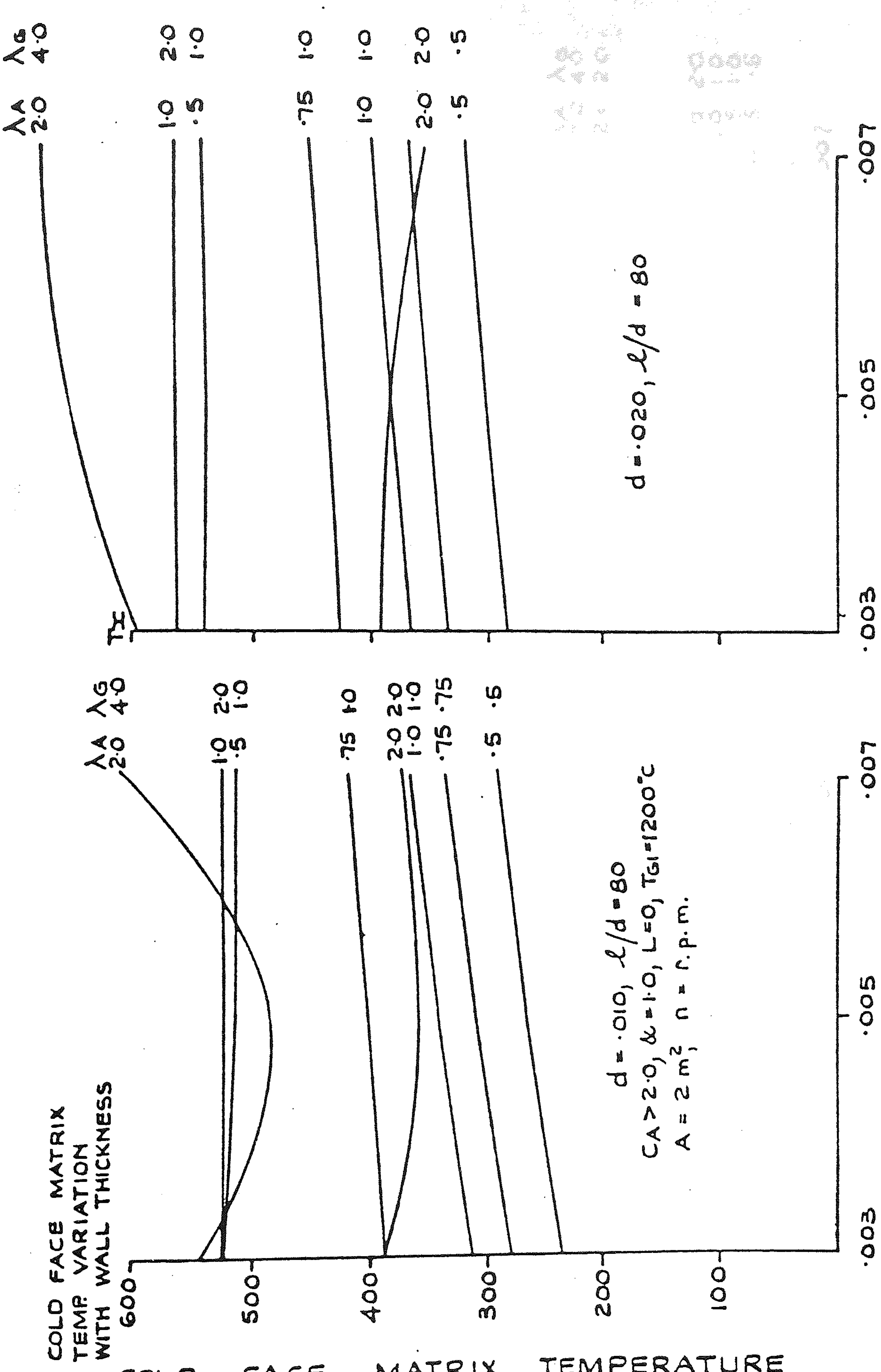
EFFECT OF HOLE SIZE ON WASTE GAS PRESSURE LOSS

FIGURE 7.47



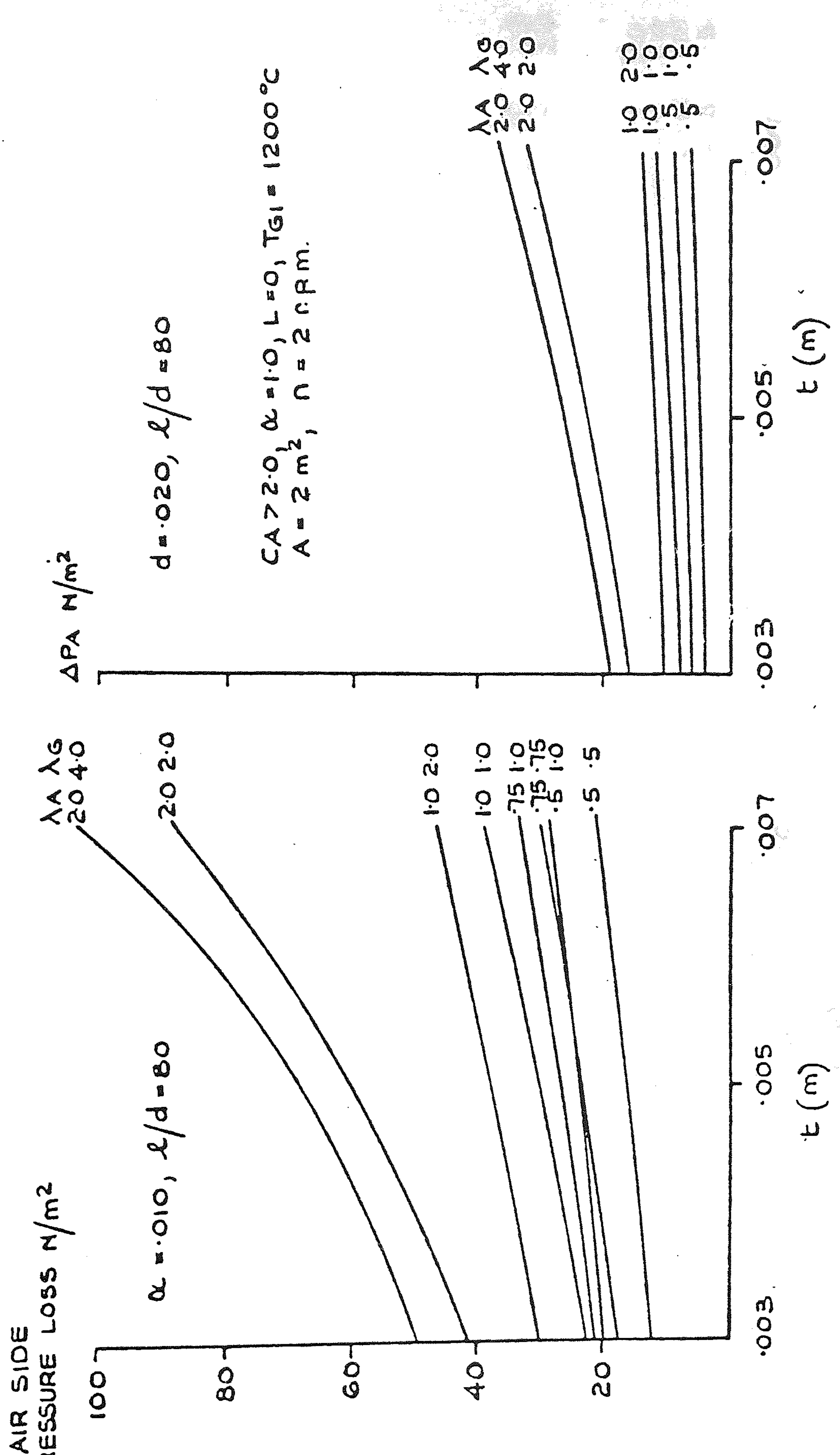
EFFECT OF HOLE SIZE AND WALL THICKNESS ON AIR SIDE EFFECTIVENESS

FIGURE 7.48



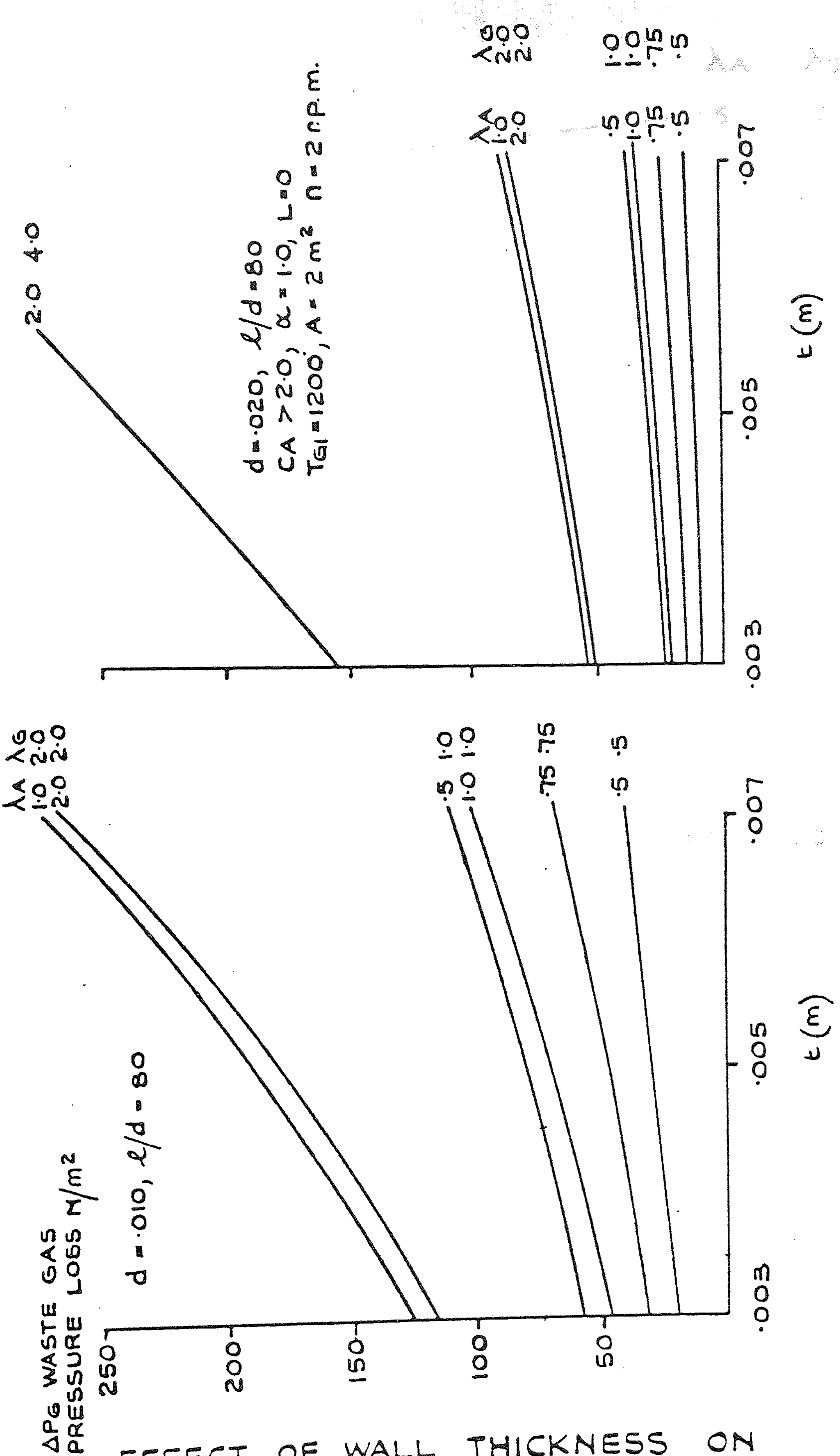
COLD FACE MATRIX TEMPERATURE VARIATION WITH WALL THICKNESS

FIGURE 7.49



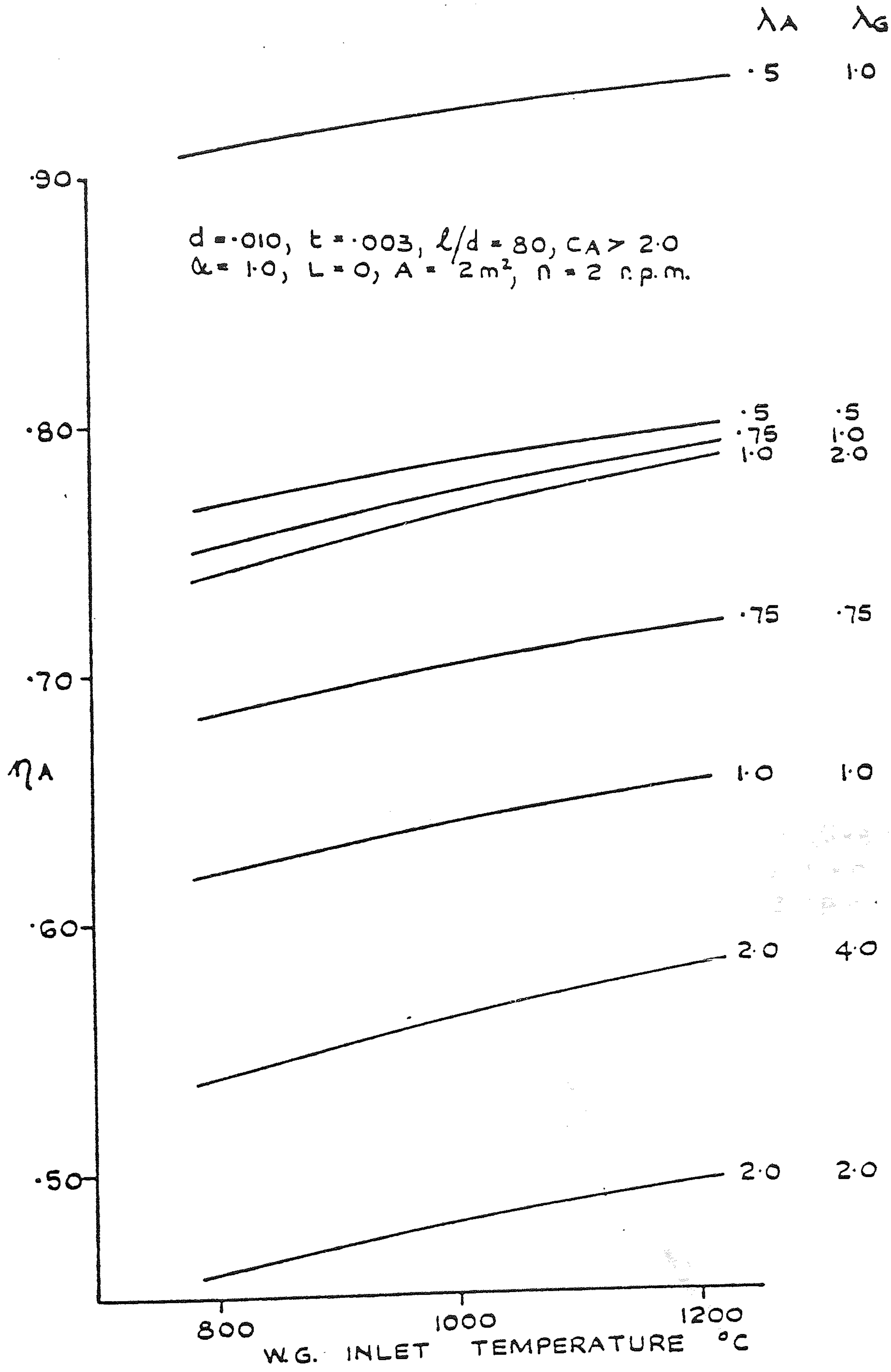
EFFECT OF WALL THICKNESS ON AIR SIDE PRESSURE LOSS

FIGURE 7.50



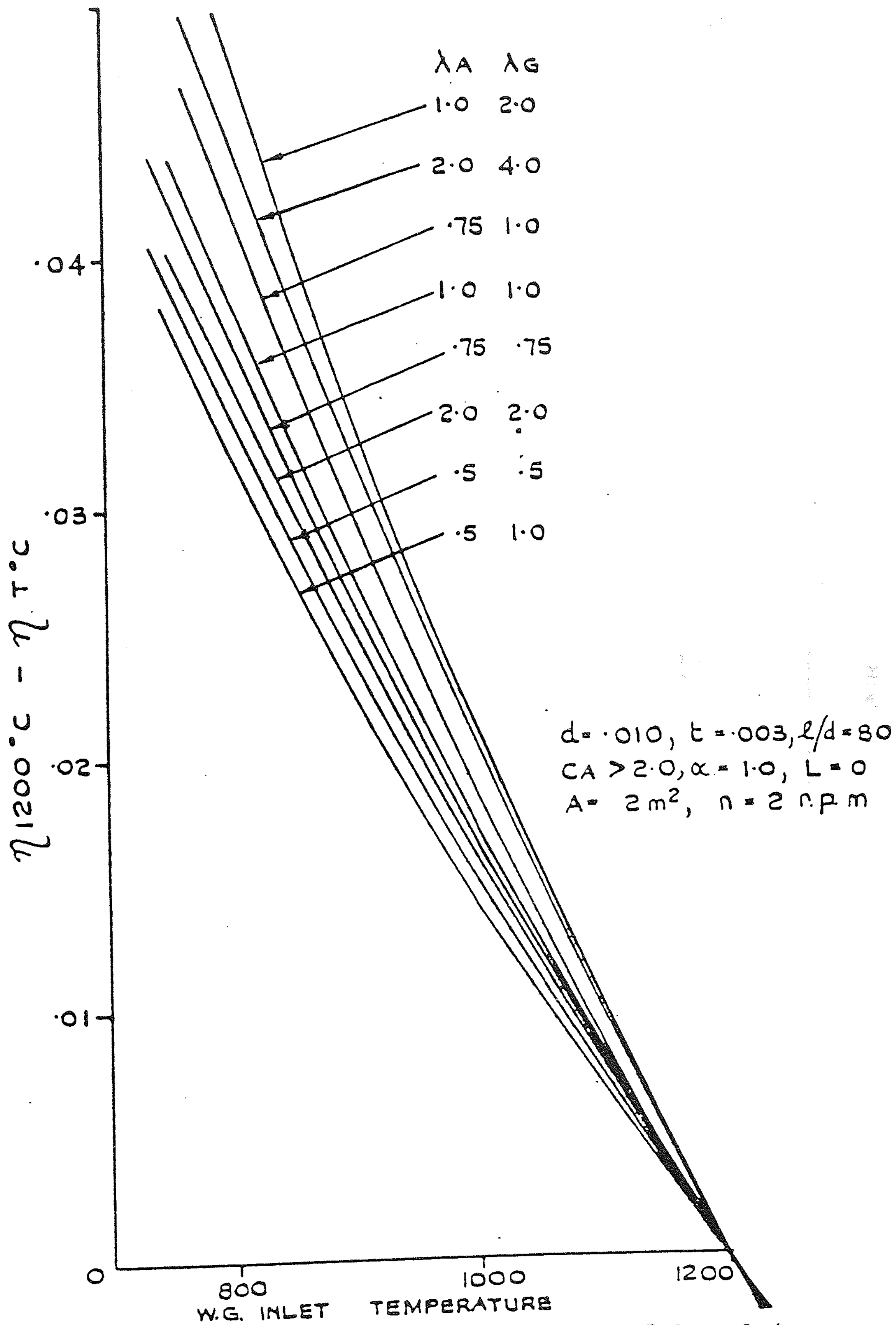
EFFECT OF WALL THICKNESS ON
WASTE GAS PRESSURE LOSS.

FIGURE 7.51



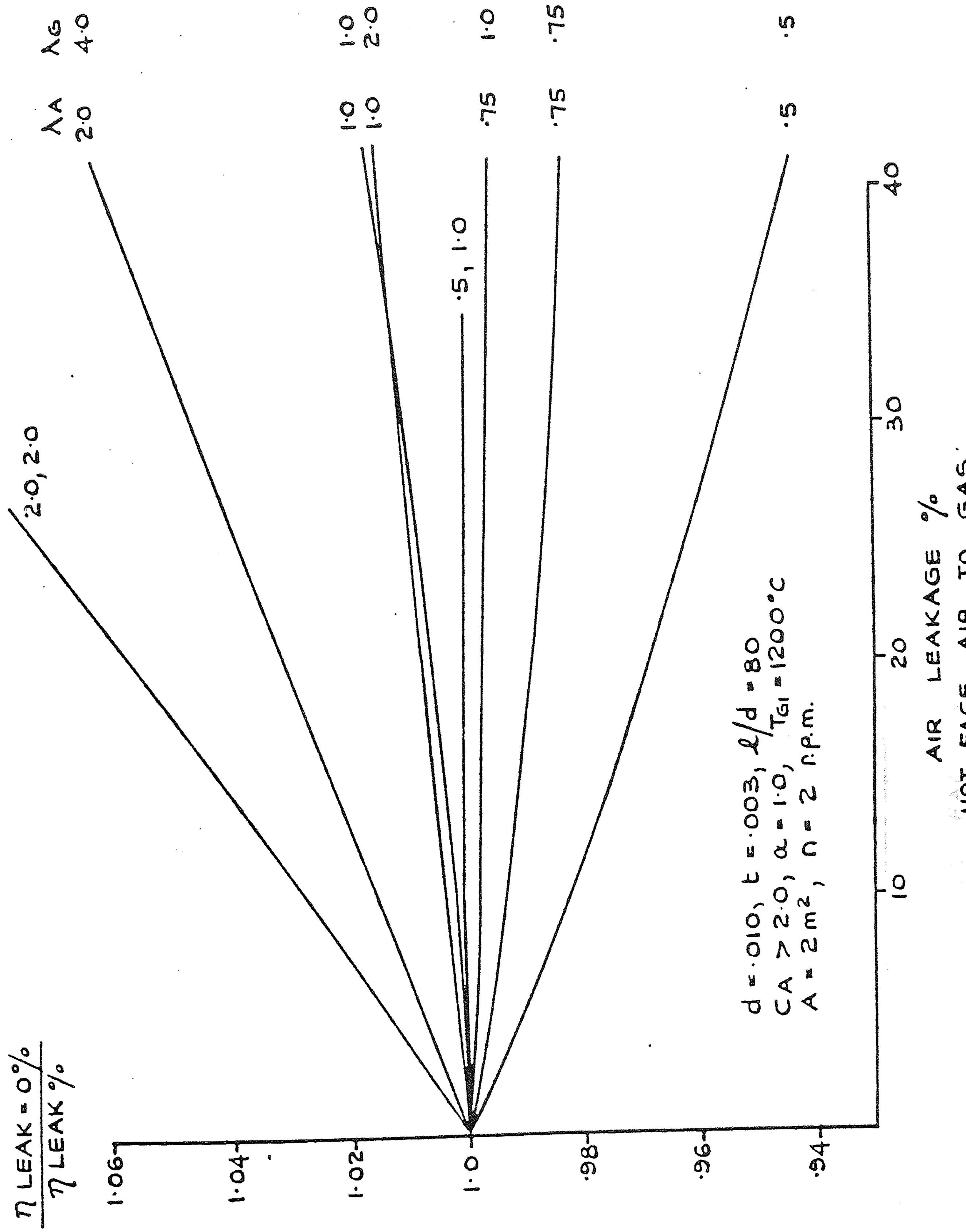
EFFECT OF W.G. INLET TEMPERATURE ON AIR SIDE EFFECTIVENESS

FIGURE 7.52



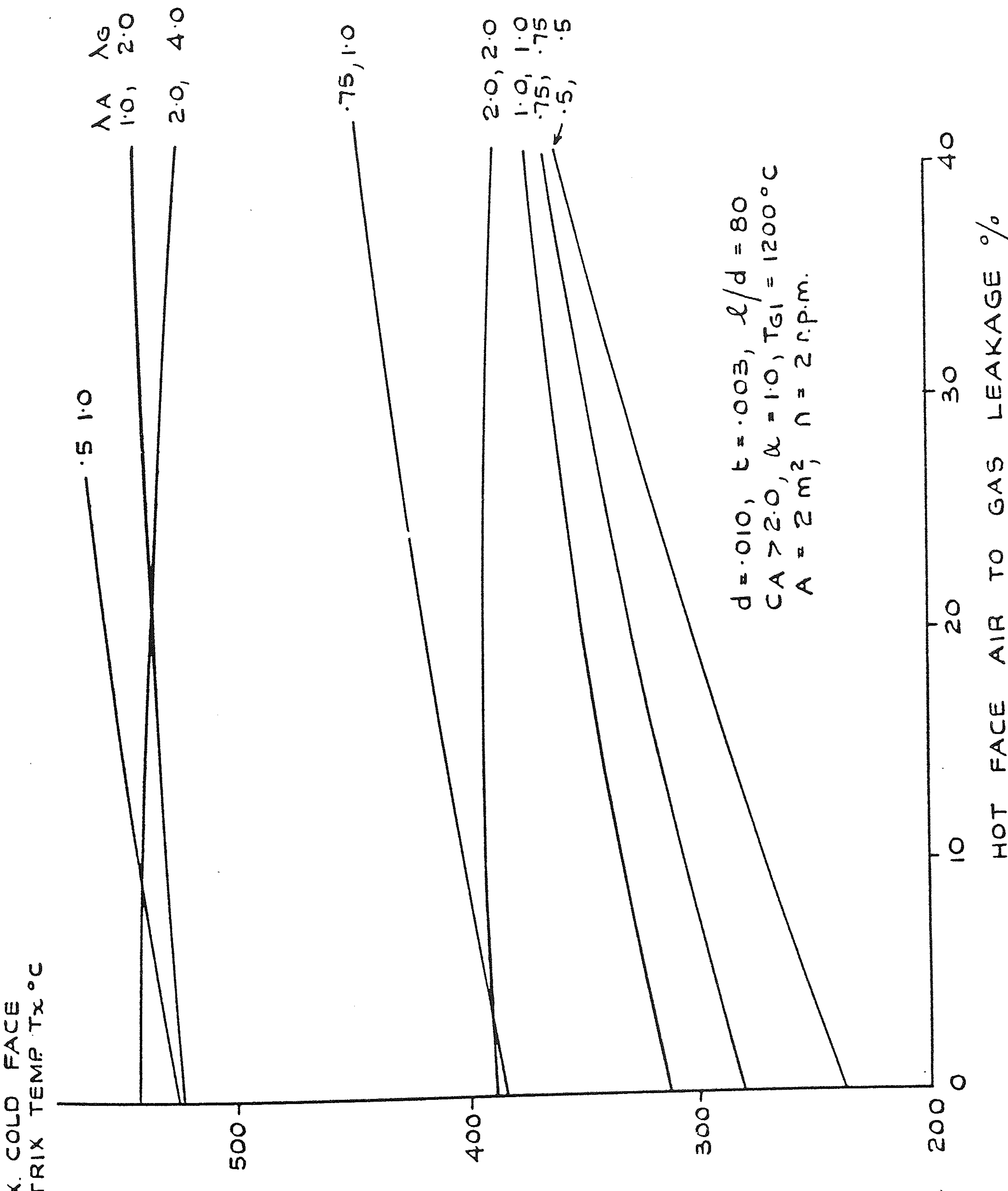
EFFECT OF W.G. INLET TEMPERATURE ON AIR SIDE EFFECTIVENESS

FIGURE 7.53



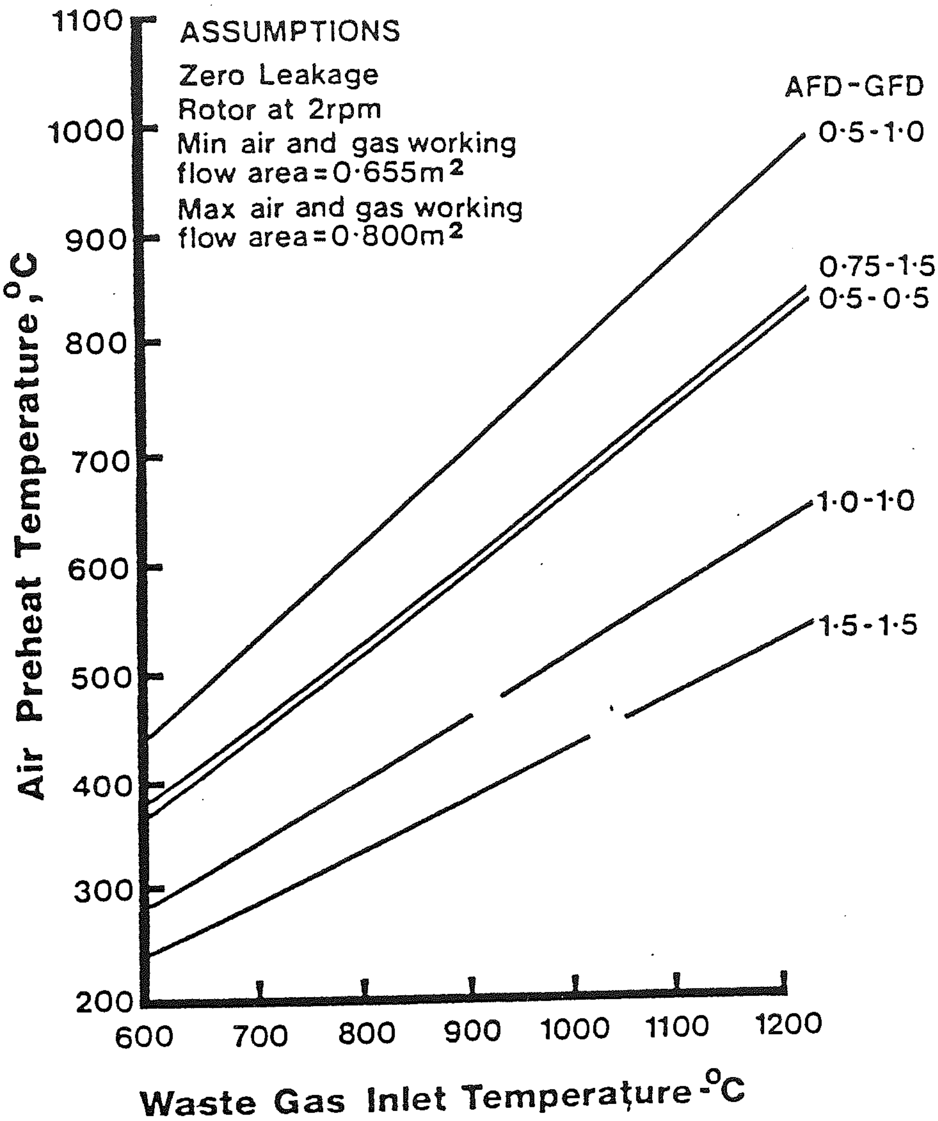
EFFECT OF AIR LEAKAGE ON AIR SIDE EFFECTIVENESS. a) HOT FACE AIR TO GAS.

FIGURE 7.54



EFFECT OF AIR LEAKAGE ON COLD FACE MATRIX TEMPERATURE (HOT FACE AIR TO GAS LEAKAGE)

FIGURE 7.55



FLUX DENSITY ($kg/s.m^2$)	MIN AREA FLOW (scfm)	MAX AREA FLOW (scfm)
0·5	54	66
0·75	80	98
1·0	107	131
1·5	161	197

Rotary Regenerator Rig Predicted Performance for Various Air and Gas Flux Densities

FIGURE 7.56

Aston University

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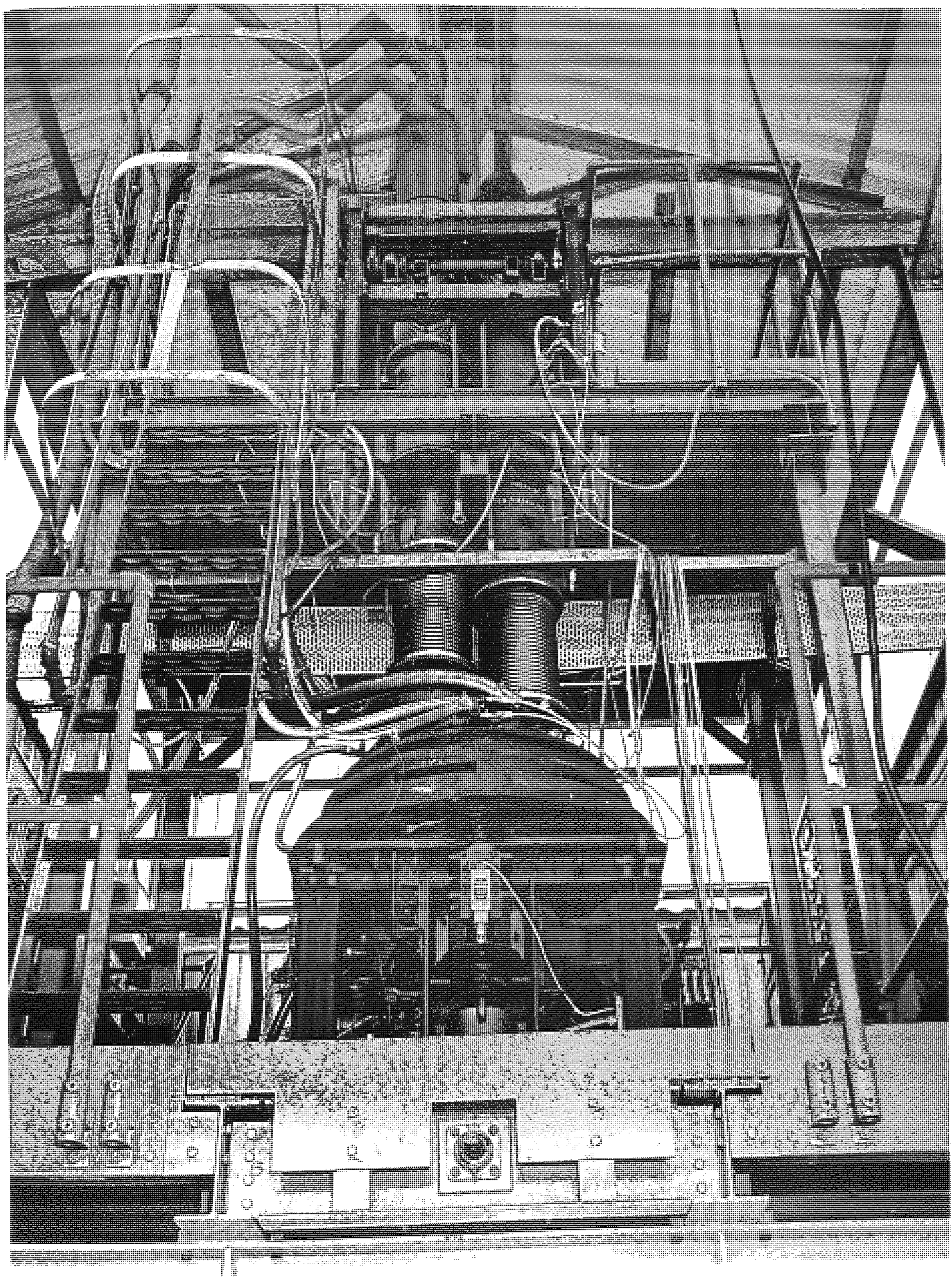
SCHEMATIC LAYOUT OF SYSTEMS AND INSTRUMENTATION

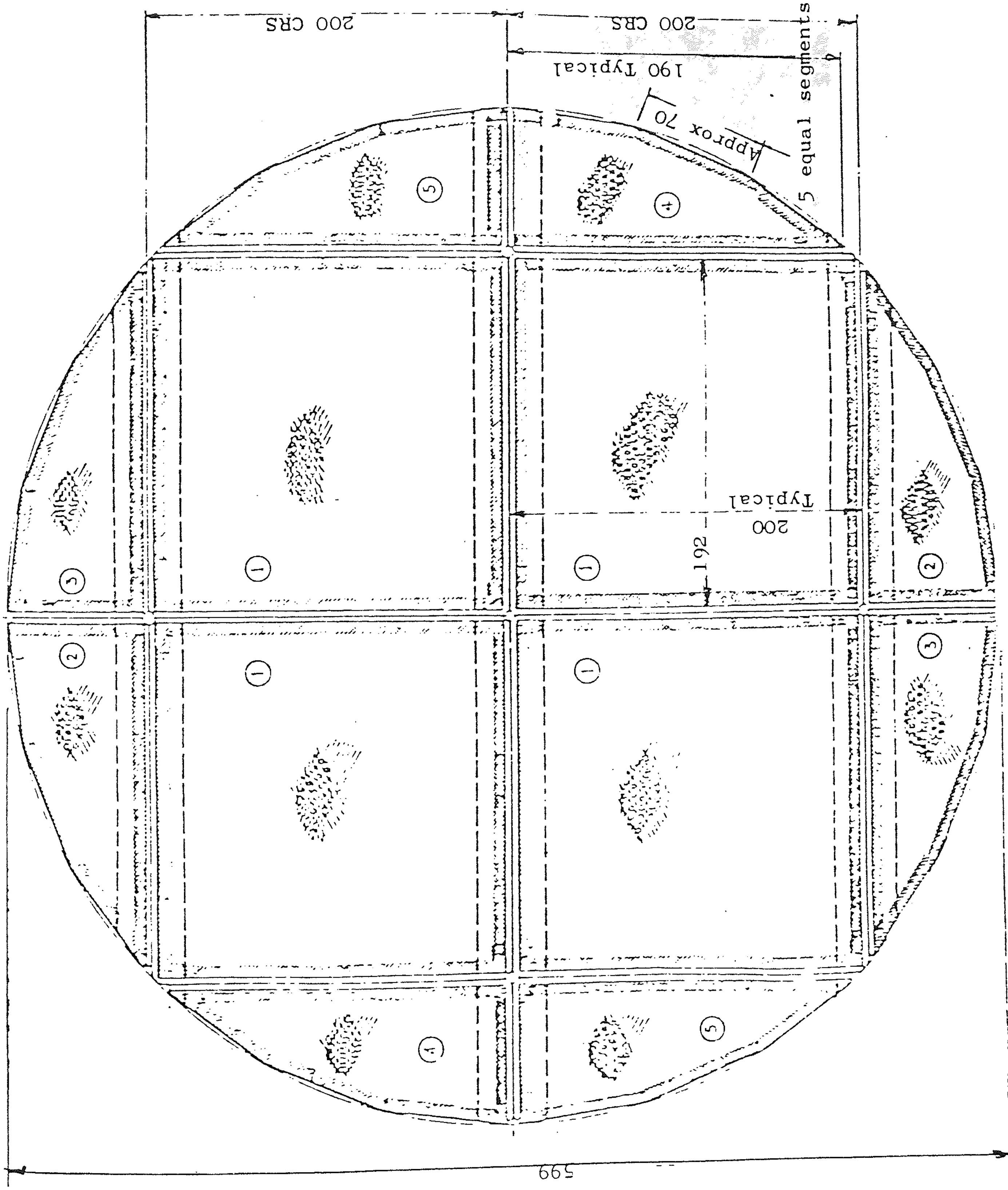
SCHEMATIC LAYOUT OF CRR PILOT PLANT

FIGURE 7.57



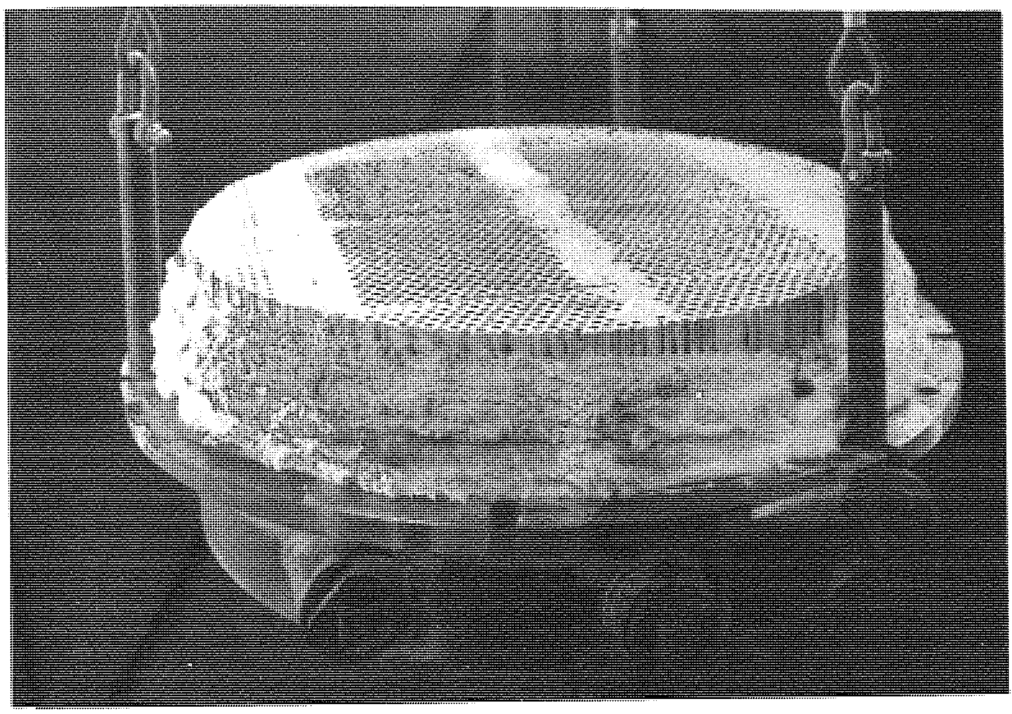
PILOT PLANT ASSEMBLY

FIGURE 7.58



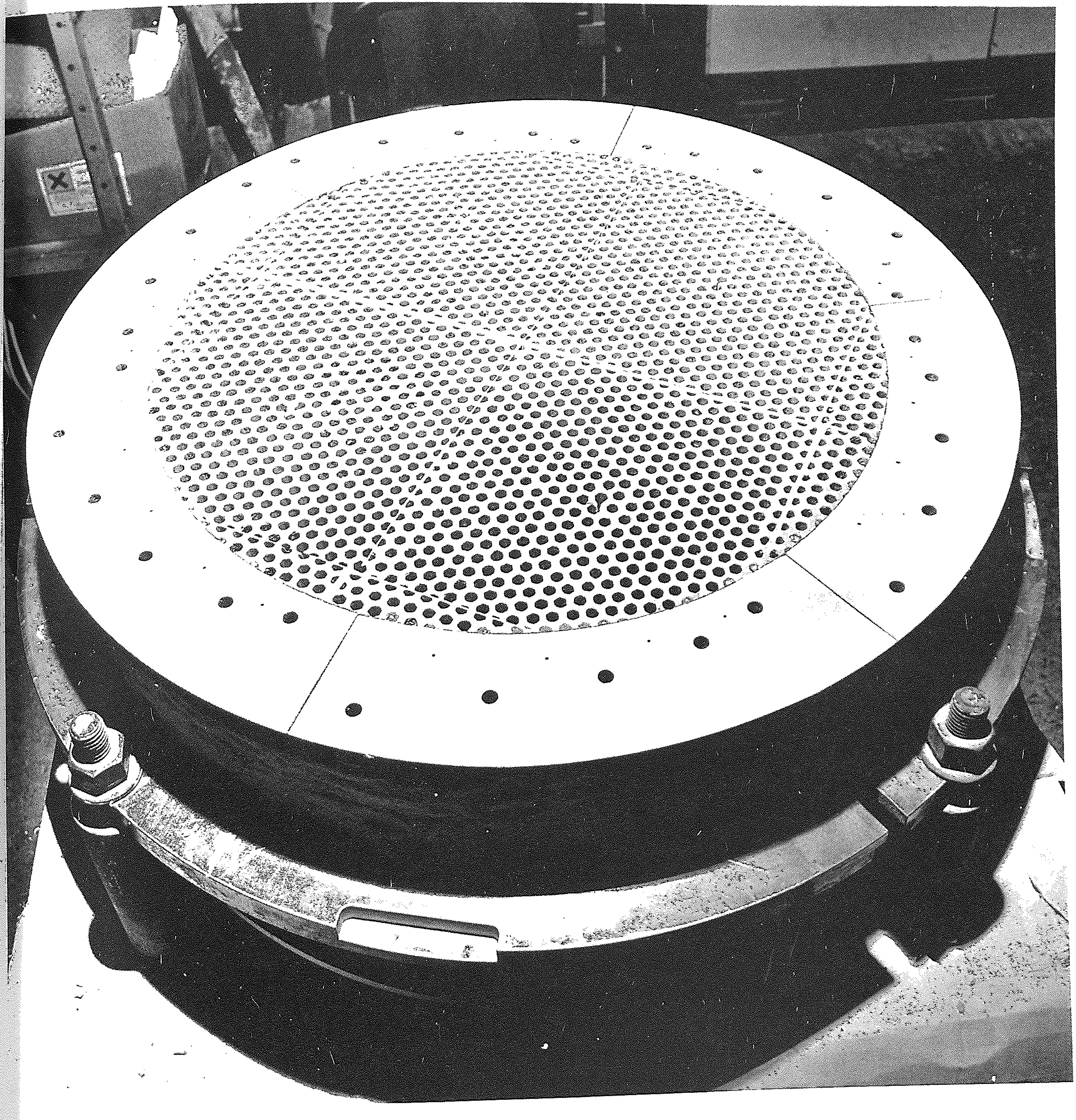
ASSEMBLY OF MATRIX BLOCKS

FIGURE 7.59



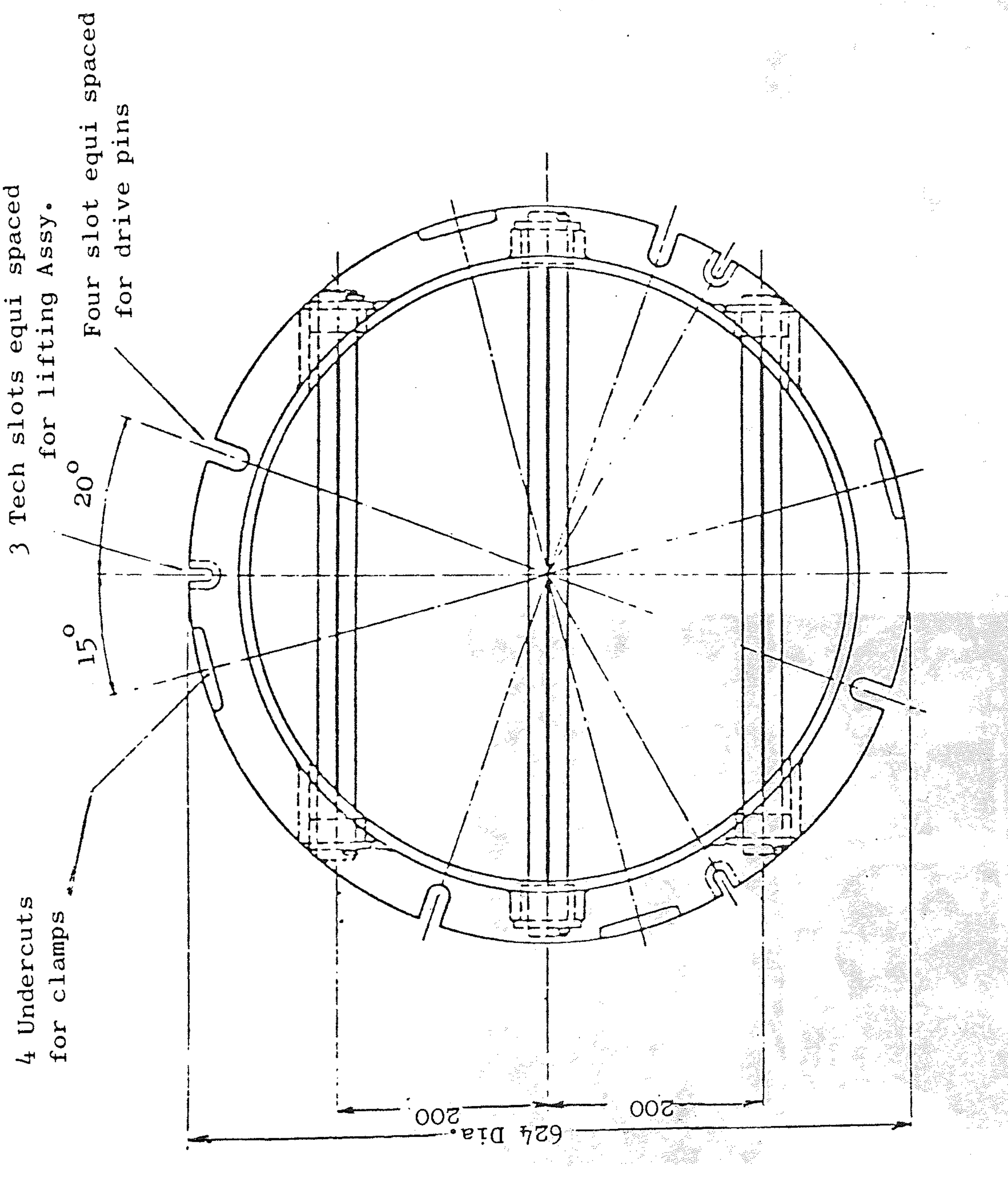
PICTORIAL VIEW
Ceramic Matrix

FIGURE 7.60



MATRIX WITH RING SEGMENTS

FIGURE 7.61

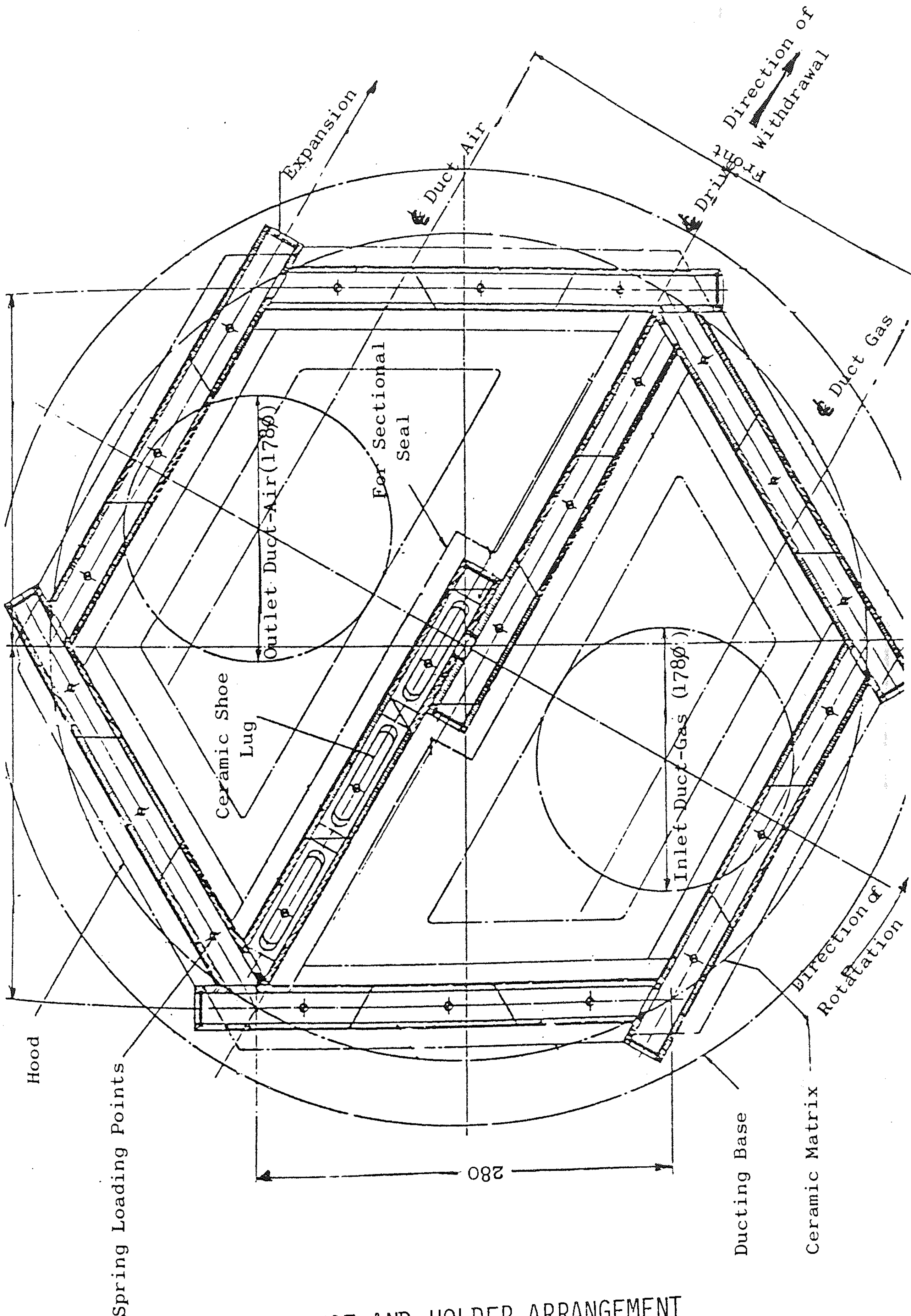


BASKET ASSEMBLY COMPLETE WITH MATRIX BLOCK

FIGURE 7.62

DRIVE MECHANISM

FIGURE 7.63



CERAMIC SHOE & HOLDER ARRANGEMENT

CERAMIC SHOE AND HOLDER ARRANGEMENT

FIGURE 7.64

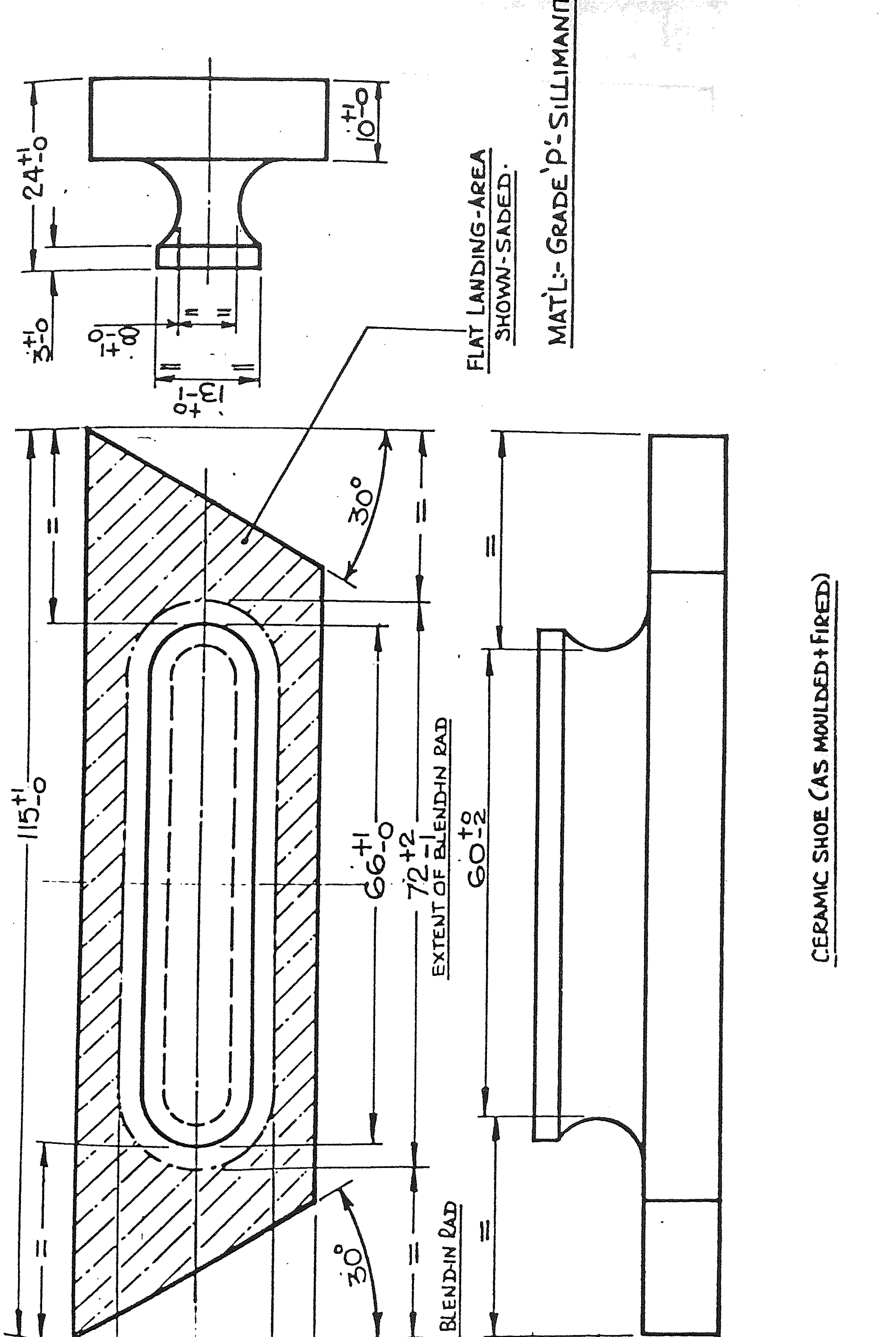
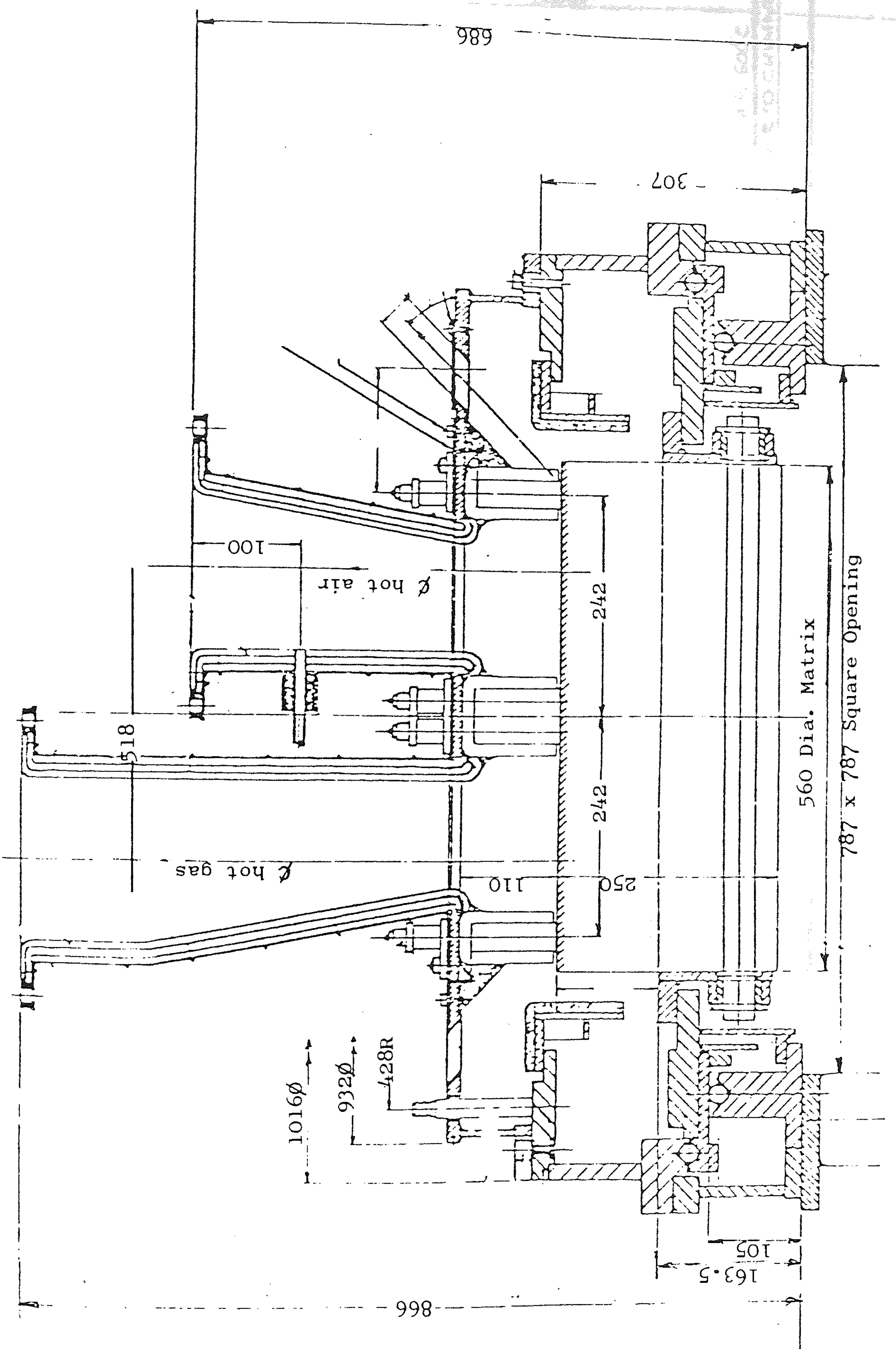
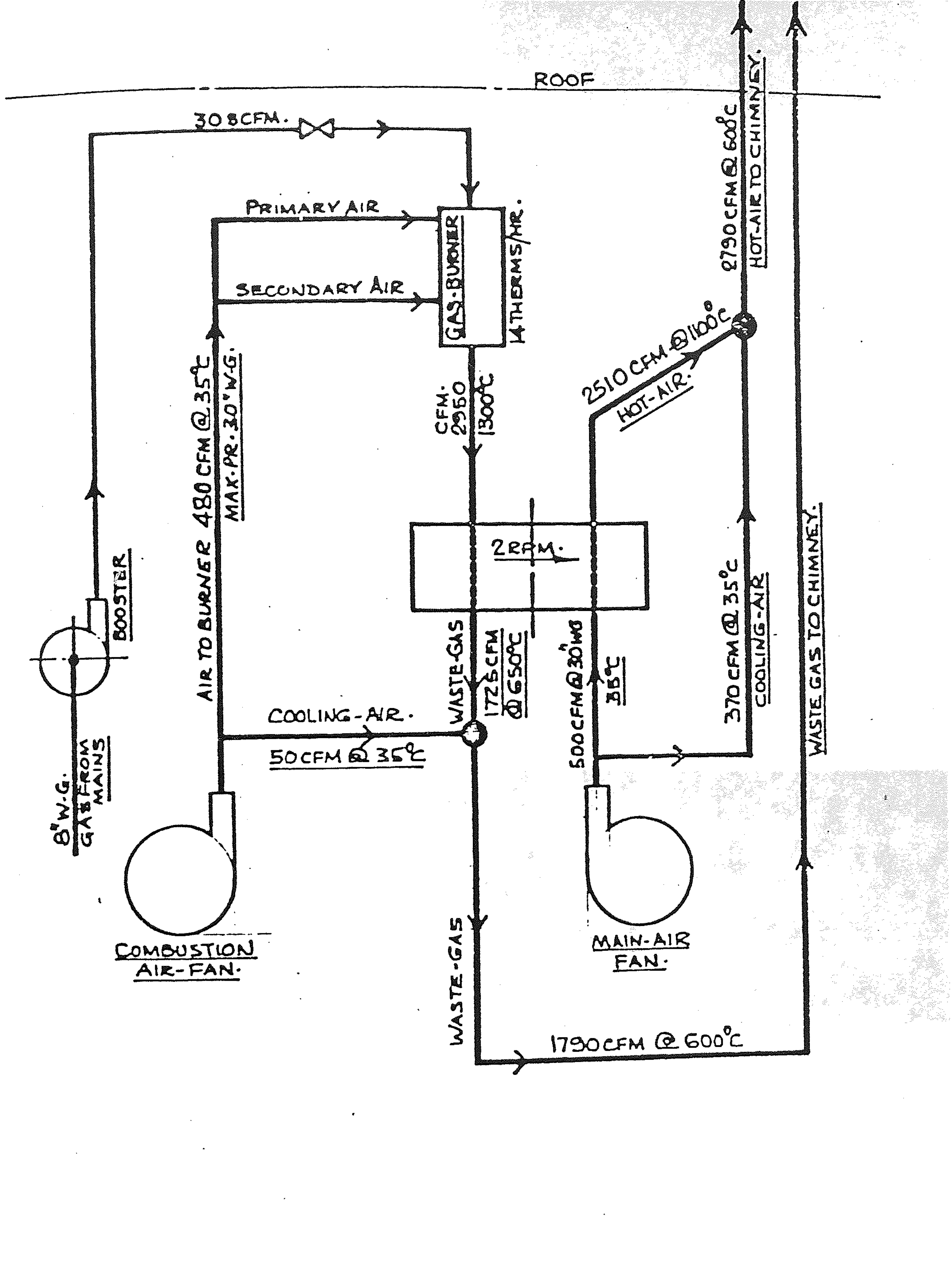

FLAT LANDING-AREA
SHOWN-SADED.
MAT'L:- GRADE 'P'-SILLIMANIT
CERAMIC SHOE (AS MOULDED + FIRED)
EXTENT OF BLEND-IN RAD
EXTENT OF BLEND-IN RAD
30°
30°

FIGURE 7.65



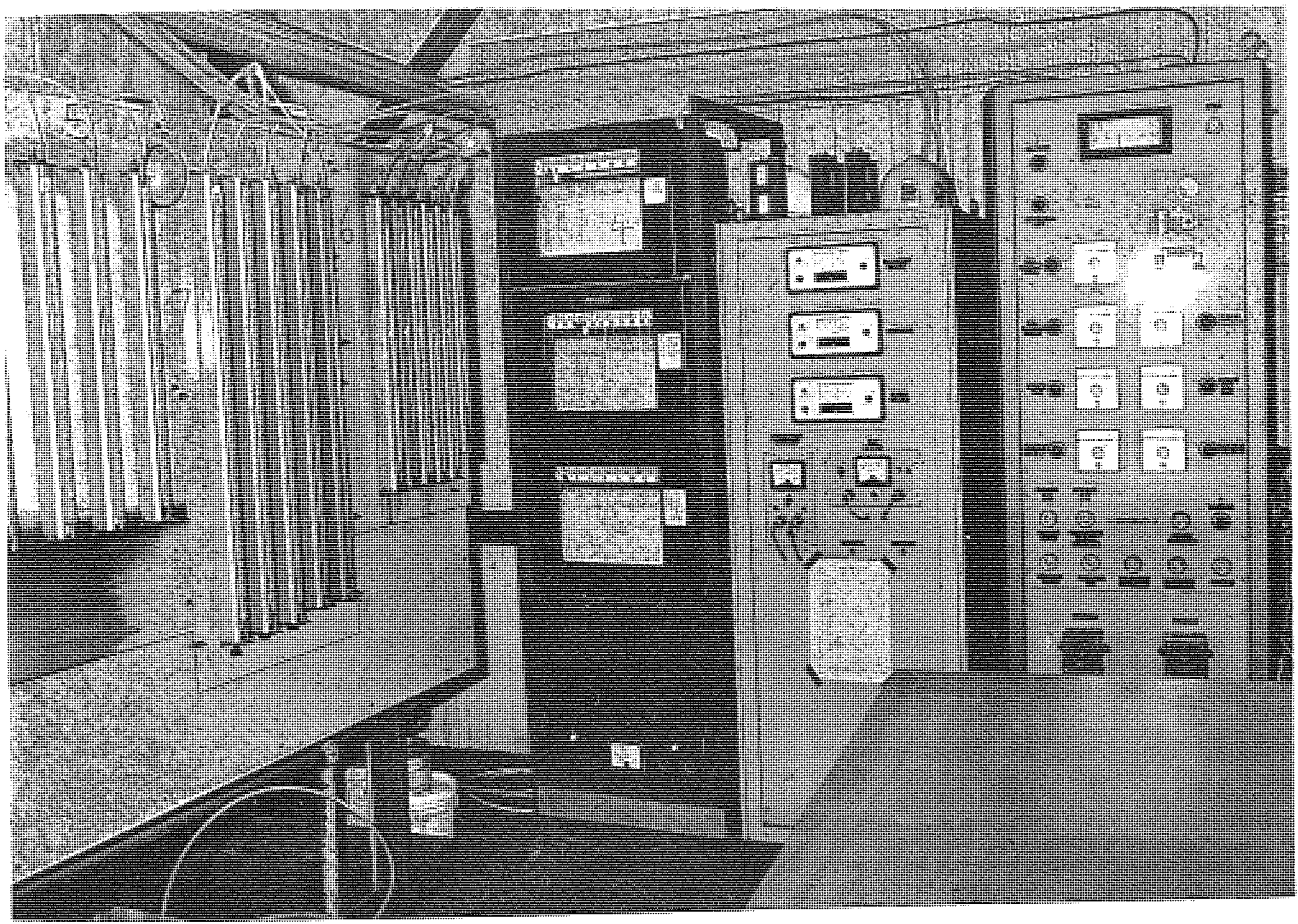
UPPER SEAL ASSEMBLY

FIGURE 7.66



SCHEMATIC ARRANGEMENT OF ROTARY-REGENERATOR TEST-RIG.

FIGURE 7.67



INSTRUMENTATION

FIGURE 7.68

ESTIMATED v ACTUAL FLOWRATE (COLD AIR)

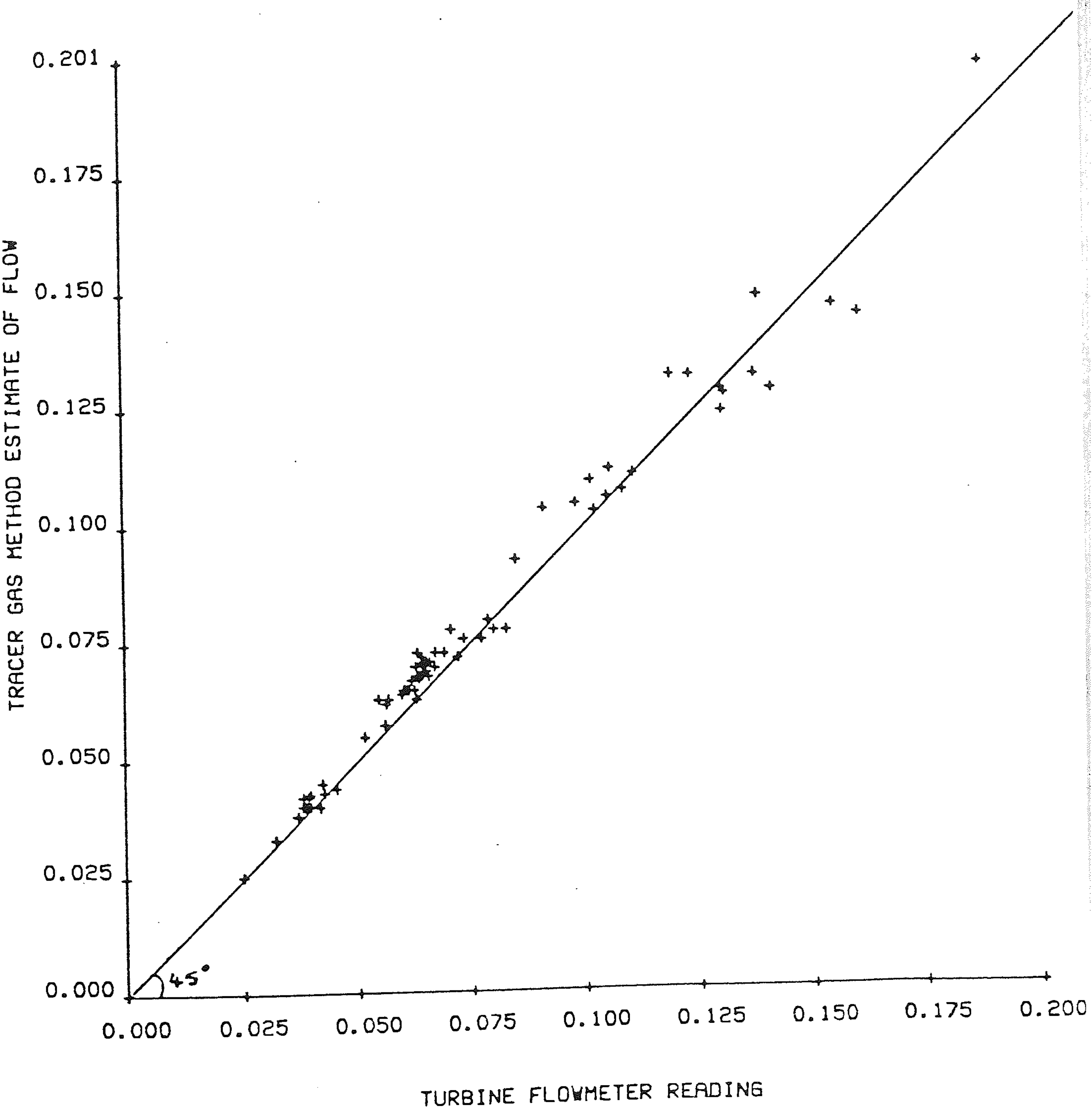


FIGURE 7.69

ESTIMATED v ACTUAL FLOWRATE (HOT GASES)

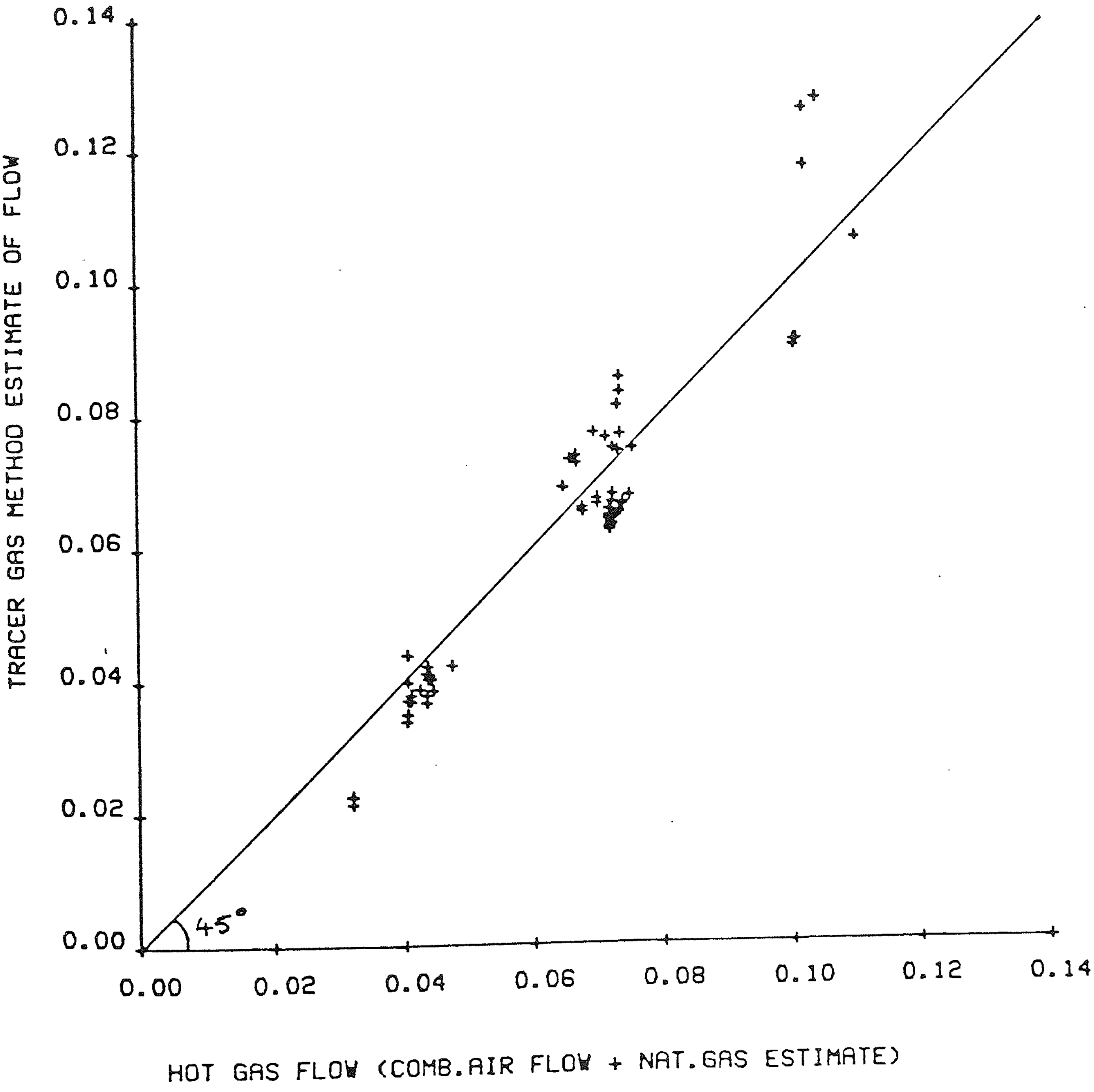


FIGURE 7.70

ROTOR SPEED v AIR PREHEAT

for various WG-IT

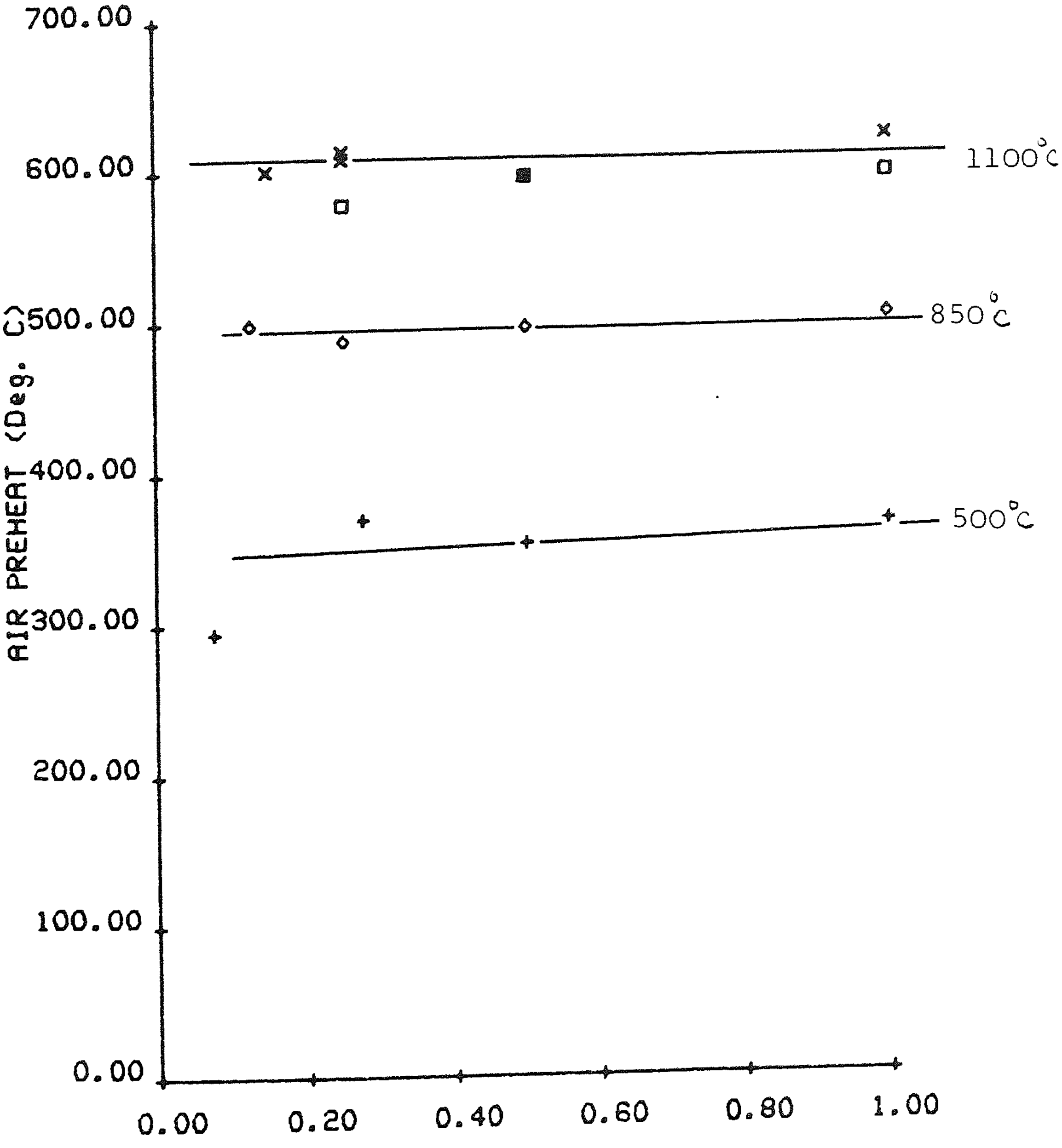


FIGURE 7.71

ROTOR SPEED v THERMAL EFFICIENCY

For Various WG-IT

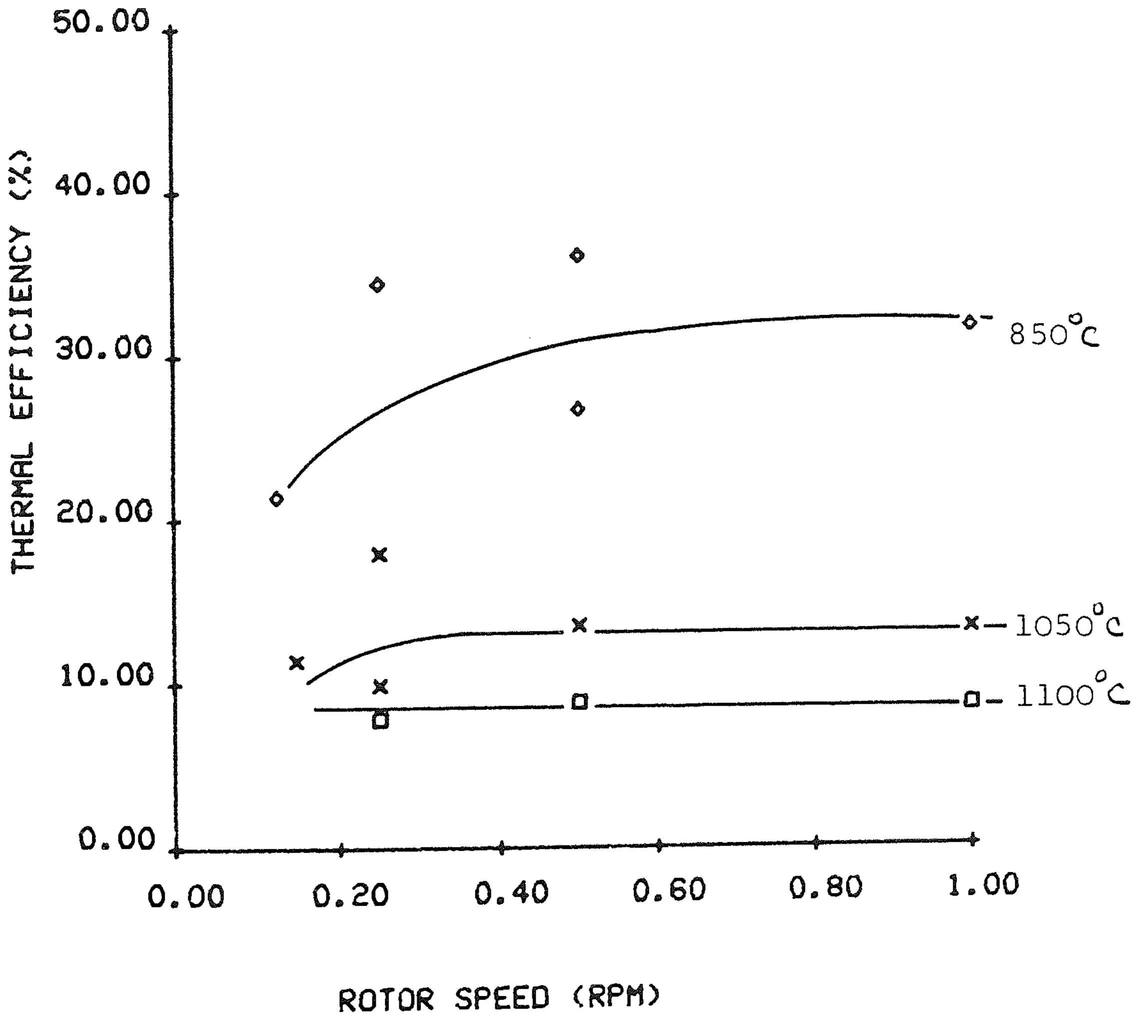


FIGURE 7.72

RELATIONSHIP BETWEEN MAOP AND MAOF

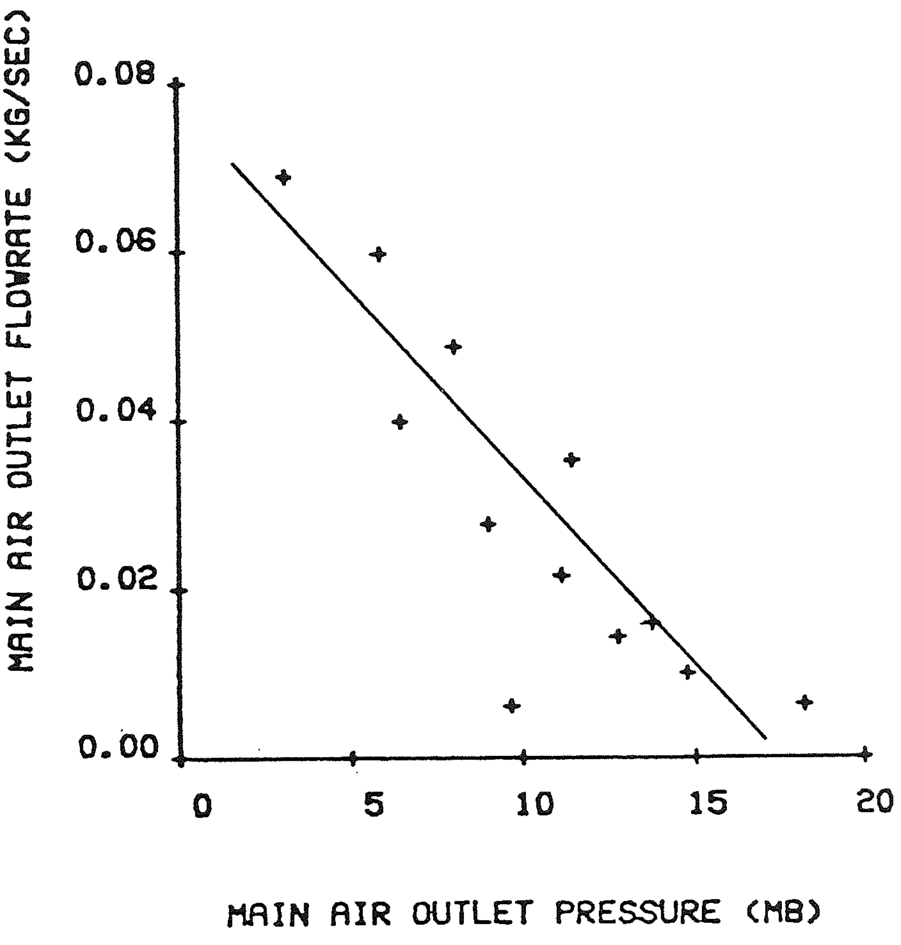


FIGURE 7.73

AIR FLUX DENSITY v THERMAL EFFICIENCY (WGIT=1100)

for Various λ_G

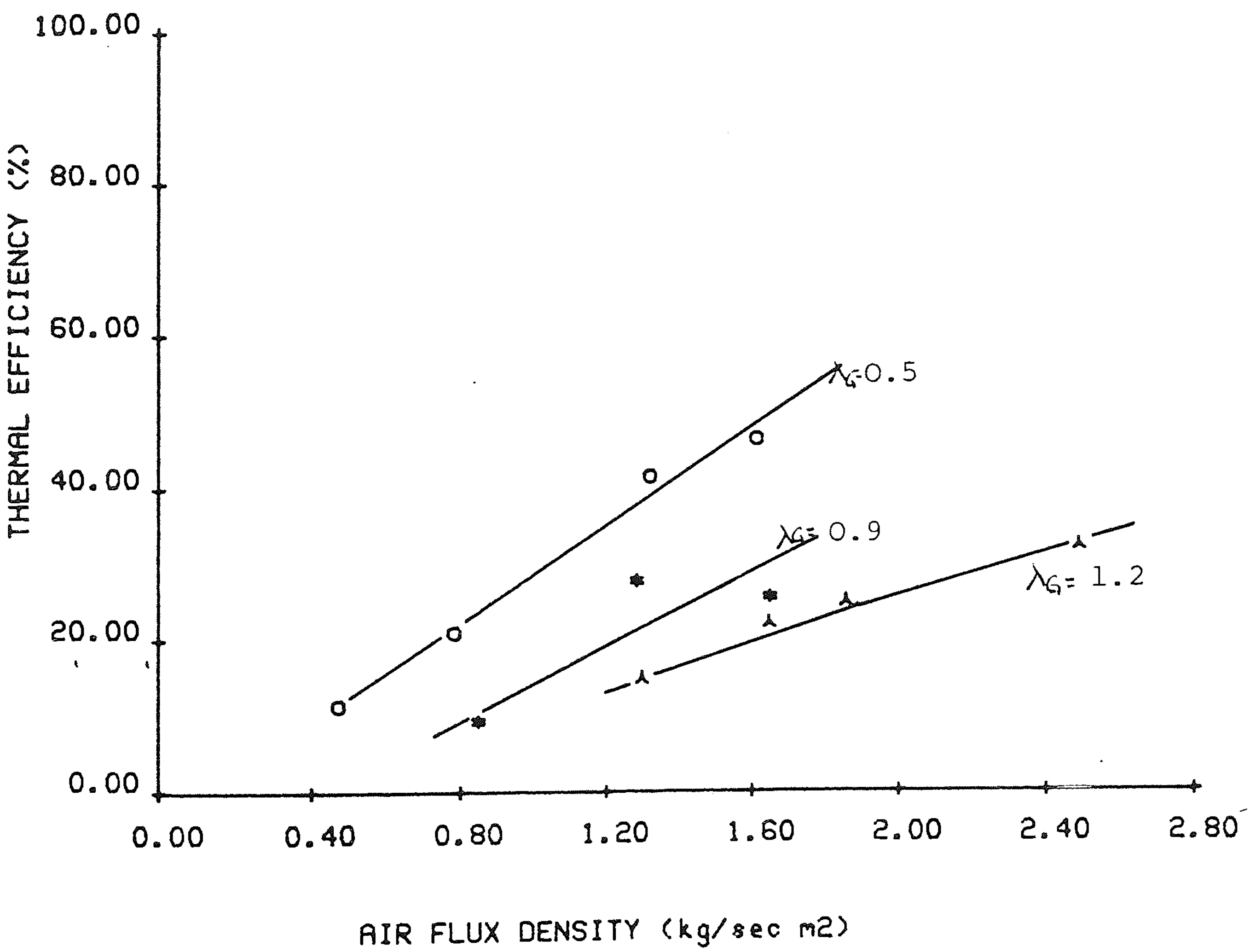


FIGURE 7.74

EFFECT OF FLUX DENSITY (FDR) V THERMAL EFFICIENCY

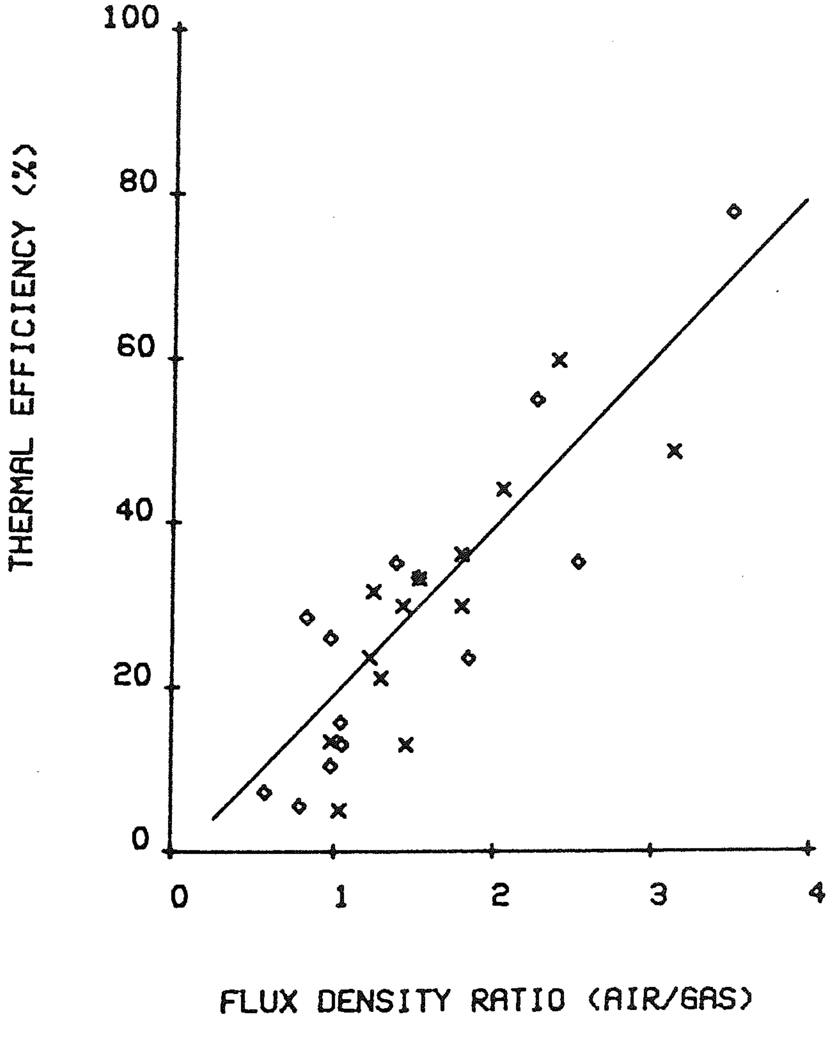


FIGURE 7.75

AIR FLUX DENSITY v AIR PREHEAT (WGIT=1100)

For Various λ_{GS}

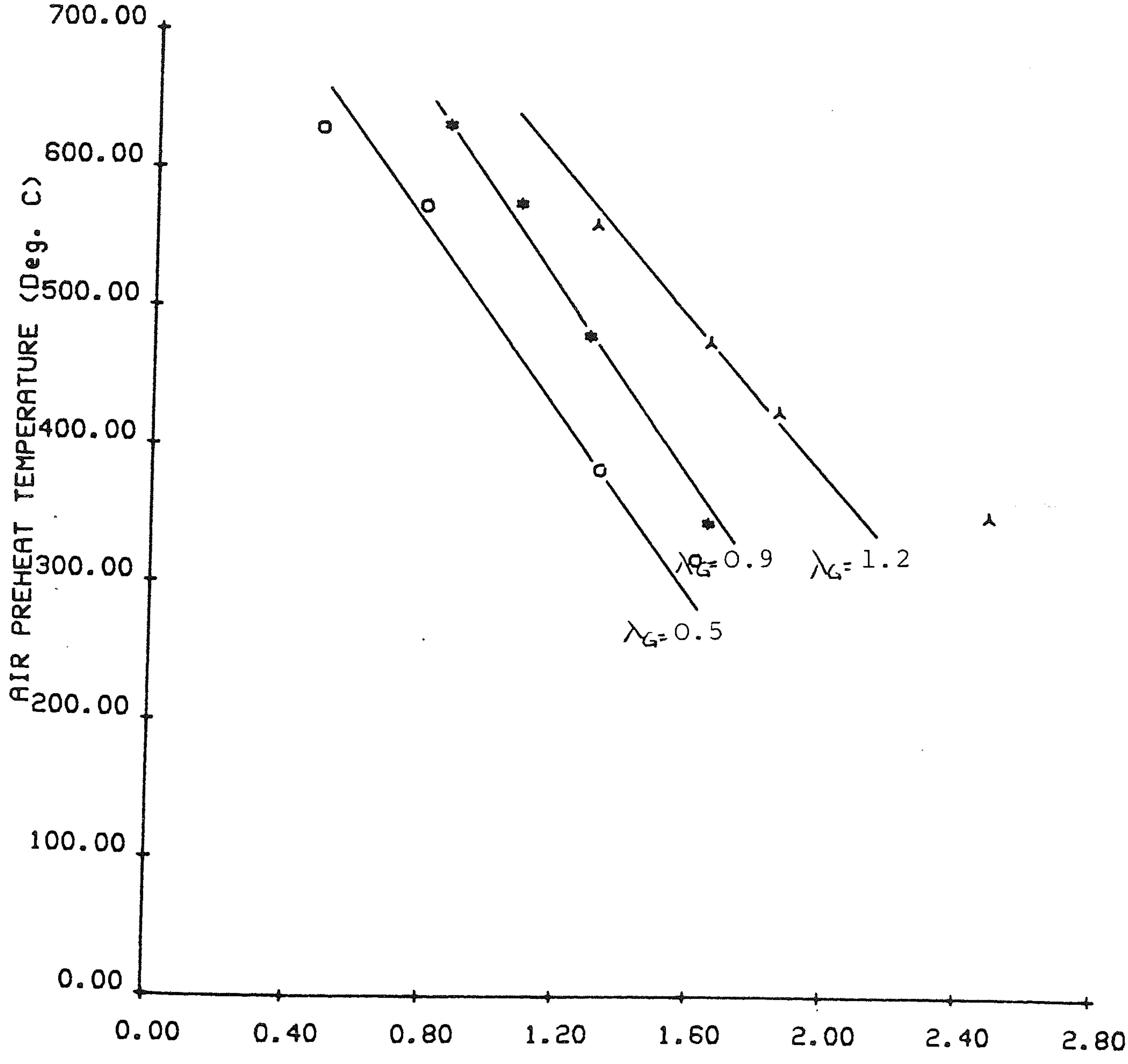


FIGURE 7.76

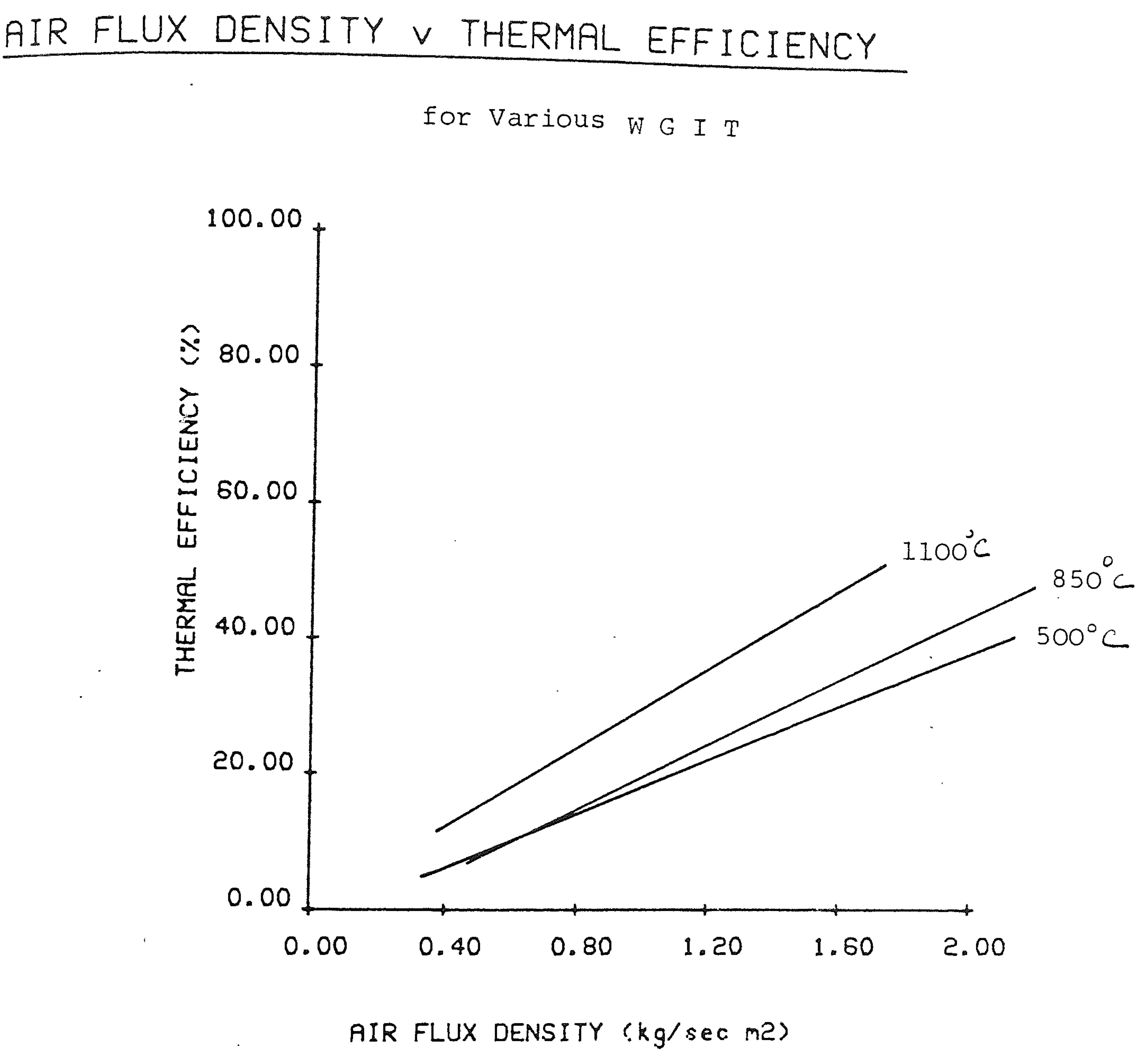

AIR FLUX DENSITY v THERMAL EFFICIENCY
for Various W G I T
THERMAL EFFICIENCY (%)
100.00
80.00
60.00
40.00
20.00
0.00
0.00
0.40
0.80
1.20
1.60
2.00
AIR FLUX DENSITY (kg/sec m2)
1100°C
850°C
500°C

FIGURE 7.77

EFFECT OF FLUX DENSITY (FDR) V PREHEAT

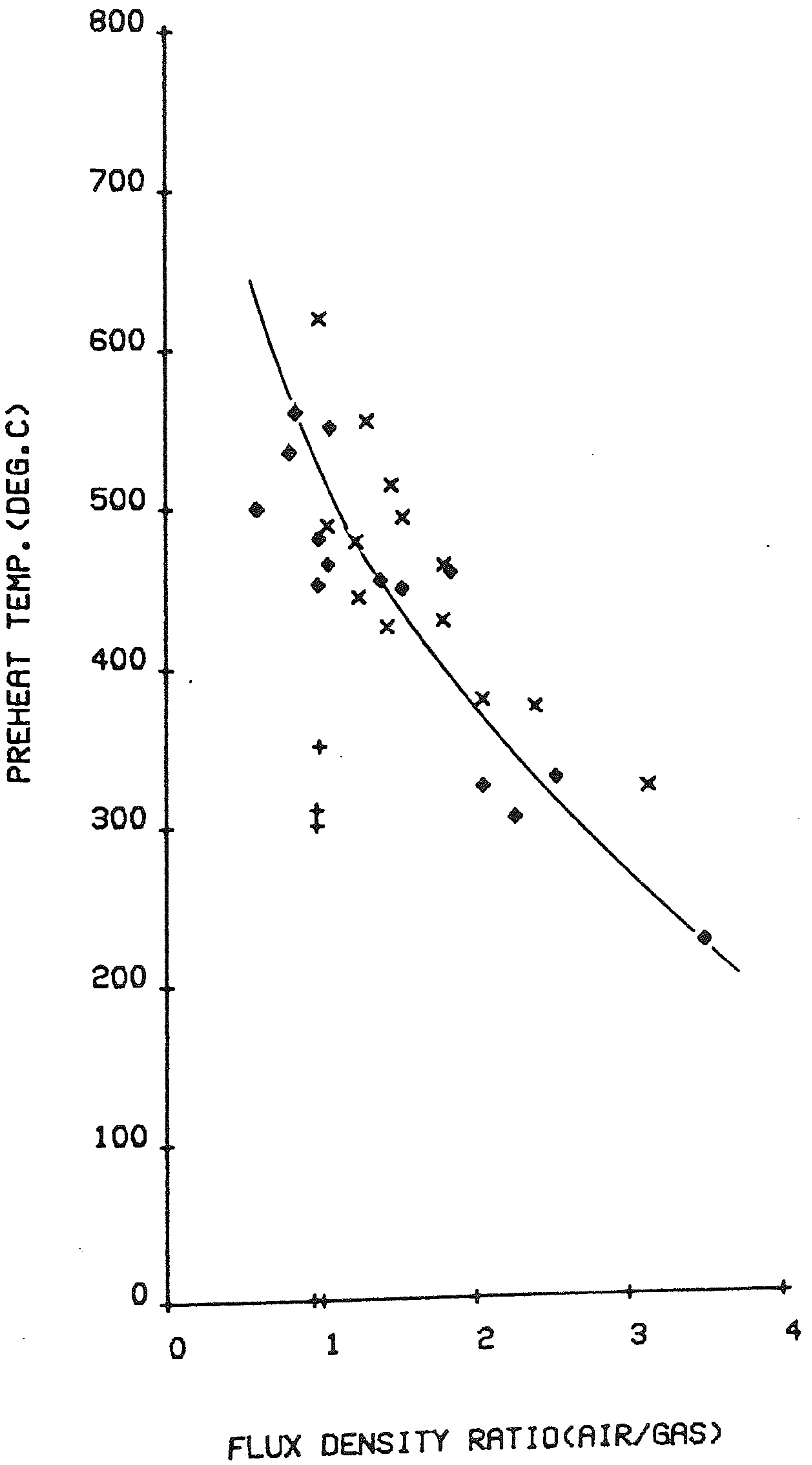


FIGURE 7.73

PRESSURE DIFF. v LEAKAGE FLOWRATE

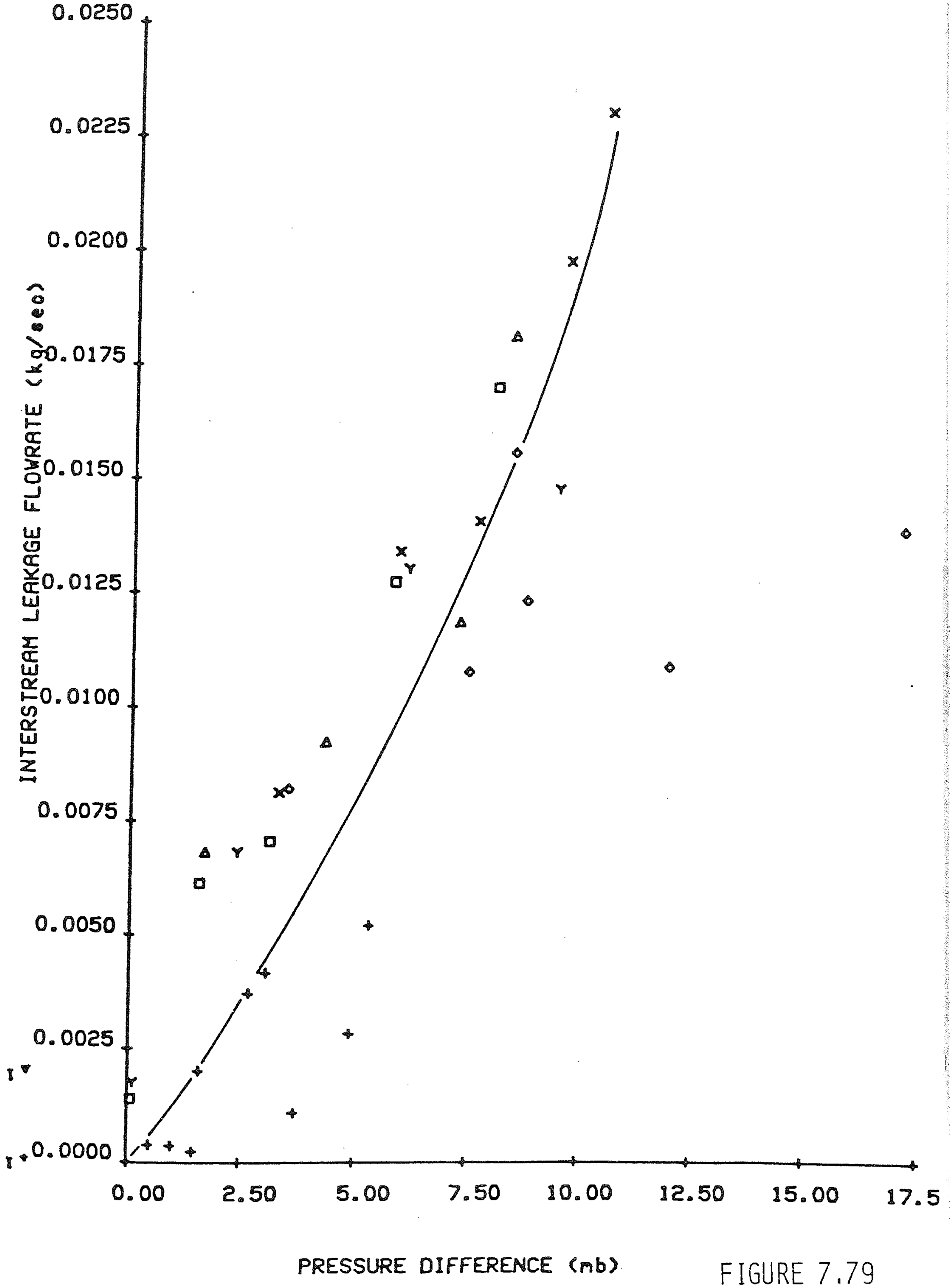


FIGURE 7.79

PRESSURE DIFF. v % LEAKAGE

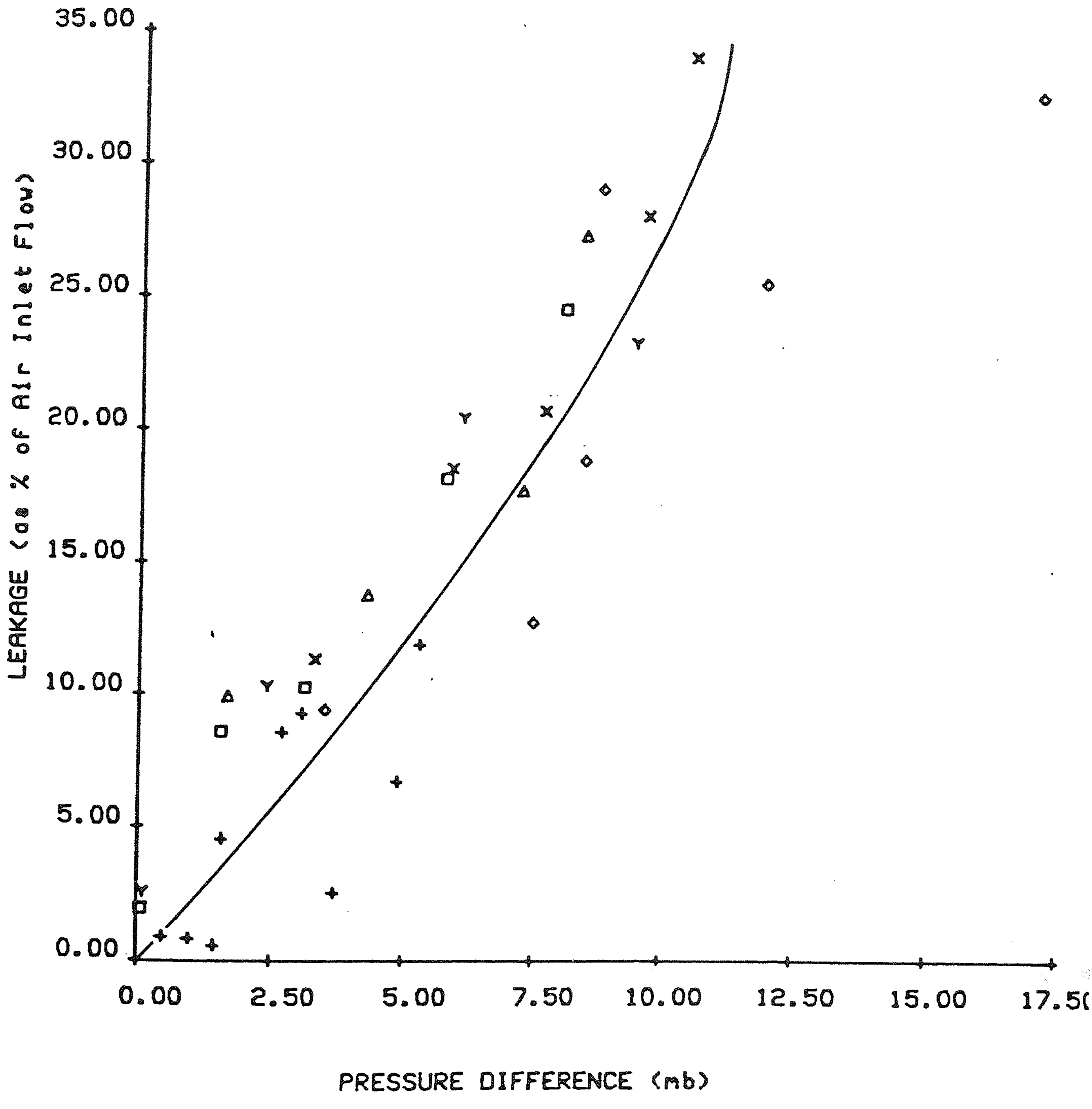


FIGURE 7.80

PRESSURE DIFF. v THERMAL EFFICIENCY

For Various WGIT

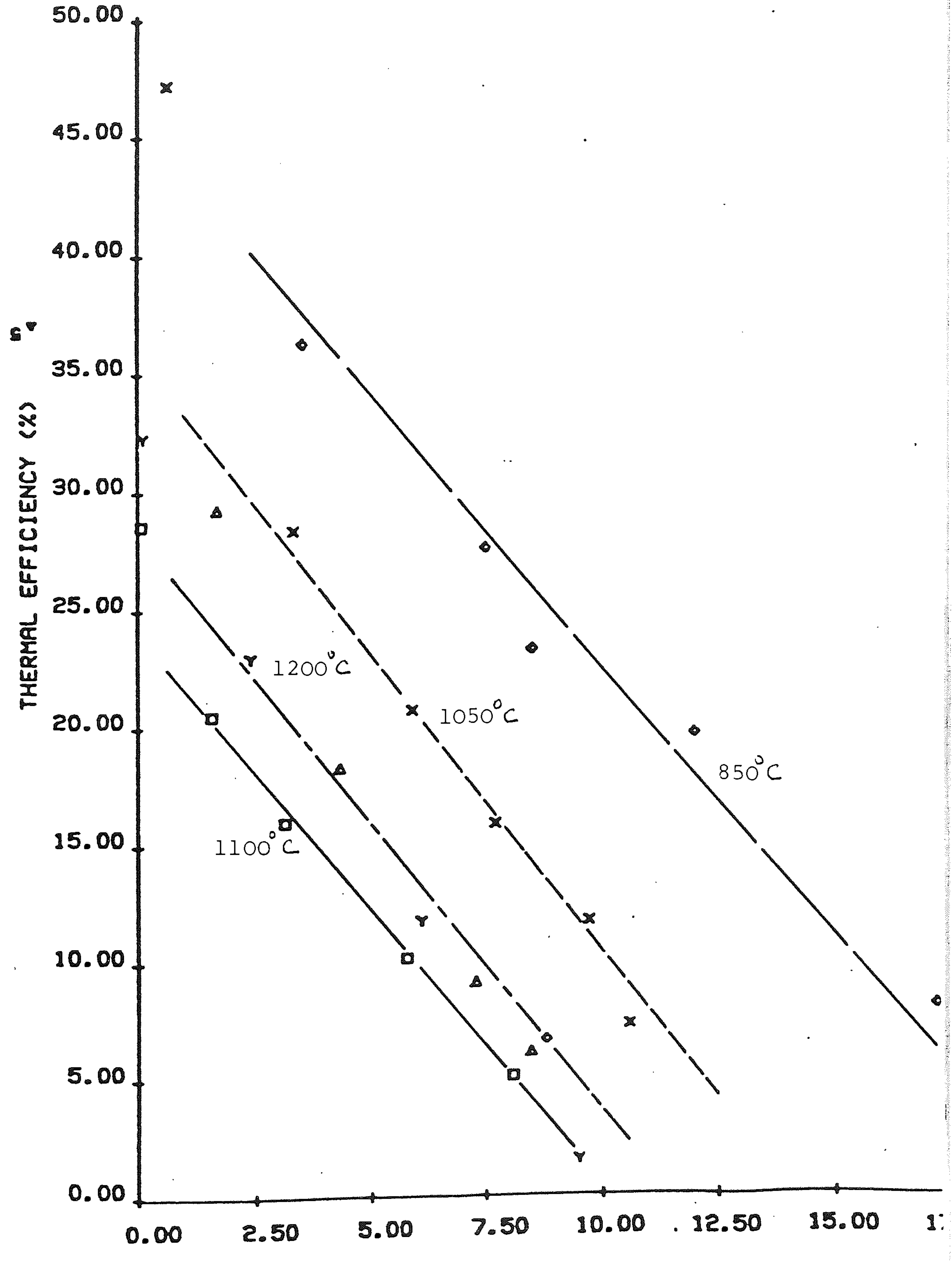


FIGURE 7.81

EFFECT OF FLUX DENSITY (FDR) V ILF

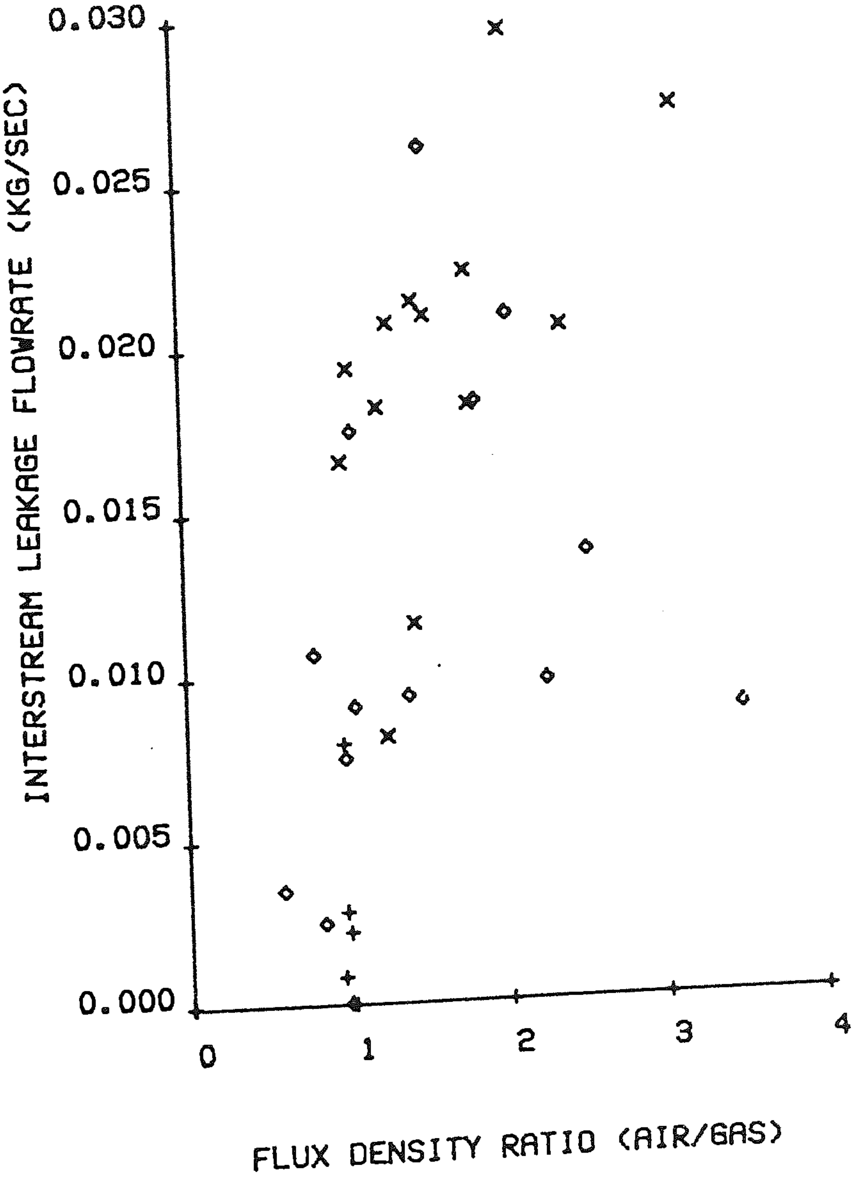


FIGURE 7.82

WASTE GAS OUTLET: PREDICTED VS ACTUAL

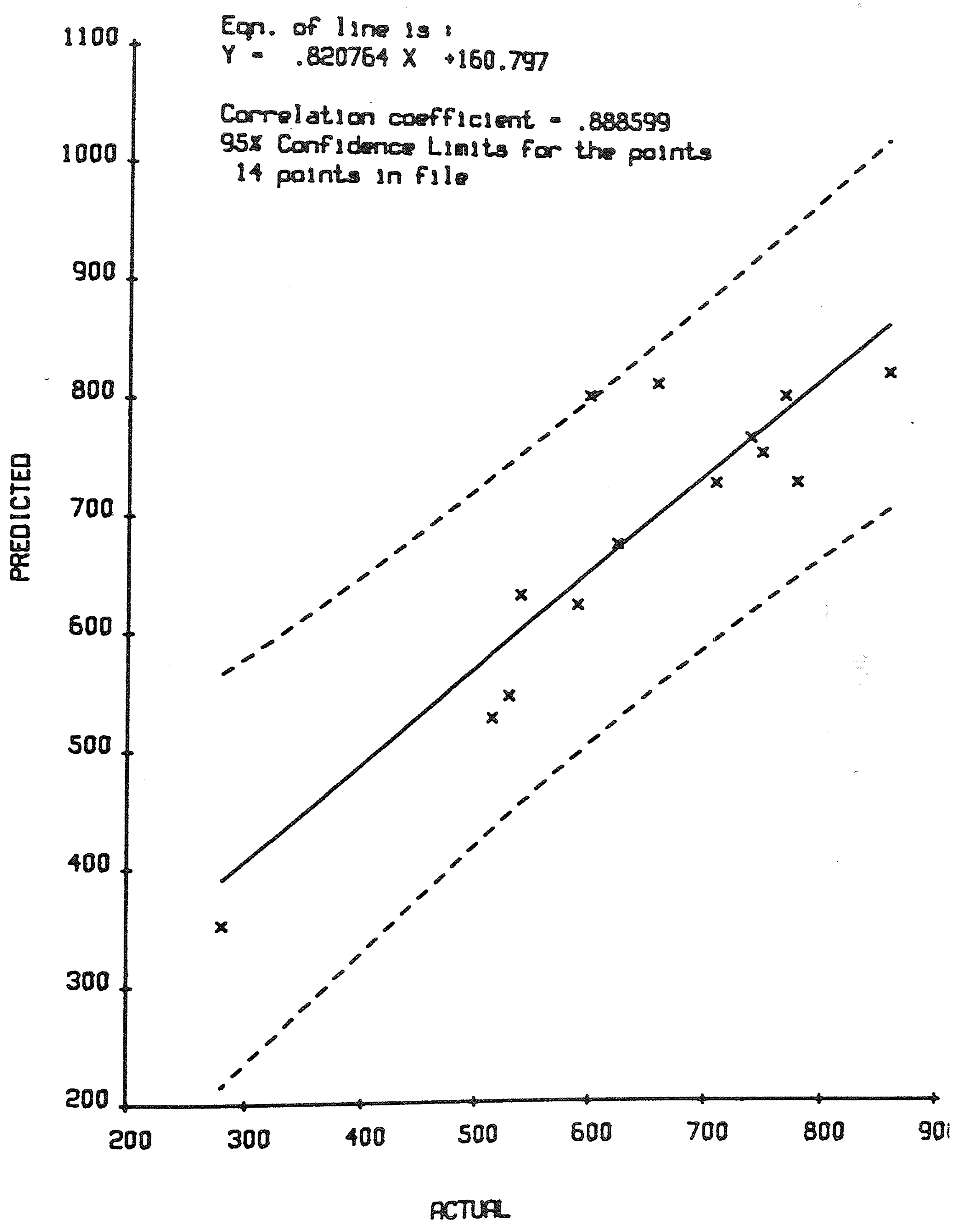


FIGURE 7.83

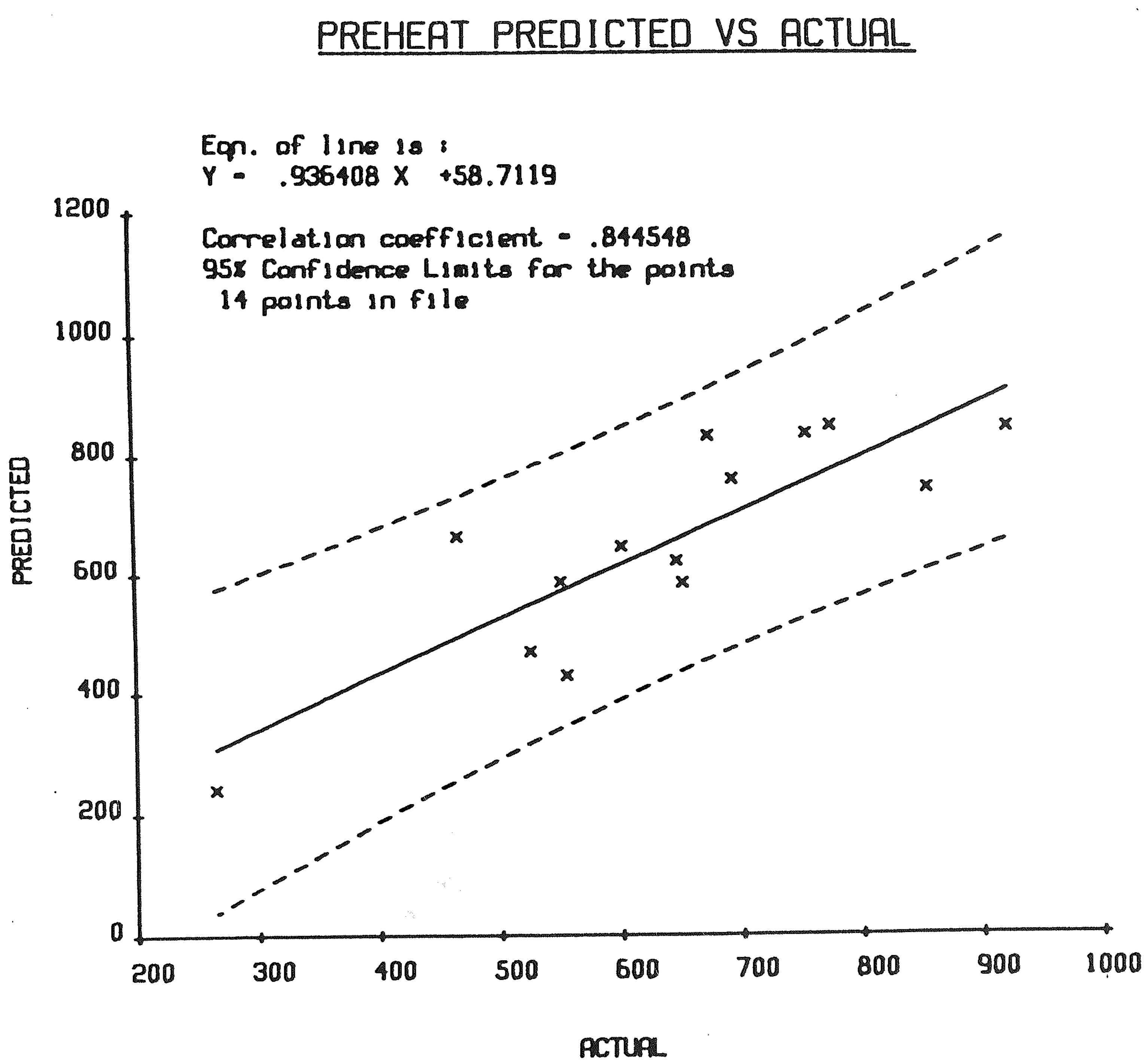

PREHEAT PREDICTED VS ACTUAL
Eqn. of line is :
Y = .936408 X +58.7119
Correlation coefficient = .844548
95% Confidence Limits for the points
14 points in file
1200
1000
800
600
400
200
0
PREDICTED
200
300
400
500
600
700
800
900
1000
ACTUAL

FIGURE 7.84

Air Preheat VS Waste Gas Inlet Temperature

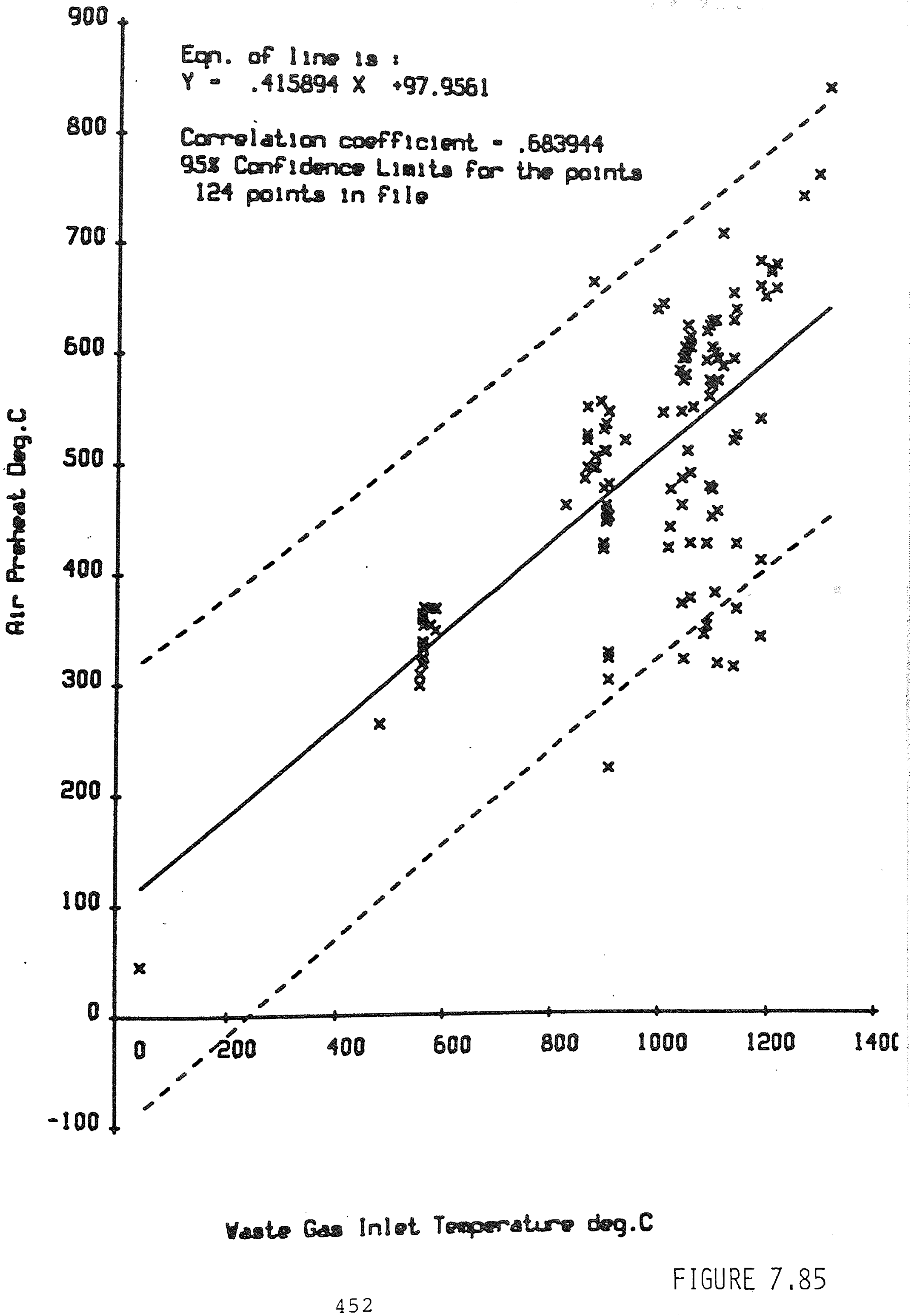


FIGURE 7.85

Interstream Leakage VS Pressure Difference

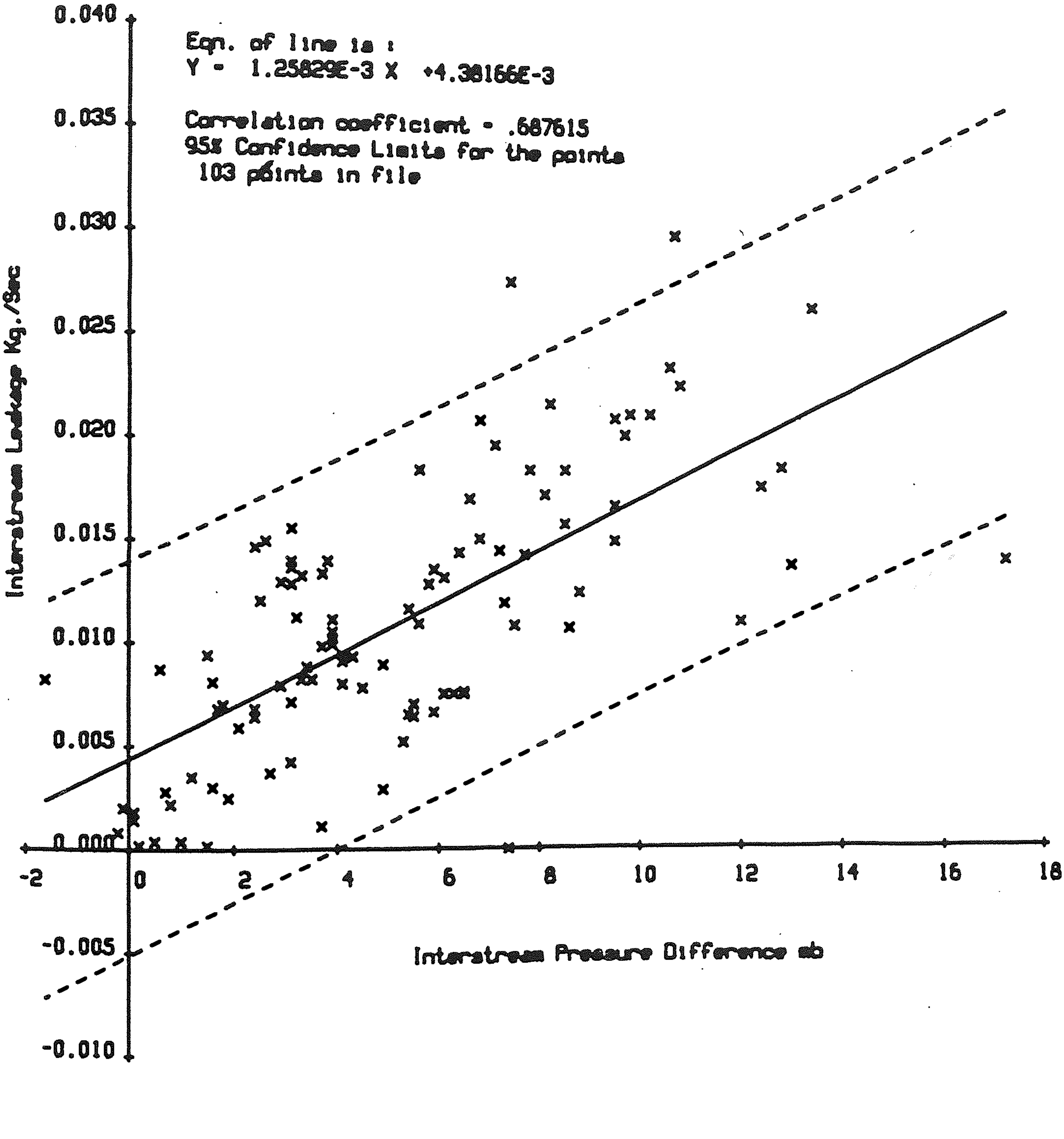


FIGURE 7.86

Thermal Efficiency VS Flux Density Ratio

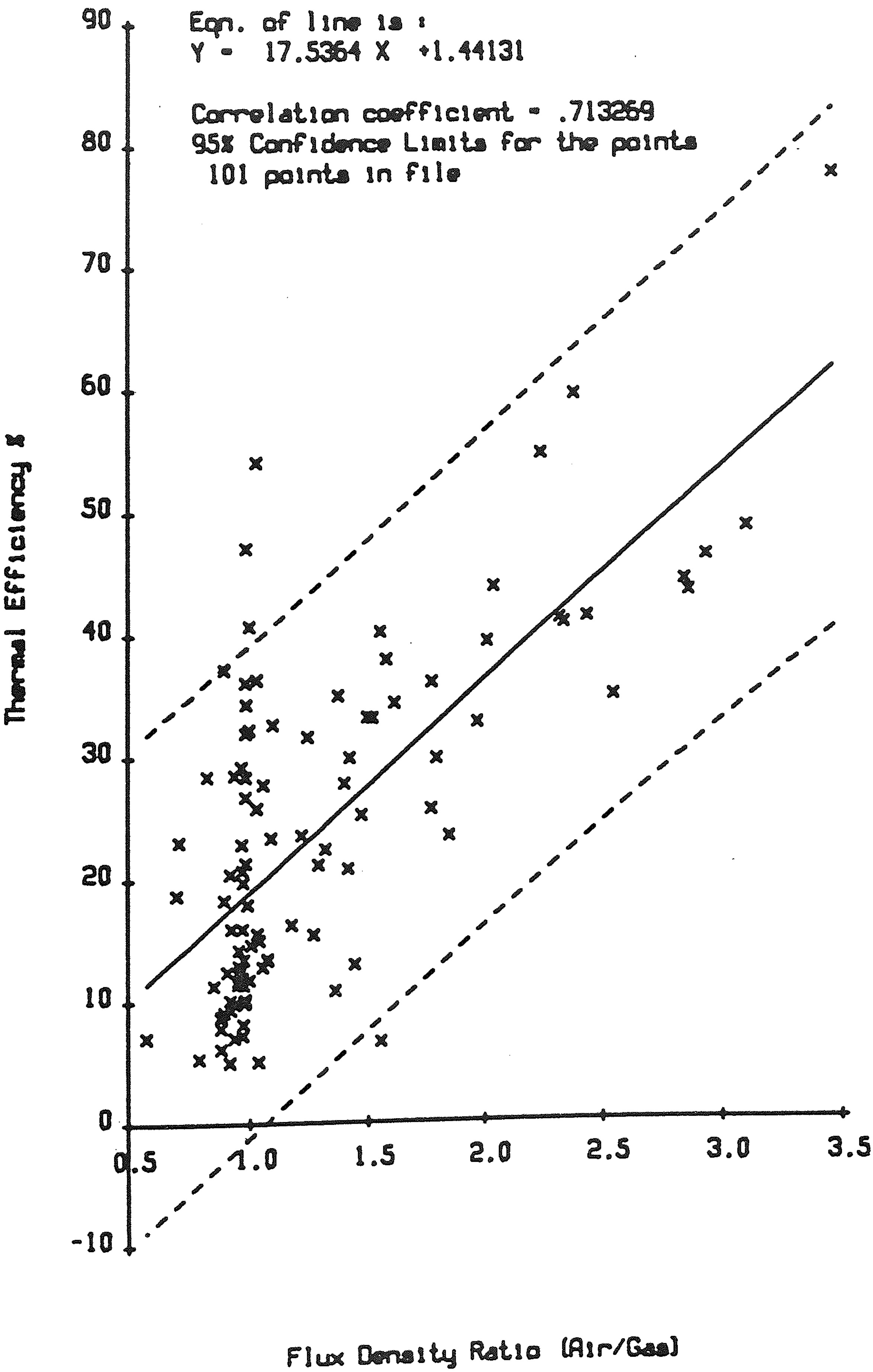
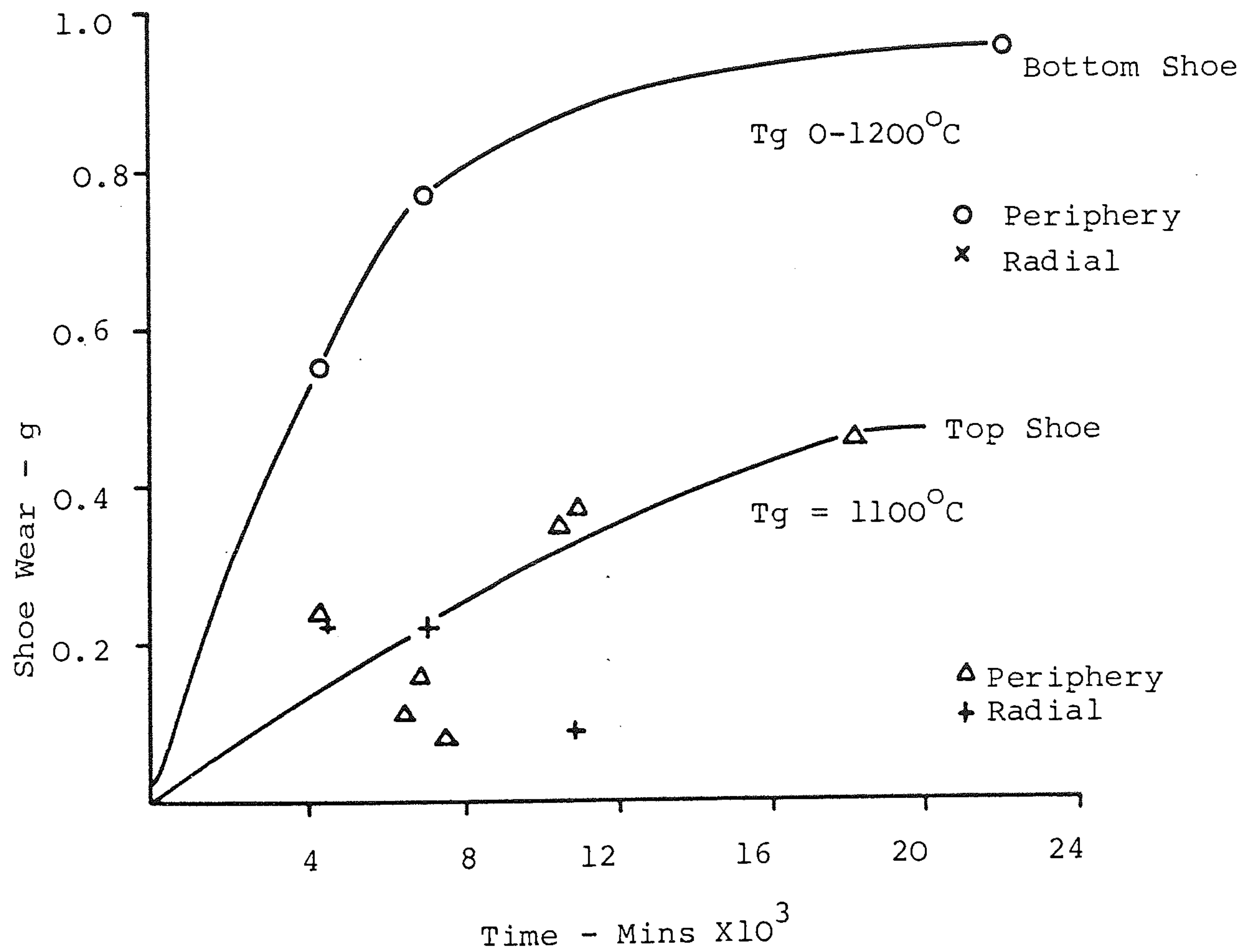


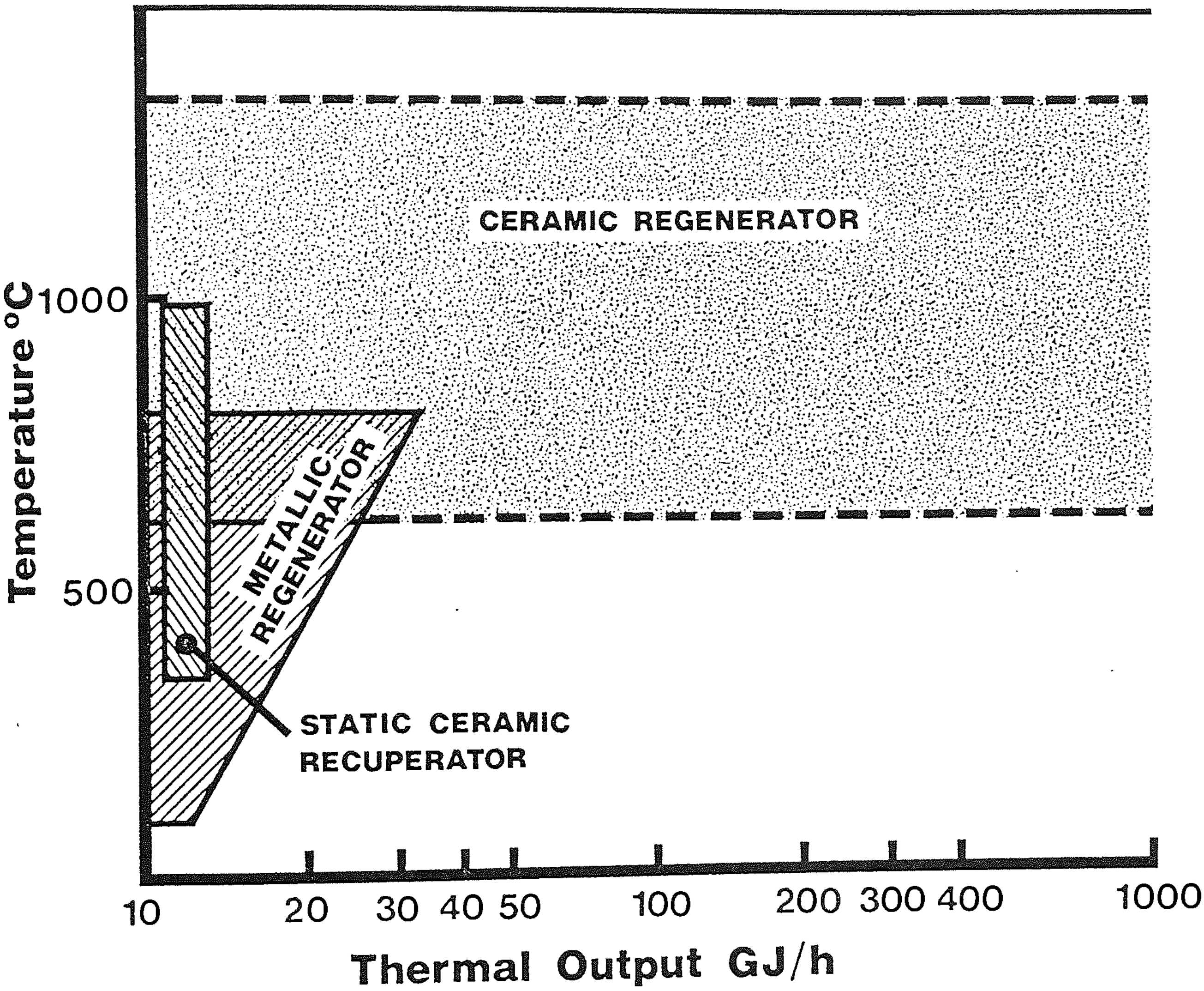
FIGURE 7.87



RATE OF CERAMIC SHOE WEAR

FIGURE 7.88

This graph describes the application of heat recovery devices to certain waste gas conditions ie thermal output and temperature



Range of Operating Heat Exchangers against Predicted Range of CRR

FIGURE 7.39

CHAPTER 8

ENERGY MANAGEMENT

8.1 INTRODUCTION

"We shall not cease from exploration
And at the end of all our exploring
Will be to arrive where we started
And know the place for the first time".

'Little Gidding' - T.S.Elliot.

Energy conservation has now come full circle from primary importance in 1945, owing to post-war shortages, to renewed prominance by 1974 [254,255] No organisation can expect to maintain its position if it fails to ensure security of future energy supply and strict control of its utilisation.[256]

Therefore senior management must increasingly concern themselves with energy[257] and ask:

(b) What are the areas of activity where there may be significant potential for energy conservation or the better use of energy?

(c) What are the procedures that could lead to a more precise identification of options.

Energy conservation requires a formal, systamatic and co-ordinated approach. A sound structure needs to be employed, upon which the search for savings and the following actions can be made. Activities have so often been haphazard in the past that potential savings have not been exploited in the full. The problems thus generally appears in defining the areas in which the investigations should take place and in implementing saving techniques. Often energy is related to other factors such as political or social problems. This involves cost/benefit analysis based on value apportionment of non-quantifiable effetc.(258) Lyle suggests that it would pay factories using large quantities of fuel to employ permanent staff to study heat balances and improve thermal efficiency of the plant. Therefore to achieve an energy conservation an effective "Energy Management" system needs to be established.

8.2 ENERGY MANAGEMENT

Functions of Energy Management can be defined as follows:

1. To motivate management and employees
2. To create awareness and influence employee attitude; this includes management of shop floor.
3. To provide effective communication channels and contact with the necessary personnel and to impart only meaningful and useful information.
4. To control and rectify the monitoring, accounting, budgeting and reporting of usage, cost and efficiency.
5. To identify inefficiency, losses and possible savings through audits measurements and other analyses.
6. To evaluate and recommend conservation projects both short and long term, at the same time controlling their implementation - A priority listing of project feasibility is recommended.
7. To provide adequate maintenance and affect good housekeeping.
8. To make provisions for communications of new ideas and technology both external and within the company.

9. To liase with other factories and existing working groups within the company.

10. To consolidate an energy policy as to purchases, stocks, usage, conservation and available finance for present and future operation.

11. To maintain sufficient technical know-how within the organisation and to initiate research and development or obtain external advice where necessary.

12. To compile a basic manual or handbook outlining the code of practice for the company.

13. To maintain awareness of world and energy developments and to advise senior management.

14. To generally co-ordinate, communicate and control energy matters in the company, due consideration being given to emphasise achievements both internally and externally.

Summarising it can be said that energy management can be broadly divided into two categories:

1. Technical Management: This can be defined where energy savings can be achieved through
   - better understanding of operating procedures
   - alteration of operating procedures, if necessary
   - reduction of energy consumption through improved operation.
   - reduction of energy losses through audit controls, adequate maintenance and good housekeeping.

2. <u>Organisational Management</u>: This can be defined where energy savings are achieved through

- establishing effective energy management structure
- establishing effective energy policy
- creating awareness and influencing attitudes of the work force
- motivation
- communication

These are now described in detail in the following section.

8.2.1 Technical Management

Roberts[259] defines 'Energy Analysis' as :

> ' A systematic way of tracing the flow of energy through an industrial system, resulting in the opportioning of a fraction of the primary energy inputs into the system to each of the outputs of that system'.

The steel industry is not only a major consumer of energy but also a producer of large amounts of fuel gases. Effective utilisation of these arising gases can provide almost half the needs of an integrated steelworks energy requirements. To achieve an effective utilisation of arising gases it is essential to understand the energy and materials flows of any process within its integrated works.

This is well illustrated by Chapman[260] who visualises energy analysis with economic analysis where the former tells of what will happen should certain choices result and the latter tells which option should be chosen.

A suite of programs have been developed to draw out the energy flow network for an integrated steelworks using the PDP 11/45 computer, a graph plotter and data from the integrated steelworks Energy Model. This facility enables more accurate and quicker analysis to be made of the results from the Energy Model. The model can be used to provide a visual display on a board in the energy control room (currently not in existance in many of the BSC Steelworks) to help the Energy Manager achieve an effective utilisation of arising gases in the integrated steel plant.

Therefore effective energy conservation not only requires the needs to minimise energy consumption by plant/process improvement but in a steelworks also requires the need to maximise the use of arising gas and thus minimise the need to purchase fuel, and reduce energy costs.

An effective area in the reduction of energy costs, which many times is overlooked, is the effective purchasing of energy. At the present time when the cost of fuel oil is running at two to three times the cost of coal, replacement of liquid fuels with coal, if possible, may offer substantial energy savings.

A case study on the role of effective utilisation of arising gases and effective purchasing of fuels for Ravenscraig Steelworks is carried out and its incentives projected. This part of the thesis was thus submitted to the works management as a discussion document for energy saving schemes.

A. CASE STUDY - RAVENSCRAIG STEELWORKS

"Replacement of Purchased Fluid Fuels with Coal and Arising Gases"

1. INTRODUCTION

As a first step in developing a strategy the current energy usage and arisings at Ravenscraig were analysed using data collected by the Energy and Coke Department of Teesside Laboratories as part of an energy audit on the Works. This analysis showed that there is a possible strategy which would allow the use of coal to replace almost all of the energy currently purchased as fuel oil and natural gas without the need to gasify the coal. The conclusion points the way to a cost effective fuel replacement strategy for Ravenscraig Works.

An analysis of energy sources and usage over the period considered, outlines a strategy for replacement of purchased fuel oil and natural gas with coal, examines the technical feasibility of its implementation and details the cost savings which could have been achieved by applying the strategy during the fourteen month period up to May 1981. Estimated investment costs together with benefits achieved are given. Areas where detailed technical and engineering studies will be needed are identified.

The report thus offers a technical/investment discussion document for Works Management, identifying areas for detailed technical consideration, which will provide plant improvement and energy savings.

2. ANALYSIS OF CURRENT ENERGY USAGE

The anaylsis of energy usage at Ravenscraig Works is based on statistics for the period April 1980 to May 1981 provided by the Works' Energy Department to Teesside Laboratories as part of an energy audit for the period. The month by month energy analysis, together with an average for the entire period, are presented in Appendix B Examination of these figures show considerable variability of the total energy requirement across the Works per tonne of crude steel, covering the range 22GJ/t to 34 GJ/t for the monthly mean energy requirements. Similarly the detailed breakdowns of energy sources and usage by plant and fuel type show month to month variability. The reason for the variability in energy usage is mainly attributed to variations in the activity levels as a result of demand fluctuations coupled with changes in the use of specific fuels occasioned by operational requirements. On average over the fourteen month period the total net energy used is 26.1 GJ/t crude steel of which 21.8 GJ/t is as solid fuels (coal, coke and breeze) and 4.5 GJ/t as purchased fluid fuels, comprising 1.5 GJ/t natural gas and 3.0 GJ/t fuel oil. Electrical power accounted for 0.75 GJ/t and a credit of 0.95 GJ/t is obtained for the coke ovens byproducts, tar and benzole.

Within the Works about 40% of the solid fuel energy input becomes available as gaseous fuel arising from the operations of coke and blast furnaces. The arising gases together with the purchased fuel oil and natural gas may be grouped together as fluid fuels which, with one exception, are solely used for heating purposes rather than as reactants in the chemical and metallurgical processes carried out in the Works. The exception is fuel oil injected to the blast furnace which acts as a reductant and carbon source in the smelting process. On average over the period the oil injected amounts to only 0.15 GJ/t crude steel.

Figure 8.1 represents the type of fluid fuels used and the uses to which they were put averaged over the fourteen month period, excluding fuel oil used in the blast furnaces. Of a total of 13.1 GJ/t crude steel of fluid fuels consumed in the period about two-thirds of the fluid fuel requirement was satisfied from works arising gases and the other third was purchased as natural gas and fuel oil.

About 60% of the total fluid fuel usage is employed for process heating purposes, with the blast furnaces, coke ovens and strip mill being the major users in that order. Steam raising is the biggest single consumer taking 32.9% of the total fluid fuel used. The majority of steam is raised (2.5 GJ/t) at high temperatures for mechanical drives of the blast furnace blowers and electrical power generators, and a smaller amount of low pressure steam (1.2 GJ/t) is used at the various process plants. It is interesting to note that the fuel requirement for steam raising (4.3 GJ/t) is almost equal to the total purchased fluid fuel requirement (4.4 GJ/t).

Some bleeding of arising gases took place which accounted for about 5.1% of the fluid fuel total, and about 2.2% was lost.

3. STRATEGY FOR REPLACEMENT OF PURCHASED FLUID FUELS WITH COAL

Over the fourteen month period for which the energy usage analysis is carried out the fuel requirements for steam raising is very close to the total natural gas and fuel oil purchases, and in the same period no coal is used directly as a fuel. Since coal fired steam raising is a well developed and widely used technology the obvious starting point for a coal replacement strategy is to consider using coal as a boiler fuel. This could directly replace fuel oil used in the boilers and would release the arising gases currently firing the boilers for use as a replacement for purchased fuels in other applications. It is then necessary to consider the possible uses for the arising gases in these applications with regard to their calorific value and flame characteristics.

Blast furnace gas with a relatively low calorific value of around 3 MJ/m^3 when combusted with air gives a theoretical heat release per unit volume of combustion products of 2.1 MJ/m^3. This is considerably less than can be achieved with the purchased hydrocarbon fuels (3.6 MJ/m^3 with natural gas and 3.8 MJ/m^3 with fuel oil) and so blast furnace gas cannot be considered for high grade heating applications such as slab reheating furnaces. However, the regenerative heating of blast furnace stoves offers the possibility of using the relatively low temperature combustion gases efficiently for blast heating. Also, six out of seven of the coke oven batteries at Ravenscraig are designed for use with blast furnace gas underfiring. Together the blast furnaces and coke ovens use 33.4% of the total fluid fuel energy over the fourteen months studies, and the blast furnace gas made during the period, was 40.9% of the total. It may be seen then that if blast furnace gas were used for all of the heating requirement of blast furnaces and coke ovens there would be a surplus, which could not be used at the other plants and would be best employed in steam raising.

The other main arising fuel gas is coke ovens gas with a calorific value of about 15-20 MJ/m^3 and a theoretical heat release of about 3.7 MJ/m^3 of combustion products. It may be seen that this is equivalent to the purchased hydrocarbon fuels and could be used as a direct replacement although burner changes would probably be required since the flame speed and Wobbe number of CO gas are significantly different from those of natural gas, and of course fuel oil burners would not be suitable for gaseous fuels.

The natural gas and fuel oil usage at the ore preparation, BOS, Concast, Slab and Strip mills plants represented 26.4% of the total fluid fuels energy, and the coke ovens gas make during the same period was 25.5% of the total.

There is then a small deficit which would have to be made up from purchased fuels.

Although the considerations above show that there is, on average, an excess of blast furnace gas over the requirement for blast furnaces and coke ovens, and a deficit of CO gas for other purposes, it must also be taken into account that the variability in operation of individual plants will call for a degree of flexibility in any strategy. This is provided in the proposed method of operation which follows.

It is proposed that the blast furnace stoves and coke ovens should be fired exclusively on blast furnace gas, with any excess being used for steam raising and any deficit made up with coke ovens gas. Bleeding of blast furnace gas should be avoided since the steam raising plant can absorb any surplus gas. The other processs heating applications should be fired with coke ovens gas, supplemented as necessary with purchased fuels. Any surplus of CO gas, after all the process heating requirements have been satisfied, should be used for steam raising. Bleeding of CO gas should not be necessary. All steam raising should be fired with coal plus any surplus arising gases which may be available.

4. PROJECTED OPERATING COST SAVINGS

In order to test the general applicability of the proposed strategy under a variety of operating conditions, and to quantify the potential operating cost savings which could be achieved, the monthly average energy data for the fourteen months considered are used to generate new energy and cost data assuming implementation of the strategy as proposed in section 3. The results of this exercise and their derivations are presented in Table 8.1.

The study assumed that overall energy requirements are the same as in the historical records except that no arising gases would be bled. Losses however are included in the totals since they are presumed to be unavoidable. Because of the small amounts involved and considering future strategy, fuel oil injection to the blast furnace is entirely excluded from the exercise. The energy requirements for the various plants are taken directly from the historical data, as are the blast furnace and coke ovens gas makes, and the total purchased fuels requirements. The costs are estimated assuming the price of imported coal was £1.10/GJ and the prices of natural gas and fuel oil at £2.67/GJ. These prices are those ruling during early 1981.

An analysis of the results show that over the fourteen month period there would have been three months where there was a net deficit of blast furnace gas, and coke ovens gas would have had to be used to supplement stove and coke ovens firing. These three months correspond to low general activity levels with weekly crude steel makes of below 20,000 t. During eleven of the fourteen months there would have been a net deficit of coke ovens gas, requiring purchase of some fluid fuels to balance the energy requirement, the largest being 1 GJ/t crude steel. The additional coal required for steam raising thus varied between 2.3 and 6.3 GJ/t over the period, with an average of 3.3 GJ/t.

The estimated cost savings ranged from £5.13/t to £11.71/t with an average of £7.02/t over the entire fourteen month period, and the total estimated cost saving was £11.6 million over the period. It is interesting to note that the saving per tonne tended to be higher during periods of reduced activity, when historically both the overall energy usage and the purchased fuels requirements per tonne have been high.

5. TECHNICAL CONSIDERATIONS

In this section the technical feasibility of implementing the fuel replacement strategy is briefly discussed. It will of course be necessary to investigate more fully the engineering of the proposed changes in fuels usage and this section does not aim to present a detailed description for all of the plant modifications which will be needed. It is intended rather to establish the broad principles, which will then require further detailed design effort and to provide a basis for budgetary capital cost estimates. The implications of the proposed changes are considered for each of the major plant units affected.

5.1 Blast Furnace Stoves

The use of blast furnace gas as the only fuel in the blast furnace stoves should present no technical problems except possibly in achieving the desired blast temperature. During those months for which comprehensive data are avaialble, the average blast temperature was 1045°C with a peak value of 1167°C. In the following discussion it is assumed that the required blast temperature would not be required to exceed the average values.

Davison[261] employed a correlation between calorific value of the fuel gas and hot blast temperature in good operating practice which predicts that blast temperatures up to 1050°C could be obtained with a fuel gas of not less than 2.98 MJ/m^3 calorific value. Over the period analysed the recorded blast furnace gas CV varied between 2.7 and about 3.2 MJ/m^3, although towards the end of the period it is consistently above 3.0 MJ/m^3, when oxygen enrichment of the blast is employed. If this practice is continued in the future there should be no difficulty in operation under the proposed strategy. However, this prediction would require confirmation by a detailed analysis of the stove design and operation.

If further scrutiny reveals that the required blast temperature could not be consistently provided using blast furnace gas/air combustion in the stoves the possibility of oxygen enrichment of the combustion air should be considered. Oxygen enrichment would permit the heat release per unit volume of combustion products to be enhanced up to the point where a pure oxygen/blast furnace gas flame would provide a higher flame temperature than a natural gas/air flame. The combustion of blast furnace gas with oxygen has been studied by the International Flame Research Foundation[262] who have confirmed experimentally that there is little difficulty in obtaining a stable flame with good temperature and heat transfer characteristics provided an appropriate burner design is chosen. This option is attractive in the Ravenscraig situation since information from BOS indicates that tonnage oxygen is available at a very favourable cost. The current oxygen contract is based in a contracted take which exceeds the A.O.P. demand level by about 500 t/day, and up to this level of additional usage oxygen may be employed at a favourable marginal cost. It may be shown that to achieve a flame temperature equivalent to that of combustion with air of blast furnace gas enriched by 20% with coke ovens gas would require about 100 t/day of oxygen for air enrichment, when the works is operating at 27,000 t/week.

5.2 Coke Ovens

There are four coke ovens plants designated A, B, C, D of which A, B and C plants have two batteries of ovens each and D plant has a single battery. A, B and C plants are provided with blast furnace gas supplies and information available to date suggests that they were designed for operation with only blast furnace gas as the underfiring fuel. This needs further investigation to confirm the feasibility of firing on blast furnace gas, but if this should reveal problems in achieving the required temperatures, oxygen enrichment of the combustion air could be considered. In the case of D plant the possibility of conversion to blast furnace gas underfiring should be investigated, but it may be more cost effective to operate as far as possible with only A, B and C plants and tolerate the use of coke ovens gas for underfiring D plant if it is needed to meet increased demand or to replace the output of one of the other batteries during repairs or maintenance.

5.3 Ore Preparation

The ore preparation area uses only 1.9% of the total fluid fuel requirement mainly for sinter ignition firing. There should be no problem in operating with 100% coke ovens gas, as has been practised in the past.

5.4 BOS and Concast

The BOS and Concast plants together use only 3.3% of the total fluid fuels, all as natural gas. This is mainly for preheating of equipment such as ladles and no problems in using coke ovens gas are anticipated.

5.5 Slab Mill

The slab mill fuel usage is almost entirely in the soaking pits, of which there are twenty-six in total. Of these, twenty are already capable of multi-fuel firing with coke ovens gas, natural gas or fuel oil. It is proposed that no modifications should be made to the soaking pits but that operations should be scheduled to maximise the use of coke ovens gas. This would also be a convenient point to use some purchased fuels to maintain the overall works energy balance as necessary. Since the soaking pits feed the slab mill which operates seven days a week, day shift only, they are in fairly constant use and no problems are expected through high peak fuel demand levels.

5.6 Strip Mill

There are three slab reheating furnaces feeding the strip mill, of which two are normally operating and the third is under maintenance. The layout of these furnaces is shown schematically in Figure. 8.2 . It may be seen that these furnaces all incorporate both fuel oil and gas fired burners. The gas burners are capable of burning coke ovens gas without modification. It is proposed that the oil burners should be replaced with multi-fuel burners which could use either low pressure gas (natural or coke ovens) or fuel oil. This would allow the reheating furnaces to be fired on whatever coke ovens gas may be available but also to use purchased fuels to balance the overall energy demand. Choice of appropriate burner configurations to provide equivalent thermal input and temperature distribution in the furnaces should not present any insuperable problems.

5.7 Steam Raising Plant

The steam raising plant currently in use is designed for operation with fluid fuels only. Enquiries made with boiler manufacturers Babcock and Wilcox Ltd., and Gibson Wells Ltd., showed that conversion of the existing boiler to coal firing would be as expensive as installing a new boiler. It is proposed therefore that the existing boiler be replaced by a multi-fuel unit capable of firing with pulverised fuel and/or arising gases. This approach has a further advantage that the old boiler could continue to be used whilst a new boiler house is being erected adjacent to the existing power services building and the changeover would cause minimal disruption to operations.

Coal preparation facilities would also be needed, although the existing coal reception and stocking areas could probably deal with the additional coal which would be about 20-25% of the current coking coal requirement. The additional equipment would be coal crushers and driers and probably conveying equipment. Ash disposal facilities will also be needed.

5.8 Fuels Distribution Systems

The existing blast furnace gas holder and distribution system which already serves the blast furnaces and A, B and C coke ovens plants could be used under the proposed strategy without alteration. If it were judged desirable to convert D coke ovens plant to blast furnace gas firing the main would have to be extended, but the distance involved is not great.

The coke ovens gas may already be fed to a low pressure gas main serving the finishing end of the works. It is not certain that the capacity of this main would be sufficient to allow all of the coke ovens gas made to be piped, but if on further investigation additional capacity proves to be necessary this could be routed on the existing pipe bridges.

It has been reported in the past that neither the combustibility nor the cleanliness of the coke ovens gas supply at Ravenscraig have been as good as could be desired. This has resulted in problems in ensuring even combustion and fouling of mains, control gear and burner jets. The successful adoption of the proposed strategy will require the coke ovens gas to be consistently clean and of stable CVx, and further detailed investigation of the byproducts plant operation may be required to identify improvements to ensure that these standards can be met.

6. CAPITAL COST ESTIMATES

In this section the estimated capital investment costs of the major modifications are presented, together with the sources of information on which they are based and any assumptions made in estimating. It must be appreciated that these cost estimates have been made to allow a first assessment of the economic viability of the proposal, and whilst they are based on the best information available at this stage they will require confirmation by a more detailed study of the engineering work

6.1 Boiler

The installed cost of a multi-fuel fired boiler has been estimated by both Babcock and Wilcox Ltd. and Gibson Wells Ltd. as being in the range of £10-12 per pounf of steam per hour capacity. If the maximum weekly output is taken to be 40,000 tonnes of crude steel and the average steam usage per tonne is assumed, this may be converted to a required steam raising capacity of 265 t/h. The capital cost of the boiler estimated from this figure is then in the range £5.8 M to £7. M. This estimate agrees reasonably well with data on package boilers supplied by Davy McKee, Oil and Chemicals Ltd.

6.2 Coal Preparation

The cost of coal handling, drying and pulverising plant has been estimated using cost data generated for the Hunterston melter/gasifier study[263] which proposed a cost of £4.5 M for a plant to treat 500,000 t/a of coal. The requirement at Ravenscraig under the proposed strategy at a weekly output of 40,000 t crude steel would be 230,000 t/a. Scaling from the Davy McKee estimate using the formula:

$$\text{Cost Ratio} = (\text{Capacity Ratio})^{0.6}$$

gives an estimated cost for the coal preparation plant of £2.8 M.

6.3 Reheating Furnace Burner Conversion

The burner changes required on the reheating furnaces are to all the oil fired burners. The actual numbers of burners and their rated output are as follows:

No.1 Furnace	-	2 banks of 5 burners each of 8.6 MW
		2 banks of 5 burners each of 5.7 MW
No.2 Furnace	-	4 banks of 5 burners each of 5.7 MW
No.3 Furnace	-	1 bank of 8 burners each of 4.4 MW
		2 banks of 8 burners each of 5 MW
		1 bank of 8 burners each of 5.7 MW

Information from Babcock and Wilcox Ltd. suggests a guide price of £10,000 per burner independent of rated output in the range considered, plus £10,000 for each independent burner control system. It has been assumed that a bank of burners would be controlled together giving a capital cost as follows:

72 burners @ £10,000	£720,000
12 burner control systems @ £10,000	£120,000
Sub total	£840,000
Local Pipework & Installation @ say 20%	£168,000
Total installed cost	£1,008,000

6.4 Grant Aid

The capital investment required to implement the proposals could qualify for grant aid from at least two quarters.

Regional Development Grants would be available at the rate of 22% currently payable in Scotland.

The Department of Energy have a grant scheme for supporting energy demonstration projects to the extent of 25% of the capital cost, from a fund of £50 M set aside for this purpose. The DOE intend this scheme primarily for private enterprise companies, and would probably not ordinarily regard BSC as a suitable beneficiary. However, informal contacts with the Department of Industry have suggested that any application from BSC would receive support and encouragement from the DOI which could sway the DOE in our favour. Obviously further discussion with the DOE is required, and until an application is either approved or rejected there can be no certainty of financial assistance from this quarter, but the possibility should be borne in mind in consideration of the proposal. There may also be the possibility of some grant aid directly from the DOI towards the cost of boiler conversion from fuel oil to coal, but this would need to be investigated further to determine whether replacement of arising gases with coal, and fuel oil with arising gases would be deemed to qualify.

6.5 Summary of Capital Costs

The overall cost estimate for implementation of the scheme is as shown in the following summary:-

Boiler Replacement	£7.0 M
Coal Preparation Equipment	£2.8 M
Reheating Furnace Conversion	£1.0 M
Sub total	£10.8 M
Contingency @ 10%	£1.1 M
	£11.9 M
Less RDG @ 22%	(£2.6 M)
Total after RDG	£9.3 M
Less DOE Demonstration Grant if available @ 25%	(£3.0 M)
Total After all Grants	£6.3 M

7. CONCLUSIONS

Analysis of the fluid fuels usage at Ravenscraig Works over the fourteen month period April 1980 to May 1981 has shown that, there is an opportunity to replace purchased fluid fuels at Ravenscraig with coal and achieve significant cost savings.

It is proposed that the blast furnace stoves and coke ovens should be fired as far as possible on blast furnace gas, supplemented if required by coke ovens gas. Other process heating applications should be fired as far as possible with coke ovens gas and, if necessary to make up any deficit in CO gas, purchased fuels should be used. The steam raising plant should be fired on coal plus any arising gases surplus to requirements after the process heating load has been fully satisfied.

The main capital investments needed to implement this strategy are replacement of the existing boiler with a multi-fuel unit, installation of coal preparation equipment, and conversion of the slab reheating furnaces to multi-fuel firing.

The proposed strategy has been applied to the pattern of energy use over the fourteen month analysis period on a month by month basis, and this has revealed no major difficulties in achieving the required works energy balance. The operating cost saving projected for this period by application of the strategy is £11.6 M (£829 K per month, or £9.9 M per year.)

The estimated capital investment cost to allow the strategy to be implemented is £9.3 M after regional development grants had been applied, and if the scheme succeeded in obtaining grant aid from the Department of Energy this would be reduced to £6.3 M.

On the basis of the above operating and capital cost estimates the payback period on the project would be about one year without DOE grant aid, or about seven and a half months with the DOE grant. This must be considered a very attractive investment proposition, offering as it does a return on capital of 100-130%.

Thus it can be stated that through efficient technical management enromous savings in energy bill can be achived.

8.2.2 Organisational Management

The end products of management are "decisions and actions"
Peter Drucker.

Management of any business is a very large complex of activities which consists of analysis, decisions, communications, leadership, motivation, measurement and control (Fig. 8.3). Of these, the process of decision making can be considered as a corner-stone of a successful management.(264)

In the case of energy management, the problems are further enhanced because of the fact that investment in energy conservation may involve technology with which the organisation may be unfamiliar, so that its senior management may have to have an intuitive understanding of how the investment will lead to financial savings. The result is that industry tends to look more critically at energy conservation investments and set more stringent financial criteria. As an example it can be said that even in present day situation where it is recognised that fossil fuels are depleting fast and the energy costs are rising sharply the company management is still more interested in investments which strengthen or expand its manufacturing base than those which merely result in energy reduction. Any energy conservation investments have to have a payback period of less than two years where as the main stream activities investment would have a payback period of three to four years or even more in some cases. Add to this the fact that people have been used to cheap energy for too long and thus are not sure how to react to this change and achieve energy conservation, compounds the problem.

This part of the thesis described the complexities of setting up an effecient management organisation and then suggests a possible model for the energy management for one of the major users of energy i.e. the British Steel Corporation.

A. MANAGEMENT SCENARIOS

Pressures for reorganisation form part and parcel of the changing world in which we now live. There are changes in market conditions involving a greater rate of competition and of innovation, while commodity markets have become less settled. There is an increasing amount of Governmental intervention in organisational affairs. The employees are coming from different backgrounds, have higher levels of education and are developing new attitudes. Each of these broad areas of change create pressures for new structures of organisation and of work, because each is changing the contingencies within which management operates.

Before suggesting any Energy Management system for the BSC it is essential to look at the basic requirements of an effective organisation system, a highly complex system with no clearly defined rules. Therefore the thesis presents a range of scenarios which have been suggested by various workers in the field, and tries to draw on their experience to develop a BSC Energy Management Model.

In following a scientific approach the work has been looked at under the headings of:

1. Organisation Structure
2. Functions of an Organisation
3. Management of an Organisation
4. People in Organisation.

(i) <u>Organisation Structure:</u>

> "Structure is a means for attaining the objectives and goals of an institution", Peter F. Drucker.(265)

The structure of an organisation is often taken to comprise all the tangible and regularly occurring features which help to shape its members behaviour. There is no single way of organising. All the components of organisation structure can be designed to take different forms and they vary considerable in practice.

The one model of organisation with which we are most familiar is bureaucracy. Earlier contributors in this field of study was Max Weber.[266] His principle contribution was the theory of authority structure which led him to characterise organisations in terms of the authority relations within them. Based on this concept he believed that the decisive reason for the advance of bureaucratic organisation has always been its pure technical superiority over any other form of organisation.

Gouldner[267] has applied Weber's concepts of bureaucracy and its functioning to modern industrial organisations. In his analysis he distinguished three patterns of bureaucratic behaviour: mock, representative and punishment-centred; each with its own characteristics, values and conflicts. He maintains that there are unanticipated consequences of bureaucratic functions which Weber left out of account. General and impersonal rules, by their very nature, define what is not allowed and thus increase people's knowledge of what is the minimum acceptable behaviour, which tends to become more of a standard. This lowers efficiency and, in punishment centred bureaucracy, leads to increased closeness of supervision to see that the rules are carried out and as a consequence leads to an increased emphasis on authority and greater interpersonal tension. This results in the continued issue of impersonal rules to deal with the conflicts and thus the cycle begins again.

Etzioni[268,269] suggests that organisations are like other social units requiring compliance from other members which may be based on the members commitments to the social units aims and purposes. Organisations cannot rely on compliance coming essentially from the fact that members are completely committed to the aims of the organisation. Therefore they have a formal system of control and have a clear system of reward and penalty to ensure compliance from their members. As organisations cannot completely rely on their members to carry out orders perfectly, it is necessary to have a hierarchy of authority. To have supervisors it is necessary to have job descriptions and specified procedures for doing things and to have a division of labour. All of these attempts are to make the organisation less dependent on the whims of the individuals by controlling behaviour. The organisation thus exercises its power by these bureaucratic means.

Etzioni argues that business organisations function more effectively when they use remuneration rather than coercion as the basis of control.

Child[270] in his research shows that more profitable and faster growing companies are those that have developed this kind of organisation in fuller measure with their growth in size above the 2000 or so employee mark. In some industries such as steel, cement and glass there are large technological economies to be gained in increased scale of production; although plant size may be limited in practice by the desire to retain production flexibility.

Joan Woodward(271 272) in her analysis does not use the sweeping classifications of organisations but attempts to summate whole ranges of characteristics or organisations, e.g. the number of levels of authority between top and bottom, the space of control or average number of subordinates of superiors, the clarity or otherwise with which duties are defined, the amount of written communication to the extent of division of functions among specialists.

The basic assumption and conclusion of Woodward's work is that meaningful explanations of differences in organisations and behaviour can be found in the work situation itself. The technology of this work situation should be a critical consideration in management practice. There is not one best way and warns against accepting principles of administration as universally applicable. Her work shows that the same principles can produce different results in different circumstances. Therefore a careful study of the objectives and technology of a firm is required before setting up the structure of an organisation .

Burns(273) in advancing the view of Woodward suggests that for a proper understanding of organisational functioning it is necessary to conceive simultaneous working of three social systems. The first of these is the formal authority system derived from the aims of the organisations; second; its technology and the third; its attempt to cope with its environment.

Thus an organisational structure should embrace both its technical objectives and the social environments making it into a socio-technical system.[274 275]

Child[270], in summarising, states that organisation structure is a means for allocating responsibilities, providing a framework for operation and performance assessment, and furnishing mechanisms to process information and assist decision-making. There are many alternative structural designs to choose from. This choice is not simply a technical matter but also reflects the contingencies such as the organisation's dominant culture, size, environment etc.

The activities of people within an organisation can be grouped together in a number of ways.[276] Modern thinking on job design has a considerable bearing on the old debate over the pros and cons of tall versus flat shapes of organisations. For it suggests that trends in organisational development should be towards a broadening of managerial spans of control and a consequent flattening of organisational hierarchies.

Tall and flat organisation structures are usually identified by the number of hierarchical levels there are in the organisation relative to its total size. A tall structure is one that has many levels relative to total size, while a flat structure is one that has few levels relative to total size. Managers face the problem of maintaining balance between the number of hierarchical levels in an organisation and the span of control of managerial and supervisory staff. The trade off between tallness and flatness in the shape of formal authority structure becomes a particularly acute problem for the large organisations in which both hierarchies and spans of control may become extended beyond the optimum. Growing organisations often find themselves multiplying hierarchial levels in an attempt to avoid increasing the burden of supervision faced by individual managers. An extended hierarchy brings considerable disadvantages of administration overheads, communication failure and low motivation among those removed from sources of major decisions. Long hierarchies based on very narrow spans of control may offer people little opportunity for personal discretion and initiative in their jobs.

Worthy[277] that flatter, less complex structures with a maximum of administration decentralisation tend to create a potentially improved attitude, more effective supervision and greater individual responsibility among employees.

In deciding on the question of shape, i.e. spans of control, especially in technology related manufacturing firms, Woodward's[278] research shows that the following factors should be considered;

1. The degree of interaction between the personnel, or units of personnel being supervised.
2. The degree of dissimilarity of activities being supervised.
3. The incidence of new problems in the supervisor's unit.

4. The degree of physical dispersion of activities.

5. The extent to which the supervisor must carry out non-managerial duties and the demands on his time from other people and units.

The greater the incidence of these factors the heavier is likely to be the burden of supervision, and hence more severe the limit on the number of subordinates a person can manage.

The activities of people within an organisation can be grouped together according to a number of different principles. A functional grouping comprises people employing a similar activity. A grouping by process recognises common abilities in plant and technology employed. A product grouping brings together people who are contributing to a common product or service. Considerations which enter into the choice very much depend on the size of the organisation, the diversity of its activities and the speed at which it must adapt to its environment. Galbraith[279] in his book discusses various organisation designs. The two most common models for grouping activities are the functional and product division type. The virtues of low overheads and simplicity characterising the functional model are balanced by the co-ordinative, motivational and adoptive advantages of the product model. The matrix developed structure often secures the best of both models.

Inadequate co-ordination between the different department and specialist staff of an organisation is often a cause of poor performance.[280] Integration is a problem that develops with the growth of an organisation and their subdivisions into separate subunits. It is exacerbated by the variations in outlook between people trained in different functional disciplines and by the conflict between specific criteria performance which are attached to separate departments. Sykes[281] study shows that points at which poor co-ordination is commonly found include the relations between sales and production, between research and other functions, and between personnel and technical officers over the job priorities. Through a range of devices such as face-to-face meetings, an improved integration in an organisation at a time cost can be achieved.

Control is an element of management which flows down from the apex of hierarchies to the shop floor. The main organisational design decisions concerned with control are:

(i) how much to formalise procedures and working practices, and

(ii) how much emphasis to place on direct supervision.

The usual approach to delegation involves a formal specification of the limits to decision making, i.e. centralised or decentralised structures.

Professor Carlisle[282] distinguished thirteen variables which are of primary importance in determining the need for a centralised or decentralised structure:

1. The basic purpose and goals of the organisation

2. The knowledge and experience of top level managers.

3. The skill, knowledge and attitude of subordinates.

4. The scale or size of organisation.

5. The geographical dispersion of the organisation.

6. The scientific content/technology of the task being performed.

7. The time frame of the decisions to be made.

8. The significance of the decisions to be made.

9. The degree to which the subordinates will accept and are motivated by the decisions to be made.

10. The status of the organisation's planning and control system.

11. The status of the organisation's information system.

12. The degree of conformity and co-ordination required in the tasks and operations of the organisation.

13. The status of external environmental factors such as Governments, Trade Unions etc.

On the whole the structural approaches which lend themselves to good performance appear to vary considerably with the type of organisation and contingencies it faces.

(ii) Functions of an Organisation

> "To manage is to forecast and plan, to organise, to command, to co-ordinate and to control".
>
> Henri Fayol.

An organisation must start with a plan, a definition of its goal. The task of its management is to build up an organisation which will allow the basic activities to be carried out.

It is sometimes asserted that the functioning of each and every undertaking is unique. However, most people recognise that there is some point in comparing their own organisation with others and learning from the comparison.

Fayol[(283)] offers a doctorine of good management based on his own experience emphasising that these are his own rules and necessarily are not of universal application. Nonetheless, most have become part of managerial know how and many are regarded as fundamental tenets. They can be outlined as follows:

1. Division of work: specialisation allows the individual to build up expertise and thereby be more productive.
2. Authority: the right to issue commands, along with which must go the equivalent responsibility for its exercise.
3. Discipline: which is two-sided, for employees only obey orders if management play their part by providing good leadership.
4. Unity of command: in contrast to Taylor's functional authority[(284)] Fayol states that each man should have only one boss with no other conflicting lines of command. On this issue Likert has favoured Fayol, for his principle has found most adherents amongst managers.
5. Unity of direction: people engaged in the same kind of activities must have the same objectives in a single plan.

6. Subordinate individual interest to general interest: Management must see that the good of the firm always paramount.

7. Remuneration: payment is an important motivator, though not the only motivator.

8. Centralisation/Decentralisation: is dependent on the condition of business and the quality of its personnel.

9. Scalar chain: a hierarchy is neccesary for unity of direction but lateral communication is also fundamental as long as supervisors know that such communication is taking place.

10. Order: both material order and social order are necessary. The former minimises the lost time and useless handling of materials. The latter is achieved through organisation and selection.

11. Equity: in running a business, a 'combination of kindliness and justice' is needed in treating employees if equity is to be achieved.

12. Stability of tenure: this is essential due to the time and expense involved in training good management. Fayol believes that successful business tend to have more stable managerial personnel.

13. Initiative: allowing all personnel to show their initiative in some way is a source of strength for the organisation even though it may well involve a sacrifice of 'personal vanity' on the part of some managers.

14. Esprit de corps: management must foster the morals of its employees and to quote:

    > 'real talent is needed to co-ordinate effort, encourage keeness, use each man's abilities and reward each one's merit without arousing possible jealouses and distributing harmonious relations'.

Brown (285) breaks away from the kind of analysis initiated by Fayol which describe management as a mixture of elements such as forecasting, planning, organising. Brown is less concerned with the nature of manager's activities as such than with the social organisation, or set of social systems, through which managers work.. This fundamental tenet is that a concious recognition of these social systems will promote good management. Brown contends that 'there seems to be quite a considerable tendency to construe all problems in industry in terms of the personal behaviour of people and to exclude the notion that we can design trouble into, or out of, the executive system'. Thus people blame difficulties on to personalities of others or their own, seldom stopping to think that the difficulty may result from the design of the social system of which their own roles are a part.

His insistence on detached analysis using these concepts lead him to conclude that:

> "Effective organisation is a function of the work to be done and the resources and techniques available to do it".

Bakke's (286) work on organisation theory is mainly focused on the problem of developing concepts, and meaningful words to denote them, with which to define and analyse organisations and their activities. This confronts him with a task of reducing the seemingly endless diversity of forms of human social organisations to some kind of common elements.

He breaks them down into four sub. categories:

1. Directive activities: being those which initiate action, such as determining what shall be done and to what standard, and giving instruction.
2. Motivation activities: rewarding or penalising behaviour.
3. Evaluation activities: reviewing and appraising people's performance, comparing alternative courses of action.
4. Communication activities: providing people with premises and data they need.

These four activities can also be defined as the fusion process, the problem solving process, the leadership process and the legitimisation process.

In the 'fusion process' Bakke defines that an organisation attempts to shape its own image on all the individuals who join it; and an individual who joins the organisation likewise tries to express his own personality by shaping the organisation accordingly, thus each experience some change. However there are times when the organisation and its members are mutually opposed. Hence the need for fusion or 'fuse' organisation, groups of individuals. The continual solving of non-routine problems in an organisation may be termed as 'problem solving process'. Bakke sets out in his contribution to the symposium edited by Mason Haire[(287)] what he believes to be a logical sequence of steps normally taken in problem solving and in 'leadership process' providing imagination and initiative. Finally he defines the 'legitimisation process' as activities to justify and get accepted the ends of the organisation and what it does to pursue them.

In any organisation, decisions depend on information, estimates and expectation. Cyert and March(288) in theory of 'decision-making' show that far from having perfect knowledge, organisations appear to act on very small portions of total available information. In addition, organisations are generally composed of different departments with conflicting interests. So in arriving at a decision-making the firm is constrained by its problems of internal management co-ordination, uncertainty of its external situation or environment or by its own limitations in assembling, storing and utilising of information.

The decision process concerns three basic concepts of orgnaisation i.e. organisational goals, organisational expectations and organisational choice. That is to say: what shall the objective be, what is expected to happen and what consequences are anticipated from actions which could be taken and what action shall be chosen.

In summary, organisational goals change in response to sub goals or interests of those who form the coalition, to a minimal level of what will be accepted all round, after restricted examination of a limited and selective range of information. In this way a full complexity of decision making is reduced to what is practicable, and uncertainty is absorbed. Organisational expectations of what may happen are likewise confined to a number of possibilities, few enough and familiar enough to be practicable.

Organisational choices are made from among the resulting limited alternatives.

(iii) Management of An Organisation:

> "Scientific management will mean, for the employers and the workmen who adopt it, the elimination of almost all causes for dispute and disagreement between them".
>
> Frederick W. Taylor.(289)

Organisations with differing structures, functioning in different ways have to be administered or managed. As long as there is management there will be the problem of how to manage better. Many researchers have sought ways to improve the understanding of administration.

Taylor's name is synonymous with the term ' scientific management'. The principle object of managment' he states should be to secure the maximum prosperity for the employer, coupled with the maximum prosperity for each employee . For the employer, 'maximum prosperity means not just large profits , in the short term but the development of all aspects of that enterprise to a state or permanent prosperity.' For the employee, 'maximum prosperity' means not just immediate higher wages, but his development so that he may perform efficiently in the highest grade of work for which his natural abilities fit him. Taylor found that what the workman wants most of its employer is high wages and what employer wants most from its worker is the low labour cost of manufacture. The existence or absence of these two elements forms the best index to either good or bad management. To achieve good management he lays down four basic principles:

1. The development of a true science of work: workers should be paid according to the task completed, having established a day's norm.
2. The scientific selection and progress development of workmen: It was the responsibility of management to develop workers, offering them opportunities for advancement which would finally enable them to do 'the highest, most interesting and most profitable class of work for which they would be best rewarded.
3. The bringing together of the science of work and the scientifically selected and trained men: In this Taylor maintains that almost invariably the major resistance to scientific managemenet comes from the side of the management. The workers he finds are very willing to co-operate in learning to do a good job for a high rate of pay.
4. The constant and intimate co-operation of management and men:

Taylor's methods have been followed by many others e.g. Grantt, Frank & Lillian Gilbreth, Beandex, etc who have developed his thinking into what is now called 'Work Study'.

Follett[290] in her writings attempted to provide an outlook on management in which organisation, leadership and power are dealt with as human problems. Her fundamental principles of organisation were:

1. Co-ordination by direct contact; regardless of their position in the organisation. 'Horizontal' communication is as important on 'vertical' chains of command in achieving co-ordination.
2. Co-ordination in the early stages:
   The people concerned should be involved in policy or decisions while these are being formed and not simply brought in afterwards. In this way the benefits of the participation will be obtained in increased motivation of morale.
3. Co-ordination as the 'reciprocal relating' of all factors in a situation:
   All factors have to be related to one another, and these inter-relationships must themselves be taken into account.
4. Co-ordinating as a continuing process:
   'An executive decision is a moment in a process'. So many people contribute to making a decision that the concept of final or ultimate responsibility is an illusion. Combined knowledge and joint responsibility take its place. Authority and responsibility should derive from the actual function to be performed, not from place in the hierarchy.

The basis for Follett's thinking is the concept of partnership. The case of her contribution is the proposition that in a democratic society the primary task of a management is so to arrange the situation that people co-operate readily of their own accord.

For Simon (291 292) management is equivalent to 'a decision-making' and his major interest has been an analysis of how decisions are made and how they might be made more effectively.

He described three stages in the overall process of making a decision.

1. Finding occasions calling for a decision: (Intelligence)
2. Inventing, developing and analysing possible courses of action: (Design)
3. Selecting a particular course of action from the available: (Choice)

On what basis does the administrator make his decision one asks? The traditional theory of economists assumed complete rationality. But we know that there is a large non-rational emotion and unconscious element in man's thinking and behaviour. 'It becomes the task of management' says Simon, 'to design an environment where the individual will approach as close as practicable to rationalise. (judged in terms of the organisation's goals) his decisions.

Most human decision-making whether individual or organisational, is concerned with the discovery and selection of satisfactory alternatives; only in exceptional cases is it concerned with the discovery and selection of optimum alternatives. Therefore Simon supports the theory that programmed decision making will be more effective than the individual decision making.

Drucker's[293,294] work reviews the top management and its critical role in representative institutions of modern industrial society, namely the large corporations. Following from this he identifies management as the central problem area and the management as the dynamic element in every business. It is the manager through his control of the decision making structure of the modern corporation who breathes life into the organisation and society. Yet managers, while becoming ever more basic resources of a business, are increasingly the scarcest, the most expensive and most perishable. Given this, it becomes increasingly important that they are used effectively.

In assessing managerial effectiveness, Drucker shows that managers are first judged on economic effectiveness and secondly on decision making. Taken together, it means that managers are evaluated in their performance in the present, the short term and long term.

Management, therefore, is the job of organising resources to achieve satisfactory performance and to produce enterprise from human and material resources. According to Drucker this does not mean profit maximisation. For him profit is not the cause of business behaviour or the rationale of business decision making in the sense of always attempting to achieve the maximum profit. The aim of any business is to achieve sufficient profit, which will cover the risk that has been taken and avoid a loss.

The central question for Drucker is how best to manage a business to ensure that profits are made, and that the enterprise is successful over time. This he explains can be achieved through the 'management by objectives'.

Objectives in a business enterprise management are to explain predict and control activities in a way which single ideas, such as profit maximisation, do not. But perhaps the most important part of management by objectives is the effect it has on the manager himself. It enables the organisation to develop its most important resource management. This is because managerial self-control is developed, leading to a stronger motivation and more efficient learning. The importance of individual managerial involvement in setting of objectives cannot be over-stressed as a motivator.

Management by objectives is a way of overcoming deficiencies by relating the task of each manager to the overall goals of the company. Management is a group operation and the existence of objectives emphasises the contribution that each individual manager makes to the group operation.

Overall then, management by objectives helps to over come some of the forces which threaten to split the organisation by clearly relating the tasks of each manager to the overall aims of the company. It increases the motivation of the manager and develops his commitment to the organisation. The result is that organisational goals are reached by having common people achieve uncommon performance.

The recurrent theme in Sloan's [295] book is the necessity of dealing with the major problem which forces any large multioperation enterprise i.e. the appropriate degree of centralisation or decentralisation of authority for decisions. The centralising approach has the advantage of flexibility and perhaps speed but places an enormous weight on the top man. The decentralisation approach has the advantage of allowing decisions to be made closer to the operational bases of enterprise but then runs the real danger that the decisions will be taken with regard to the best interests of the particular operating division itself without concern for the best interests of the Corporation as a whole. Sloan goes on to illustrate, with examples from General Motors, that centralisation clearly offers great advantages and systems of co-ordination for purchasing, corporate advertising, engineering and so on.

Top management, according to Sloan, has the basic tasks of providing motivation and opportunity for its senior executives i.e. motivation by incentive compensation through stock option plans, and opportunity through decentralised management. But co-ordination is also required; and good management rests on a reconciliation of centralisation and decentralisation. His basic view was that policy co-ordination is achieved through committees. It is evolved in a continuous debate to which all may contribute and is basically an educational process. Executive administration is the clear responsibility of individuals who carry out the evolving policy.

(iv) People in Organisations:

Organisations are systems on interdependent human beings. The behaviour of the members of the organisation clearly affects both its structure and its functioning, as well as the principles on which it can be managed. Most importantly human beings affect not only the methods used to accomplish them but even the aims of organisations in which they participate.

Mayo has often been called the founder of both the Human Relations movement and industrial sociology. Mayo's[296] generalisation was that work satisfaction depends to a large extent on the informal social pattern of work group. Where norms of co-operativeness and a high output are established because of a feeling of importance, physical conditions have little impact.

Herzberg[297,298] and his colleagues' major findings of a survey showed that events that led to satisfaction were of quite a different kind from those that led to dis-satisfaction. Five factors were high lighted as strong determinants of job satisfaction:

a) Achievement
b) Recognition
c) Attraction of work itself
d) Responsibility

and e) Advancement

The reasons for dissatisfaction can be classified as:

a) Company policy and administration

b) Supervision

c) Salary

d) Inter-personal relations and working conditions.

A closer examination showed that these two feelings are not the opposite of one another but are rather concerned with two different ranges of man's needs.

Herzberg advocates that instead of rationalising and simplifying the work to increase efficiency, as the motivation-hygiene theory suggests the jobs should be enriched to include the motivating factors in order to bring about an effective utilisation of people and to increase the job satisfaction.

McGregor[299] in the 1950's enunciated two sets of propositions and assumptions about man in the organisation.

<u>Theory X</u>

1. The average man is by nature indolent - he works as little as possible.
2. He lacks ambition, dislikes responsibility, prefers to be led.
3. He is inherently self-centred, indifferent to organisational needs.
4. He is by nature resistant to change.
5. He is gullible, not very bright, the ready dupe of the charlaton and the demagogue.

Therefore the implications for management are:

1. Management is responsible for organising the elements of productive enterprise - (money, materials, equipment, people) in the interest of economic ends.
2. With respect to people, this is a process of directing their efforts, motivating them, controlling their actions, modifying their behaviour to fit the needs of the organisation.
3. People must be persuaded, rewarded, punished, controlled; their activities must be directed.

<u>Theory Y</u>

1. People are not by nature passive or resistant to organisational needs. They have become so as a result of experience in organisation.
2. The motivation, the potential for development, the capacity to assume responsibility, the readiness to direct behaviour towards organisational goals are all present in people. It is the responsibility of management to make it possible for people to re-organise and develop the human characteristics for themselves.
3. Management is responsible for organising the elements of productive enterprise in the interest of economic ends, and methods of operation so that people can achieve their own goals best by directing their own efforts towards organisational objectives.

The work of McGregor (300) and Likert (301) concludes that modern organisations, to be effective, must regard themselves as interacting groups of people with supportive relationships to each other. In the ideal, each member of the organisation will feel that its objectives are of significance to him, that the job is meaningful, indispensible and difficult, that to do it effectively he needs and obtains support from his superiors - who regard the giving of this support to make him effective as their prime function.

B. PROPOSED MODEL: BSC ENERGY MANAGEMENT

(i) INTRODUCTION

> "The beginning of administrative wisdom is the awareness that there is no one optimum type of management".
>
> Tom Burn

The problem of energy conservation cannot simply be resolved by technical solutions. In the case of BSC, which has considerably reduced its energy bill from 27 GJ/t liquid steel in 1979 to 23 GJ/t liquid steel in Feb.1981, it can be seen that the most of gains have come from improved plant loading and from short-term energy reduction actions at the plant. However these achivements probably represent less than two thirds of the total short-term possibilities and probably less than one half of the total long and short term potential. This shows that the energy used in industry is as much influenced by the attitudes of management and employees as by the efficiency of machinery and production techniques.

There are various types of organisational structures one can adopt as discussed earlier in the chapter.

It is suggested that to achieve an efficient energy conservation programme it is essential to develop a strong management policy based on traditional concepts e.g

(a) Evaluate the possibilities

(b) Organise a management structure and allocate responsibilities

(c) Set objectives

(d) Monitor performance

(e) Regular review

Spray[302] suggests that to arrive at an effective management structure one must ask:

1. What alternative ways of evaluating energy management performance are conceivable?
2. What alternative ways of evaluating energy management performance actually have been utilised?
3. What are underlying causes of various levels of performance?
4. What are the consequences of various levels of energy performance on other processes?
5. What advantages/disadvantages are inherent in various approaches to evaluating managment performances?

In the BSC, each energy manager is responsible for its own works energy management, though does not have authority to communicate directly to the director level. In addition, similar to process improvement committees, a working party exists in energy management, on the corporate level, which consists of energy managers of all integrated works together with some energy managers from research; called Energy Utilisation Committee (EUC).

However the committee has only an advisory role which sometimes can slow down the progress of implementation of schemes in works even though they may be based on sound energy management basis.

Therefore the major problems in mounting a meaningful programme of work in the Corporation can be summarised as follows:

1. Corporate energy functions are neither clearly defined nor understood.
2. Effective energy management structure at both works and corporate level does not exist. The Energy Utilisation Committee (EUC), which comprises of fifteen members representing both works and research orgnaisation, lacks authority to implement any energy saving schemes at works. In addition inconsistencies in works energy statistics and a lack of detailed works energy programme further hinders its progress.

   Divided responsibilities of the members results in an energy organisation which is unlikely to exert sufficient influence over the works to sustain continual improvement in energy consumption.
3. Separate energy monitoring/control system does not exist at Division/Corporate level

4. Whilst works energy managers understand their responsibilities for efficient utilisation of energy at their works, it is difficult to identify responsibilities for inter-works issues relating to energy, except where specific projects have been established.

5. Not only are the resources scarce but the role of the energy manager is diffused i.e. he has both advisory and monitoring duties but no responsibility for the level of energy used.

6. Each works has a different structure tradition and management style.

In order to overcome these problems and thus achieve an effective corporate energy policy the following 'Energy Management Model' is suggested.

BSC: "ENERGY MANAGEMENT MODEL"

1. Create an Executive Energy Function

Most large companies and many of the smaller ones have been resorting to decentralisation for sometime now. The organisational preference represents the effort to reduce the burden of the chief executive and partly an attempt to pass on the delegation of authority and responsibility for decision-making to as low a level in the management hierarchy that it can support.(303)

The corporation with its present policy of decentralisation is trying to establish itself into manageable units. Establishing an energy management function in each of the two major businesses with an Energy Executive reporting directly to the Chief Executive of the business unit can tie the energy management to business planning. To achieve an effective management it is essential that strong, senior executive management is recruited which has professional and personal credibility to command the respect of the works directors, as the control of energy costs and execution of energy improvement action plans still will be the works responsibility. More important, they should have the necessary business perspective and ability "to make things happen".

Their major responsibilities could be

- Define the format and content of works energy improvement action plans including objectives, level of details and horizon.
- Monitor both current energy utilisation and adherence to energy acts on plans and recommend steps necessary to maintain progress to works operating management.
- Challenge current standards and stimulate improvement through interworks and international comparisons, optimising the use of existing talents through cross-fertilisation. Through discussion on long term supply contracts develop energy contingency plans for alternative supplies in case of supply interruption.

A suggested organisation structure for energy management is presented in Fig. 8.4.

2. Energy Monitoring at Corporate Level

In order to provide a single, central focus for energy management in the Corporation, an approach which has been successful in many other organisations abroad, and to allow the Managing Director/Chairman of the company to retain the role of prime instigator of change in the energy policy scene, energy function at the corporate level should be strengthened.

The function, thus established, should be led by a senior executive whose responsibilities may be defined as:

- Provide specific information and advise on current world best practise in energy.
- Co-ordinate and assist with the development of energy action plans for the two major business units.
- Develop the energy action plans for tubes and other speciality profit centres.
- Monitor the progress of the corporation towards achieving its energy targets.
- Chair the Energy Utilisation Committee which should review and discuss:
  - technical subjects relating to energy implications of plant operations.
  - results of trials of research in progress
  - energy performance by works
  - inter-plant comparisons on energy matters

It is considered that the corporate function should be largely advisory and not executive, since it is neither realistic nor desirable for it to take full responsibility for the works energy functions; rather the function should ensure that the business units tackle energy management and planning in a consistent and effective way.

3. Develop detailed Energy Improvement Plant at each Works

In order to remain energy effective, the corporate energy function should support and co-ordinate development of energy conservation plans for various works and ensure that inter-works and international company comparisons are undertaken and the information resulting is disseminated. In assessing 'limiting' energy savings potential against world leading practices e.g. Japanese, areas of greatest leverage and cost/benefit analysis of projects necessary to reach world best performance levels should be considered.

Specific plans detailed sufficiently to support tight progress control should be drawn up for the short - and medium term and these should include:

- Tasks, resources required, time scales, reporting milestones and responsibilities.
- Resources needed for developing longer-term plans
- Costs and benefits for both capital and house-keeping projects.
- Priorities allocated in line with cost-benefits and resources availability.
- Revised objectives and standards in line with a time phased execution plan.
- Commitment from those responsible for implementation

In defining three to five years plans results should be based on the 'best world practice'. Capital expenditure, positive cash flows associated with each project should be phased overtime, and prioritised for approval.

Finally, a detailed investigation should be carried out to finalise a longer-term strategy for energy reduction and the long-term implications of any short or medium term project decisions should be assessed and clarified.

4. Effective Works Energy Management

The real savings in energy are only to be made at the works level in the end. Therefore, it is essential that an effective works energy function is established for all steelworks on the model suggested below:

- The function should operate at an equal level to Works Manager and Chief Engineer, directly reporting to the Works Director.
- The energy function should be managed by experienced executive, working full time on energy matters and exercising sufficient influence to ensure that energy is given a degree of consideration commensurate with its effect on overall cost.
- He will be responsible for ensuring that energy plans are drawn (short, medium and long) prepared, co-ordinated, agreed and ultimately executed and is done in line with the overall company objectives.
- In the 'ship' style organisation where responsibilities remain decentralised, his responsibility for the overall level of energy consumed must be shared with

individual managers and with the Chief Engineer who has responsibility for power services. However, he should have ultimate responsibility for energy reporting and monitoring and by his direct access to the Works Director should be able to activate as well as monitor.

- The energy function should be created and positioned in the current organisation so that maximum use can be made of existing support staff - clerical and technical - and so that project resources currently under the control of the Chief Engineer can be made available for energy tasks with minimum disruption to current reporting lines.
- The energy executive should be supported by the current Energy Manager working full-time and some clerical support, which might be shared depending on the workload. Other resources will need to be assigned during preparation of plans and they should be assigned in project mode to the energy function.
- Works energy staff - the existing complement of fuel technologists should continue to report ultimately to the respective Works Manager, but should have dotted line relationship to the Energy Executive to ensure he is adequately appraised of the energy situations in the works. The same should apply to energy staff currently reporting in line to the Chief Engineer.
- Resources needed to plan and control energy should be drawn on a project basis from their functional departments e.g. R & D, Engineering, as they currently

operate. For the duration of a specific assignment on energy, these staff should report directly to the Energy Executive.

- Projects of a capital nature, with energy implications, should have a dotted line reporting relationship to the energy function so that progress is reported and objectives and standards can be revised.
- In such a streamlined form, the new function can operate in a task force mode of operation and, when appropriate, can be easily placed into the works technical organisation.

5. DEVELOP AND INSTALL IMPROVED MONITORING AND CONTROL PROCEDURES

- A more structured and consistent approach to energy monitoring and reporting should be developed at works level without becoming too bureaucratic.

- Performance measurement should be unambiguous and clearly indicative of trends and variances.
- Standards, flexed and unflexed, should be distinguished from objectives and targets.
- Progress reports on capital and housekeeping energy projects should show estimated and target completion dates, status, responsibilities and resources required.
- Energy costs should be included and highlighted.
- Data validity should be addressed as a priority.

- A standard and concise format for reporting to division and corporate levels should be developed; ideally simple extracts from the works reports could form a comprehensive and co-ordinated package on information without requiring extensive systems and procedural support.

- Uniform procedures for flexing standards at the works level should be agreed with the Corporate Energy Executive and applied to all performance reports.

- Statistics provided by the Corporate Energy function should serve a dual purpose in providing.
  . Corporation management with information on energy trends and targets and current progress.
  . A basis for constructive comparisons between various BSC plants and other companies.

- Procedures for collection and provision of such data should be geared to these specific purposes and should be co-ordinated with works level control and procedures.
  . Performances need to be detailed at a process level
  . The scope of what is included in each statistic needs to be agreed and made consistent with measurements at the works
  . The basis of central statistics (eg. net or gross calorific value) need to be made consistent with works conventions
  . Energy costs should be included in the reporting format.
  . Performance, and trends, should be compared with the "limiting" energy potential defined for each works

and intermediate objectives and standards should be quoted.

It is recognised that the magnitude of task required to bring the BSC energy consumption levels down to the best competitive levels are enormous but strengthening the energy leadership through the Chairman and establishing strong energy management functions at both Works and Corporate level, even from the limited technical and managerial resources available to the Corporation, can offer significant energy savings.

It is concluded that establishment of a strong energy management function based on the suggested model should offer the Corporation a realistic assessment of the potential energy improvement together with a plan defining the timing and resources required to achieve and sustain it. The model will also offer an effective mechanism to monitor progress at works, division and corporate levels.

(iii) ENERGY AWARENESS

Creation of a management structure policy alone does not ensure a smooth running of energy management.

> "'Boss' Kettering of General Motors is reported to have said that the hardest substance in the world is the human skull, to judge by the force needed to drive a new idean through it".

Energy has been so cheap for so long that even though energy prices have escalated many fold since 1973, people on the shop floor (management of workers) still are not registering the urgency of energy conservation.

Works are still operating on production target basis. Milner[304] explains this on the theory of human contstraints.

Only through the committment of senior management can a right climate be created for the workers to become aware of the need to conserve energy[305]

There are a number of factors through which energy conservation awareness can be improved. A wall board chart as shown in Fig. 8.5 can serve a useful starting point. Other factors that must be considered are:

Motivation

In a free competitive economy, a company's only long term advantage lies in its human resources. Other advantages that arise from technological improvements, the opening of new markets and lower material or labour costs all prove to be relatively short-term. So basically, it is the initiative, the will, and the motivation people bring to their work, that companies are learning to rely upon for its survival and growth[306]

To nurture this motivation, more and more companies are seeking fundamental elements to provide just the right climate for its growth. Part of this search stems from the realisation that money alone is not enough to buy motivation.

From a general motivation point of view a number of guide lines emerge. Mars [307] summarises the work of writers offering perceptive suggestions for productive climate in R & D. They confirm the view that motivation is a more complex variable than the simple reward/punishment system though it is an important feature. Individuals need for autonomy, freedom, association, status and achievement seem also to be significant.

Orth, Bailey & Wolck [308] discuss the behaviour of scientists and engineers in organisations and discuss a number of motivating factors. In general there is a strong emphasis on the pattern of job as dictating behaviour and hence the motivation[309,311]

Herzberg et al [312] developed the concept that job enrichment offer motivation which has been further supported by Paul and Robertson (1970)[313] through extensive experiments using this technique within I.C.I.

Career development and environmental factors can further motivate people.

In Japanese industry, Ishihara[314] states that the concept of Jishan Kanro (JK) activities is used which states that 'Man has initial ability to create and desires to challenge higher goals (Fig. 8.5) However it is accepted that the concept can only be applied to Japan because it is typical of the Japanese mentality.

Communication

Once the broad policy on energy conservation has been established, this must then be transmitted, understood and agreed by all the line managements concerned, which can only be achieved through an effective communication system. Drucker[315] states that communication is a total involvement of the people concerned. Through good communication one demands that the recipient become somebody or do something or believes something but always appeals to his motivation. In other words, if the communication fits with the aspirations, the values and the purpose of the recipient, it will be well received.

Therefore, in effective communicating, whatever the medium, the question should be asked "What is the communication within the recipient's range of perception and does it meet his motivation."

In arriving at an energy conservation programme it is essential that all departments have an appreciation of the total situation as well as their own contributions. Improved communications within dpeartment or within works or even in inter-works on energy matters can avoid waste through matching of operations.

Commitment of Accountability

It is absolutely crucial that before any major company wide energy conservation programme is begun, there is a clear and determined commitment starting at board level and continuing all the way down the line. Roberts[316] states that the minimum acceptable level of commitment can be related to the percentage reduction in consumption to be achieved within a stated length of time.

Every organisation is required by now to maintain financial accounts and many extend this function further to produce cost accounts. The latter are detailed statements of exactly where costs, including energy costs, are incurred. The system can be improved to provide energy costs as a percentage of the total cost or per unit. Experience shows that merely telling the manager his energy costs can have dramatic impact. Generally the managers and the operators had been aware where the energy could have been saved. Only through holding them accountable can this energy be saved.

Following this study a BSC AWARENESS MODEL is developed which incorporates above points in a visual form.

AWARENESS MODEL

It is suggested that to motivate workers to a better awareness of the needs to conserve energy in the steelworks an effective method of communication should be established. This could be based on a similar mechanism to that of safety.

The suggested MODEL is presented below:

1. The management should create an environment in the works where management commitment to energy savings is clearly demonstrated.
2. A board based on Fig.8.5 displaying the cost of energy used/t produced for each of the processes to be installed at the entrance of steelworks gate for all to take note of.
3. The board should also display the energy used in best

world practices for that process and thus offer a target to be achieved.

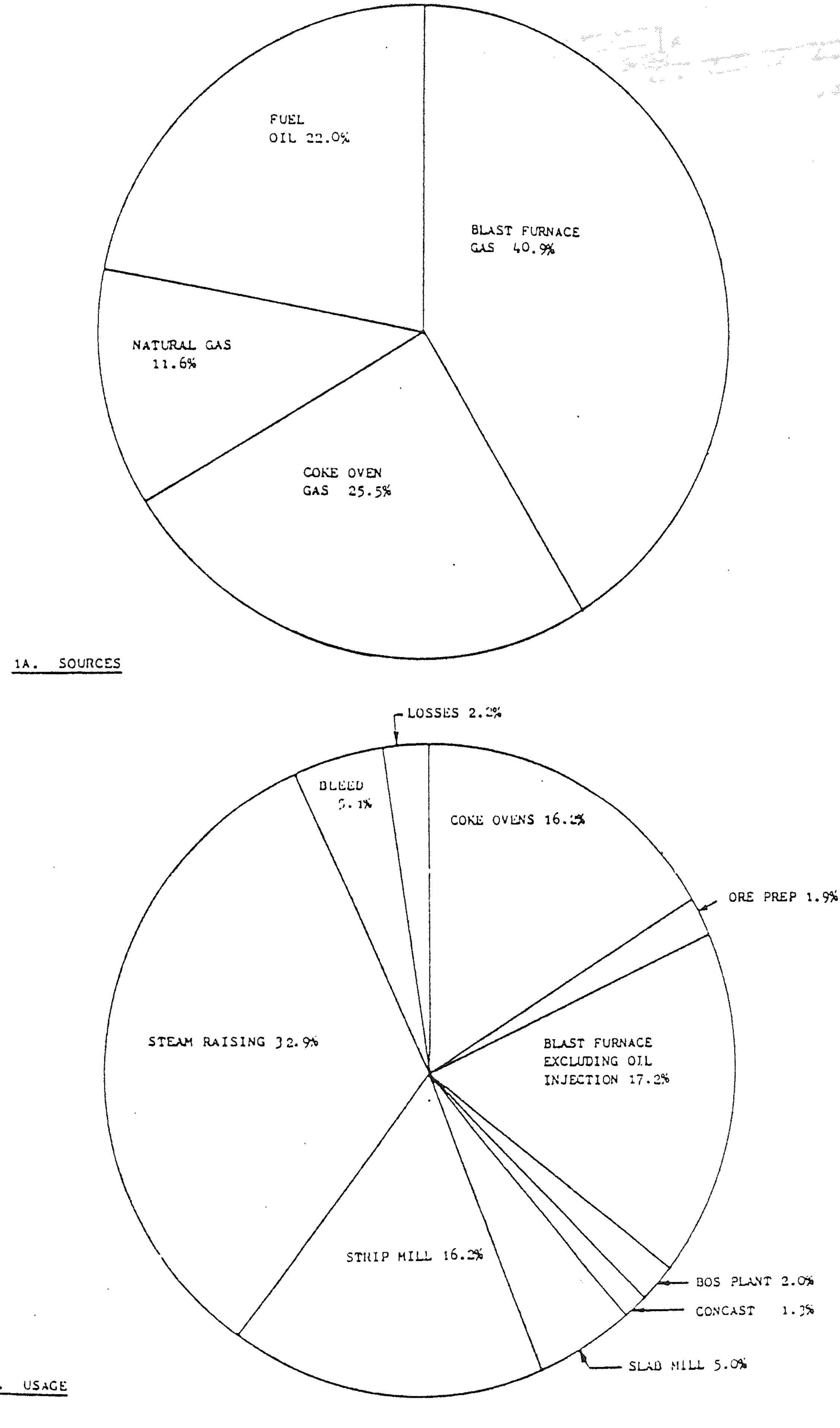
4. The progress of energy savings in pounds should be monitored on the board on a weekly basis.
5. An effective energy monitoring/control system should be installed in the works.
6. To motivate people to have good housekeeping, an area where major energy savings can be achieved at a minimum of cost, a reward/punishment mechanism should be developed based on an individual or group system.

8.3 CONCLUSIONS

The British Steel Corporation, a major consumer of energy, is already achieving some success in its energy consumption. But most of this saving comes from bringing on stream newer, more efficient plants, eliminating energy intensive processes like open hearth furnaces, closure of obsolete plants, maximum use of scrap and high grade iron ore and through improved plant loading. However these achievements represent some less than half of the total long and short term potential.

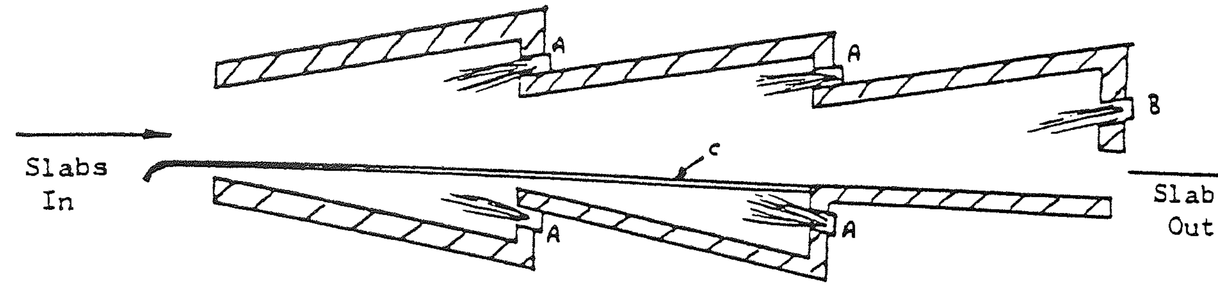
This chapter outlines the values of the effective utilisation of arising gases in works linked with a reassessment of the need of in house power generation and shows that the bought in liquid fuels for the processes can be substantially reduced (in some integrated steelworks it can even be balanced). Cost savings can also be made not only through energy conservation but through an effective buying strategy of bought in fuels e.g. which fuel at what time.

Major energy savings can be achieved through the establishment of a strong energy management structure and corporate policy at a relatively low cost. The thesis suggests a possible model on which management structure could be based. It also suggests a mechanism through which everyone in the works can be made aware of the need to conserve energy and how they can be motivated to achieve savings through good housekeeping at relatively low cost. The chapter concludes that energy management by objectives Fig. 8.7 can offer substantial energy savings.

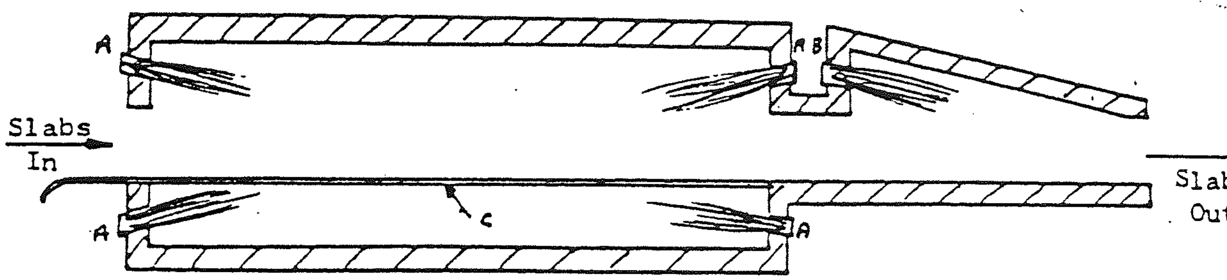


FUEL UTILISATION AT RAVENSCRAIG

FIGURE 3.1



SCHEMATIC OF NOS. 1 AND 2 REHEATING FURNACES



SCHEMATIC OF NO. 3 REHEATING FURNACE

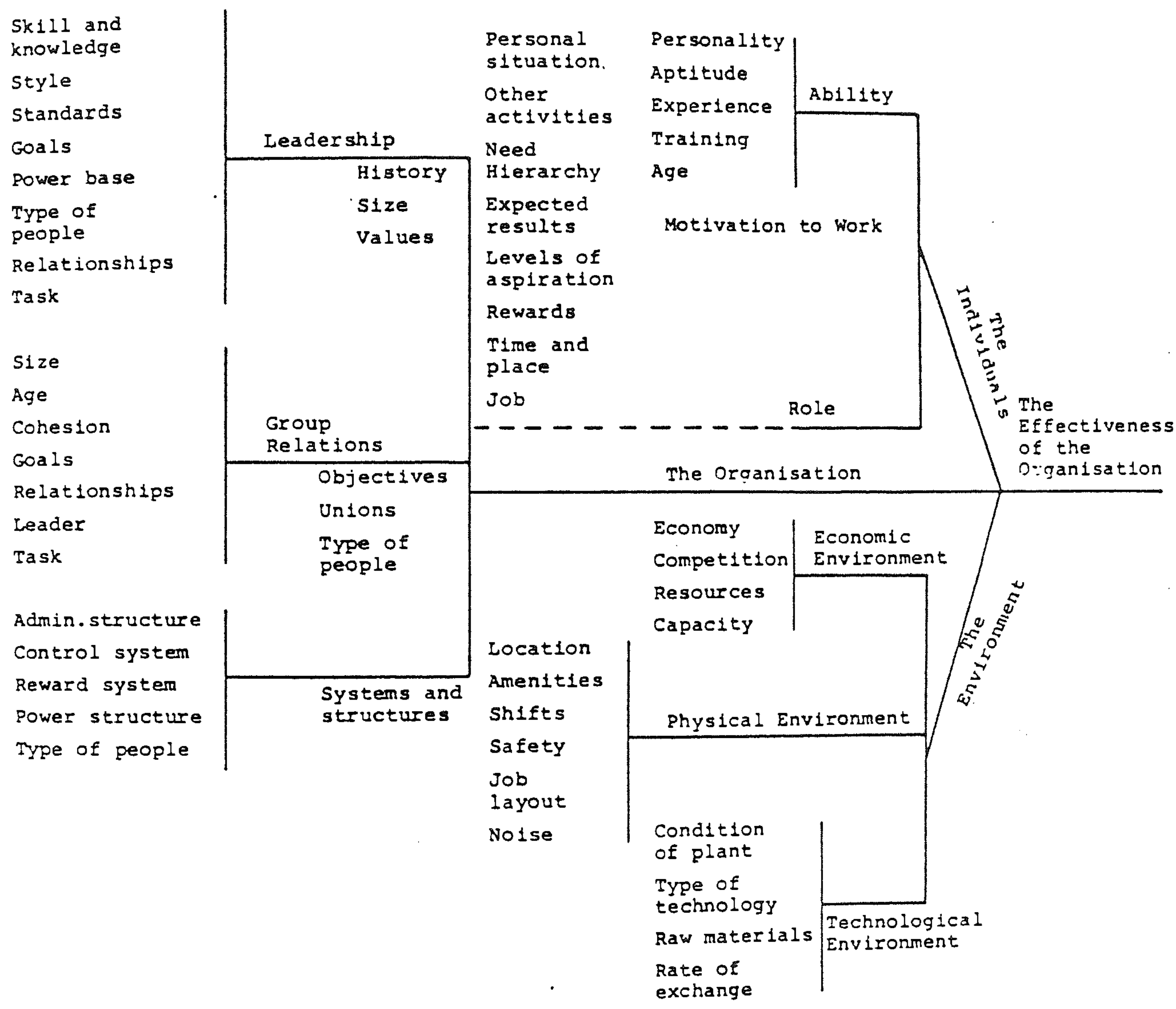
A = Oil Fired Burners

B = Low Pressure Gas Fired Burners

C = Water Cooled Skids

RAVENSCRAIG REHEATING FURNACES

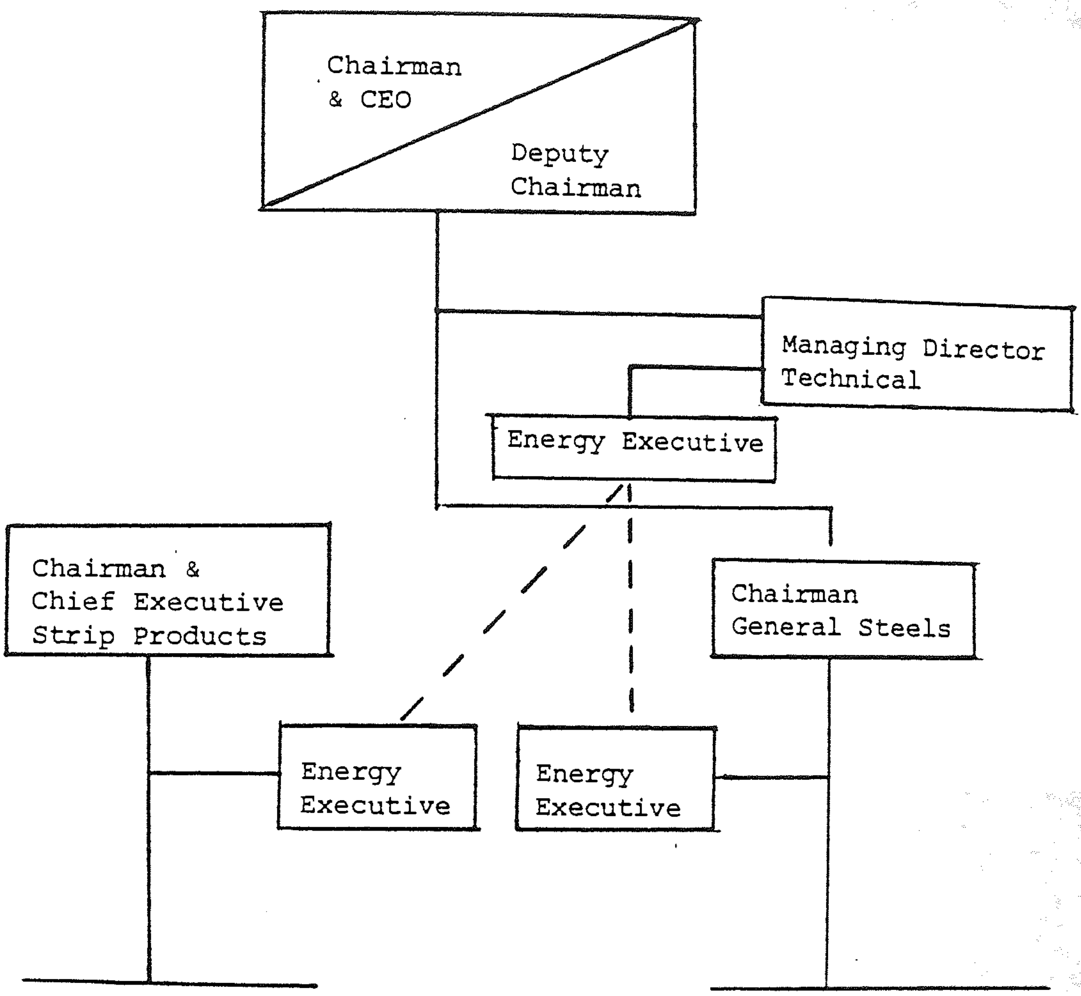
FIGURE 8.2



SOME FACTORS AFFECTING ORGANISATION EFFECTIVENESS

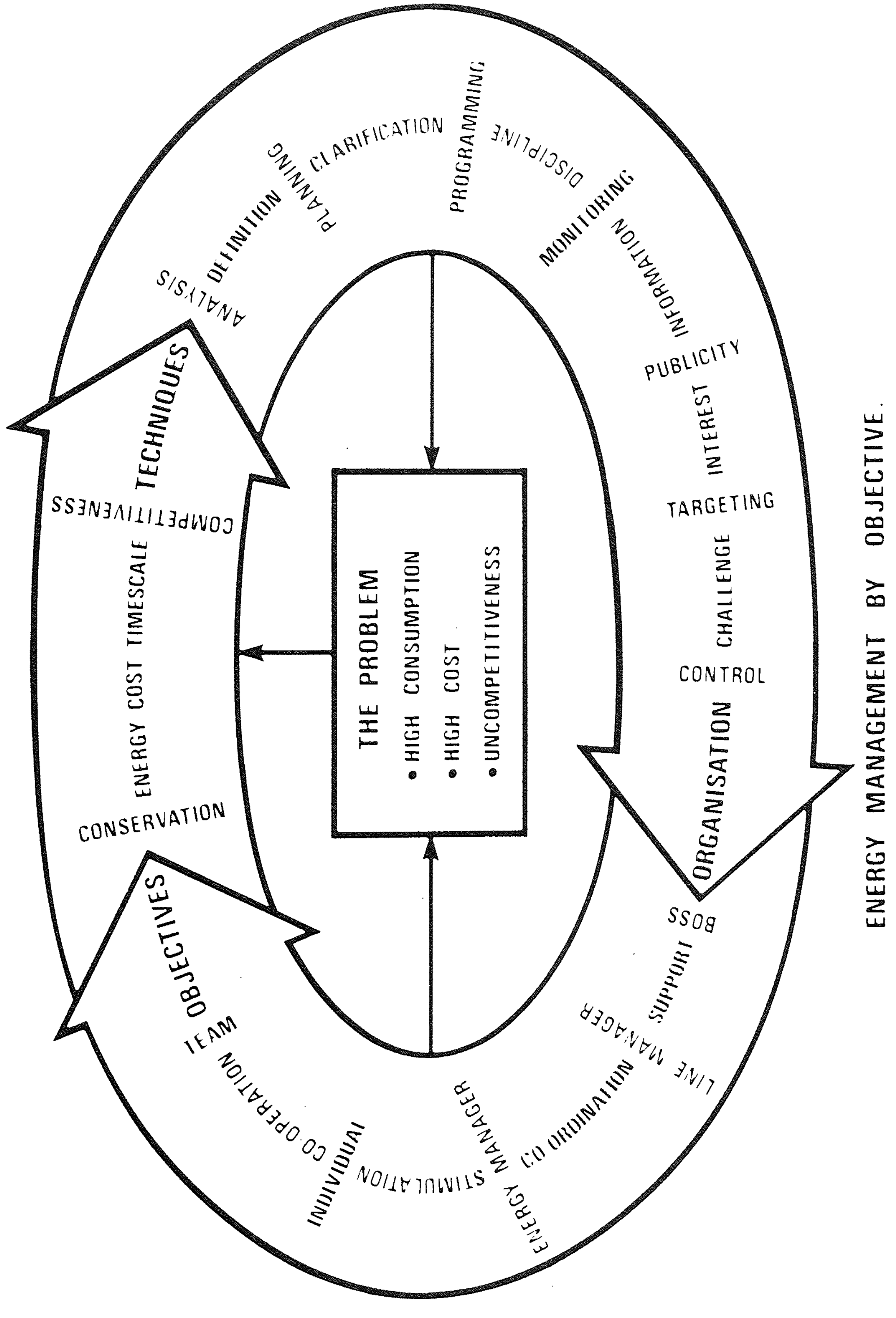
FIGURE 8.3

PROPOSED ENERGY MANAGEMENT STRUCTURE:



PROPOSED ENERGY MANAGEMENT MODEL

FIGURE 8.4



ENERGY MANAGEMENT BY OBJECTIVES

FIGURE 8.7

CHAPTER 9

CONCLUSIONS

9. CONCLUSIONS

The thesis clearly outlines the need to conserve energy as energy resources are limited and world demand has been doubling almost every fourteen years during the last quarter of a century. The last few years have witnessed an unprecedented increase in the price of energy available to industry in the United Kingdon and world-wide following the united action of OPEC Members on crude oil pricing first in 1973 and then in 1979. This has clearly defined the grave dangers and political implications on such a heavy reliance on imported oil for the consuming nations and has prompted the government to set the UK Energy Policy with the major objective to maximise the economic utilisation and exploitation of natural oil and gas resources over a time period.

As much as 8 per cent of the national total and nearly 25 per cent of the industrial total of energy delivered in UK is consumed by the iron and steel industry making it the biggest single industrial consumer (other than energy industries) of energy. Any energy savings in steelmaking thus show a substantial effect on the national and the Corporation's economy.

As an energy intensive industry the BSC has always been aware of the need to maximise energy consumption and has been actively pursuing the means of specific energy reductions through the process upgrading routes. Even so, as much as 70 per cent of the input energy is rejected to the environment in the form of waste gases and cooling water (29%), radiation losses from hot slabs (19%), radiation losses from the process plants (17%) and in slag cooling (6%). The recovery and recycling of waste energy has not been fully exploited in the corporation and thus offers a potential area for study. Furthermore, enhanced energy recycling offers a means of increasing the cost effectiveness of new and existing plants thereby maximising the return on assets at a time when major investment into new process plant is not appropriate.

An integrated steelworks constitue a large scale energy recycling system where enery is consumed, converted, recovered and transferred in an extremely complicated fashion between processes. Therefore, a correct picture of energy conservation can only be obtained by viewing things from an all inclusive stance. The thesis thus describes typical enrgy flows for various processes in a steelworks and goes on to outline energy conservation schemes for each of the individual porcesses.

Although the use of best available technology is the first move in decreasing energy consumption, immediate gains in the reduction of energy consumption should not be expected to come solely from the implimentation of exotic technology and sophisticated new equipment. The costs of inadequate pipe lagging thickness and unrepaired steam leaks offer high returns at modest costs in this area. Reduction in gas bleed loss through improvement in flexibility of fuel usage within works is the other area to consider.

Battelle Institute in a study carried out in 1975 (The Potential for energy conservation in steel industry) concluded that of the total achievable energy savings, 70 per cent can be attained using present technology and only 30 per cent by introducing new technology. This thesis concurs with this and shows that large savings can be achieved using present day technology (though novel to the Corporation) and then discusses its Research and Development work in the development of a novel Ceramic Rotary Regenerator in the application of heat recovery from high temperature waste gases, where no such equipment exists in the world.

The energy savings that can be achieved are thus summarised below:

Energy Recovery from Coke-ovens:

Sensible heat in coke is a major area of heat rejection. Dry coke coolong (DCC) is believed to offer a long term opportunity for the recovery of high grade steam, in parallel with producing significant environmental improvements. As an alternative to generating electricity or general process use the heat could be used to preheat the coal charge thus reducing the energy input to coking process.

Methods of energy recovery are discussed and its application in BSC is presented.

The case studies for DCC application in the BSC show it to be a capital intensive project with long pay back period. In addition the success of DCC application is largely dependent on the works ability to absorb the steam recovered either as process steam or in works power generation.

Energy Recovery from Sinter Plant:

Study indicates that substantial quantities of heat are rejected through cooling of sinter from the exit temperature of 600°C down to ambient. On conventional plants the sinter is cooled using forced circulation of cold air which in turn is rejected to atmosphere at about 250°C, with no attempt at heat recovery.

A scheme for heat recovery from sinter plant at Redcar is developed.

Application of such a scheme for partial cooling of sinter at Redcar has yielded energy savings of 32 per cent i.e. £450,000 p.a. with a pay back period of less than a year, a figure well above the predicted value , with air preheats of 300°C to 350°C. The scheme is to be further enhanced to improve savings.

Energy Savings from Iron-making:

Modern blast furnaces are designed to operate at top pressures in excess of 2 bar gauge to get better metal yields. The gas from top of such a furnace carries a large amount of recoverable energy by virtue of its pressure. By allowing the gas to expand through an energy recovery turbine, an energy recovery of upto 30 per cent of the total power required for the furnace blower can be achieved without any environmental problems.

The systems currently available on the market, though novel to the BSC, are discussed. The study shows that the economic case for the installation is very much dependent on parameters such as top pressure, top gas rate, clean gas temperature, local cost of electric power etc.

The installation of such a turbine is mainly applicable to Redcar No.1 furnace. Assuming an average electricity cost of 2.7 p/Kwh and economic assessment shows a savings of some £2.0 million per year with a pay back period of about two years. The project is actively being progressed in the Corporation.

Energy Savings from Steelmaking Plant:

BOS gases represent a major source of energy rejection by virtue of their high temperature, typically 1600°C above the vessel, and considerable fuel value, typical calorific value of 8 MJ per Nm^3 resulting from the high carbon monoxide concentration. There are several methods of dealing with the fume laden gases arising from the converter and these are discussed in detail in this thesis.

The minimum combustion system seems to offer the best overall solution, giving fuel gas recovery at a rate of 475 MJ per tonne of steel together with steam raising from the sensible heat of gases at the hood cooling circuit.

A design study for such an installation in the BSC is presented. The feasibility study shows that although the scheme is capital intensive, the pay back periods of two to four years can be achieved.

Heat Recovery from Soaking Pits and Reheating Furnaces:

High grade waste heat available from soaking pits and reheating furnaces offer substantial energy recovery opportunities. The study shows that air preheat offers the best mode of recuperation at a minimum of cost.

Different types of recuperative systems available in the market are compared and a design study for Ravenscraig soaking pit is presented.

In addition, the need for the development of the Ceramic rotary regenerator is discussed.

A general summary of the above schemes is shown in Fig.9.1

BSC Ceramic Rotary Regenerator (CRR):

A part of our novel work is the development of a CRR unit which is discussed in detail in Chapter 7.

The concept of a rotary regenerator is simple where heat is transferred by storage in and recovery from a porous ceramic matrix which is rotated about its vertical axis across the flow paths of hot gases and cold combustion air flowing in a counter current mode. The rotor structure is thus subjected to a complex non-symmetric thermal loading which results in both high hoop and tangential stresses within the matrix. Allied with the need of a composite rotor is the requirement for a flexible sealing system to prevent excessive leakage across the interface of the rotor and header system.

A design study was undertaken and a prototype rotor of 0.6 m diameter constructed. Through extensive pilot plant work the selection of ceramic material for the design of rotor and seal assembly is proved acceptable. Further improvement in engineering design are needed to produce acceptable sealing system. However, the general design of the system is proved successful.

A mathematical model predicting thermal performance was developed and verified on the pilot plant. In spite of high leakages, thermal recovery of upto 80 per cent with maximum air preheat of 770^{o}C from waste gases at 1200^{o}C is achieved.

With a daily stop/start system the pilot plant matrix has been subjected to a far higher duty than will be required in a steelworks operation on a continuous basis. On the other hand, the matrix has not been subjected to the effect of slag and other impurities which are generally present in the steelworks environment.

Application of rotary regenerator to a reheating furnace, giving an increase in air preheat temperature from 400^{o}C (conventional) to 770^{o}C, would yield an additional 20 per cent fuel savings.

A summary of this work is shown in Figure 9.2.

The sections discussed have clearly indicated that, given the availability of suitable capital for plant investment purposes, substantial quantities of energy could be made availa le from steelworks.

The recovery of waste heat can only be justified if it can be put to use preferably within works.

Frequently the overall energy balance of a steelworks is such that substantial quantities of recovered steam cannot be absorbed internally without making internally generated fuel gases redundant at the boiler plant or elsewhere. In the light of this, investment in new internal power generating plants became a logical adjunt to major energy recovery schemes. Where this is not feasible heat export schemes (Fig. 9.3) may offer financially attractive opportunities to the BSC at some sites given the limiting conditions such as load (preferably in excess of 36GJ/h), proximity of heat source (preferebaly not more than 5 - 6 kilometres) and high average load factor (preferably in excess of 60%) is met.

In conclusion a key factor, sometimes underestimated in the energy consumption/conservation equation, is people; their knowledge and their attitudes. Therefore, various energy management scenarios are discussed and a model for the BSC Energy Management is suggested.

Aston University

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Heating Applictions External To Works

Application	Grade of Heat
District Heating	P. H. W.
Fish Farming	Water 40°C
Horticulture	P. H. W. or Water ≥ 40°C
Absorption Refrigation	P. H. W. or Water ≥ 80°C
Drying	P. H. W. or Water ≥ 50°C
Industrial Space Heating	P. H. W.
Industrial Process Heating	P. H. W.

P.H.W. refers to pressurised hot water at 120°C to 140°C

FIGURE 9.3

APPENDICES

APPENDIX A

GRAPHICS DISPLAY OF MATRIX TEMPERATURE CHARACTERISTICS

CERAMIC ROTARY REGENERATOR

A programme has been developed - SLICE - which generates a file for the X - Y plotter. This file, when plotted, displays the temperature profile through the matrix of the ceramic rotary regenerator, the display splitting the matrix into the axial stations with temperature labelled isotherms. The display is in an isometric form to allow for easier assimilation and is divided into gas and air side.

Any number from 1 - 10 axial stations can be displayed. The isotherms are automatically determined by the input data which is, for convenience, the output from the 'Algol' MASH CRR mathematical model. The 'Basic' MASH versions can easily be altered to also give suitable data format.

Running the Programme   Run (22, 19) SLICE.

The new inputs are:-

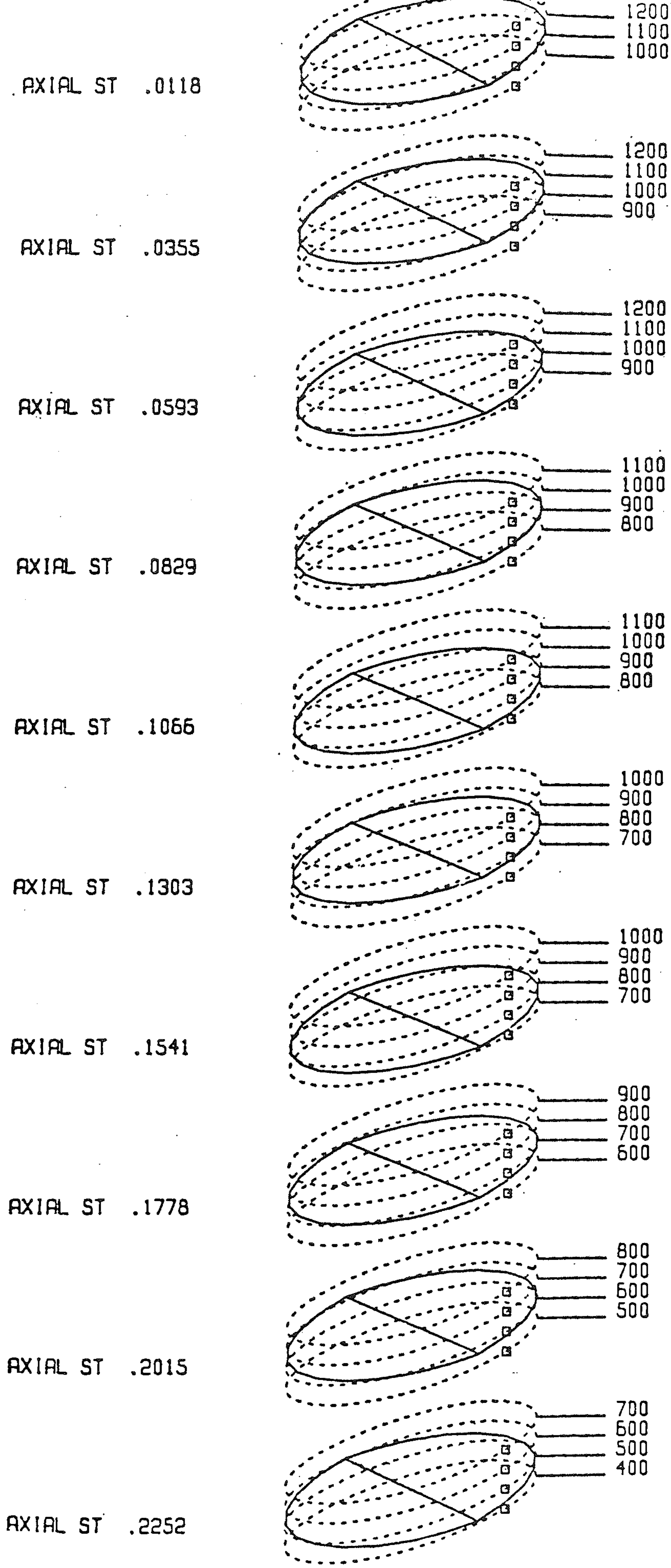
NAME OF INPUT FILE?	i.e data file to be used
NAME OF OUTPUT FILE?	i.e data file to be spooled to plotter by the user.
RADIUS?	i.e size of display in X-direction a null return gives a reasonable size.
TIMESTEPS?	i.e number of timesteps in input data file.
No. OF AXIS STEPS?	i.e number of axial stations to be displayed - null return gives all 10.
TEMPERATURE GAP?	i.e physical separation of isotherm lives which are, at present, set 100°C apart. Again a null return gives a reasonable size of display.

Error Messages

If a file is named for input which cannot be found, a mesage "NO SUCH FILE" is returned plus the request "NAME OF INPUT FILE". Other errors are handled by the normal system.

Example

Using STIR 2 . BAS (a modified version of STIR for use with latest GMASH algol program) a simulation of a CRR run with a gas inlet temperature of 1300°C, using GMAS.ASC. generated a data file MASHOU.PRO. This file was processed by SLICE . BAS to give the display attached. All 10 axial stations are displayed with relevant isotherm envelopes.



GRAPHIC DISPLAY OF TEMPERATURE DISTRIBUTION

FIGURE A.1

APPENDIX B

MATRIX SUPPORT GRID

In the preliminary design applications, a matrix block size of 60 x 60 x 70 cm has been specified. This will mean that the metallic support bars will be spaced at 60 cm intervals and so, for a matrix blockage fraction of 0.4, the uniformly distributed matrix weight loading on the bar could be taken to be:

$$Wm = (0.6 \times 0.7 \times 1.0 \times 0.4 \quad) \text{ kg/m} \qquad \ldots\ldots\ldots\ldots(1)$$

$$= 0.168$$

where = material density

Three materials, namely mullite, alumina-bonded silicon carbide and silliminite, have been identified as being suited to the rotor matrix duty and of these mullite has the highest density and therefore the value of this material has been used in all subsequent calculations.

$$Wm = 0.168 \times 2800 = 470 \text{ kg/m}$$

In addition to this, the bars will have to carry their own weight over the unsupported span and this will amount to a uniform load equivalent to bar weight per unit length. Because of the nature of the duty, a creep-resistance stainless steel material, Immaculate 5, has been selected for the support grid. This material has a density of 7880 kg/m^3 which gives a total bar loading of

$$W = Wm + 7880\ b^2$$

$$= (470 + 7880\ b^2) \text{ kg/m} \qquad \ldots\ldots\ldots\ldots(2)$$

where b = square bar diensions in meters.

For a simply supported beam of length l the maximum bending movement, Mm, for uniformly distributed loading may be taken to be

$$Mm = (W.l^2/8) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

and for the bar section with diagonal load orientation the section modulus is given as

$$Z = (b^3/6 \quad 2) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and so the maximum bending can be obtained as

$$m = Mm/Z$$

$$= \frac{(470 + 7880\ b^2).l^2.6 \quad 2.9.807}{8b^3}\ MN/m^2 \qquad \ldots\ldots\ldots (5)$$

At an operational limit of 500°C for maximum bar temperature Immaculate 5 material has the following strength properties:

(i) 0.1% Proof Stress $166.8\ MN/m^2$

(ii) Time yield creep strength $162.1\ MN/m^3$

For a maximum bar stress level of 160 MN/m^2 equation (5) gives a maximum unsupported bar span of

$$lm = \left(\frac{32729.b^3}{(- + 16.8\ b^2)}\right)^{\frac{1}{2}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

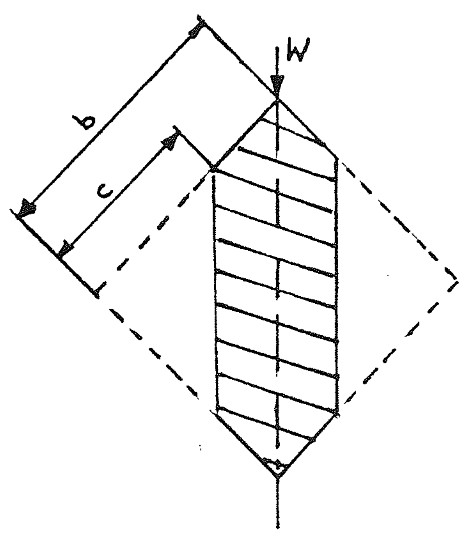
Span lengths for varying bar sizes, as obtained from Equation (6), are tabulated below:

Bar Size b (m)	*'Dead' matrix width a (m)	Max. Unsupported Span lm (m)
0.05	0.07	1.98
0.075	0.11	3.55
0.100	0.14	5.29
0.125	0.18	7.12
0.150	0.21	8.95

* Width of matrix area blanked off to axial flow.

$a = 2\ b \sin 45$

It will be noted that, for large bar spans, the blanked off area is a significant fraction of the 0.6 m block width. Spans may be reduced by using auxiliary support beams but the these in turn require blanking of the rotor face area. Reductions in the amount of blanking could be achieved by the use of bars of truncated square section which would retain the inclined face support concept and have only slightly reduced load-bearing capacity. The following sketch illustrates this proposal.



For the modified section the bar UD loading expression (2) will be altered to

$$W = 470 + 7880\ (b^2 - c^2) \quad kg/m \qquad \ldots\ldots\ldots\ldots(7)$$

The new section modulus will be

$$Z = \frac{b/2}{(b^4/12) - c^4/12)}$$

$$= \frac{b}{0.118\ (b^4 - c^4)} \qquad \ldots\ldots\ldots\ldots(8)$$

Hence

$$m = \frac{470 + 7880\ (b^2-c^2)).1^2.b}{8 \times 0.118\ (b^4 - c^4)} \times 9.807 \quad MN/m^2 \quad \ldots(9)$$

For a maximum working stress of 160 MN/m^2 this expression gives

$$lm = \left(\frac{32769\ (b^4-c^4)}{b\ (1 + 16.8\ (b^2-c^2))}\right)^{\frac{1}{2}} \qquad \ldots\ldots\ldots\ldots(10)$$

For a fixed blanked width of 0.07 m

$2(b - c)\ \sin 45 = 0.07$

$b - c = 0.05$ m

Maximum span length for varying values diemsion b are tabulated below:

b (m)	Max.unsupported span l m (m)	'Dead' matrix width (m)
0.075	3.54	0.07
0.100	5.22	
0.125	6.91	
0.150	8.56	

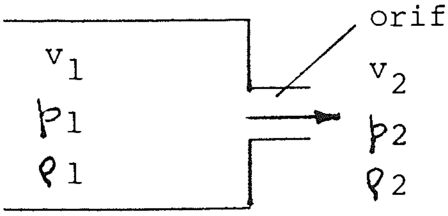
APPENDIX C

PROXIMITY SEAL : LEAKAGE ASSESSMENT

ASSUMPTIONS:

1. The leakage rate is dependent on pressure difference across the seal, density of the leaking fluid and the seal gap.
2. The leakage rate is independent of the fluid flow rates through the CRR.
3. The leakage rate for the 3 metre diameter CRR can be estimated from that of the 0.72 metre long seal test-rig, by ratio.
4. Typical conditions for a CRR would be a pressure difference across the seal of 4" w.g. and a combustion air flow of 2.5 kg/sec.

Assume that the proximity seal acts like an orifice allowing a volume rate to excape from a large chamberm at constant pressure to atmosphere via a small area.



v_1 can be considered as 0

From Bermoulli's equation for the flow of compressible gas

$$\left[\frac{\gamma}{\gamma-1}\right]\left[\frac{p_1}{\rho_1} - \frac{p_2}{\rho_2}\right] = \frac{v_2^{\,2} - v_1^{\,2}}{2} = \frac{V_2^{\,2}}{2} \quad \text{(for V = 0)}$$

For adiabatic flow

$$\frac{p_1}{\rho_1^{\gamma}} = \frac{p_2}{\rho_2^{\gamma}}$$

$$\rho_2 = \rho_1\left(\frac{p_2}{p_1}\right)^{1/\gamma} = \rho_1\, r^{1/\gamma} \qquad \text{for } r = \left[\frac{p_2}{p_1}\right]$$

Therefore

$$\left[\frac{\gamma}{\gamma - 1}\right]\frac{p_1}{\rho_1}\left[1 - r^{(\gamma - 1)/\gamma}\right] = \frac{1}{2} v_2^{\,2}$$

$$v_2 = \sqrt{2\left[\frac{\gamma}{\gamma - 1}\right]\frac{p_1}{\rho_1}\left[1 - r^{(\gamma - 1)/\gamma}\right]} \qquad \text{.......(1)}$$

$$Q = C_d A v_2 \qquad \text{---- (2)}$$

<u>Application of Test Results</u> seal length 0.72 metres

Gap mm	Leakage Flow $m^3/sec \times 10^{-3}$	Air Temp. ^oC	Air pressure N/m^2	Density kg/m^3	Seal length m
0	3.68	21	102320	1.212	0.72
.25	5.5	22	"	1.208	"
.50	9.97	24	"	1.200	"
.75	15.15	27	"	1.188	"
1.00	20.6	30	"	1.176	"
1.25	30.15	34	"	1.161	"

These leakage rates can be extended to show the rates through a 3 metre seal. The leakage rates will be altered by ratio

Leakage rate by expansion $m^3/ sec \times 10^{-3}$	3.68	5.5	9.97	15.15	20.6	30.15
Leakage rate for 3M seal $m^3/sec \times 10^{-2}$	1.53	2.29	4.15	6.31	8.58	12.56

1. <u>Touching Seal</u>

Use equation (1) to estimate velocity through gap.

$$V_2 = \sqrt{2\left[\frac{\gamma}{\gamma-1}\right]\frac{p_1}{\rho_1}\left[1 - r^{(\gamma-1)/\gamma}\right]}$$

with $\gamma = 1.4$

$p_1 = 102320$

$\rho_1 = 1.212$

$r = \frac{p_2}{p_1} = 0.9903$

$$V_2 = 40.5 \text{ m/sec.}$$

Use equation (2) to estimate area of gap if $C_d = 0.8$

$$A = \frac{Q}{C_d V_2} = \frac{1.53 \times 10^{-2}}{0.8 \times 40.5} = 4.72 \times 10^{-4} \text{ m}^2$$

This implies average gap of 1.5 mm along seal gap.

Simulate air leaking at 700°C

$$\rho\, 700^\circ C = 1.293 \times \frac{102320}{101325} \times \frac{273}{973} = 0.366 \text{ kg/m}^3$$

Equation (1) $V_2 = 2\left[\frac{\gamma}{\gamma-1}\right]\frac{p_1}{\rho_1}\left[1 - r^{(\gamma-1)/\gamma}\right]$

$= 73.77$ m/sec

Equation (2) using $C_d = 0.8$ & $A = 4.72\times10^{-4}$ m^2

$$Q = C_d A V_2$$

$= 2.786 \times 10^{-2}$ m^3/sec

$= 1.01 \times 10^{-2}$ kg/sec

$\rho\, 700^\circ C = 0.362$ kg/m^3

Volume of leakage of cold air = 1.53×10^{-2} m^3/sec

($21^oC = 1.2$ kg/m^3)

mass flow = 1.836×10^{-2} kg/sec

If CRR 3M unit has 2.5 kg/sec of combustion air flowing through it the per cent leakage is

$$\% \text{ leakage} = \frac{(1.836 \times 10^{-2} - 1.01 \times 10^{-2})}{2.5} 100$$

$$= 1.1\%$$

2 Varying Seal Gaps (Summary)

Gap mm	Leakage of cold air $\times 10^{-2}$ m^3/sec	C_d	Leakage of air at 700^oc $\times 10^{-2}$ m^3/sec	Total Leakage air % of 2.5 kg/sec
0	1.53	0.8	2.79	1.1
.25	2.29	0.74	4.16	1.7
.50	4.15	0.67	7.54	3.1
.75	6.31	0.67	11.37	4.6
1.00	8.58	0.68	15.39	6.2
1.25	12.56	0.80	22.38	9.0

APPENDIX D

ROTARY REGENERATOR: REFRACTORY WEAR

ROTOR SPEED = 0.34 r.p.m.

MATRIX DIAMETER = 0.560m

SEAL SHOE THICKNESS = 13 MM

LOAD ON SHOE - WEIGHT OF SHOE = 4 ozs.

WEIGHT OF HOLDER - (OUTER) = 21 ozs.

(CENTRE) = 17.5 ozs.

Time for 1 revolution = 2.94 mins.

Distance (max) travelled for 1 rev. = πD = 1.76m

Therefore distance (max) travelled per day = $\frac{1.76}{2.94} \times 60 \times 24 = \underline{862\ m}$

IN SHOE WEAR RIG: Distance (max) travelled per day = $\frac{4}{12} \times \frac{24 \times 60 \times 41}{3280} \times 1609.3 = \underline{5998m}$

Reference Tests III and IV

LOAD 3.8 lbs WEAR RATE = 0.292 mm/day

LOAD 8.6 lbs WEAR RATE = 0.930 mm/day

$$\underline{\underline{\text{WEAR RATE} \propto (\text{LOAD})^{1.4}}}$$

PREDICTED WEAR RATE ON LABORATORY RIG

LOAD ON SHOE = 1.56 lbs (0.706 kg)

$$\frac{0.292}{x} = \frac{(3.8)^{1.4}}{(1.56)^{1.4}}$$

$x = 0.08396$ mm/day (i.e. Wear rate on rig)

Wear Rate on Rig $= 0.08396 \times \frac{862}{5998}$

$= 0.0121$ mm/day

Assumption Shoe wear of 10 mm before replacement

Life of seals = 829 days (continuous running)

<u>$2\frac{1}{4}$ years</u>

<u>FOR</u> Matrix Diameta = 2.5m and rotor speed = .1 r.p.m.

Distance travelled per day $= \frac{\pi \times 2.5}{10} \times 60 \times 24 = \underline{1131 \text{ m}}$

Effective Wear Rate $= 0.08396 \times \frac{1131}{5998} = .0158$ mm/day

Life of seals = 632 days

= <u>$1\frac{3}{4}$ years</u> CONTINUOUS RUNNING

APPENDIX E

COMPUTER PROGRAM (CRR)

1. PROGRAM VARIABLES

			Section of Program
ABS TEMP	=	TEMP + 273.15	CP WALL PROG
CP WALL	=	300* (LN ABS TEMP) - 867	"
MASS TO VOLUME	=	Library Programme	Determines Gas Parameter
GAS PRESSURE	=	Gas Pressure	"
HYDIA	=	Hydraulic Diameter of Passage	"
PCO_2	=	Partial Prrssure of CO_2 x mean beam length	"
PH2O	=	Partial Pressure of H_2O x mean beam length	"
DE	=	Spectral Overlap	"
EH_2O	=	Type of emissivity independant of temperature	"
ECO2	=	Type " " " "	"
ET	=	Total emissivity " " "	"
ECOEFF	=	Total emissivity times Stefan Boltzman constant	"
RE	=	Reynold's Number	Calculate Factor
FLUID FLUX DENSITY	=	Fluid flux density (kg/m^2s)	"
MUFLU	=	Viscosity of gas at certain conditions	"
XPLUS	=	Length/diameter ratio divided by Re No.	"
SUMX	=	Length down the passage (accumulated)	"
NUX	=	Nusslet Number	"
RED	=	Re No divide by 1000	"
HTCOEFF	=	L = corrective heat transfer coefficient	"
KFLU	=	Thermal conductivity of gas	"
CP FLU	=	Specific heat of a gas	"
X	=	Absolute gas temperature(Function of I &J)	"
Y	=	Absolute matrix temperature(Function of I & 3)	"
RAD HTCOEFF	=	Radiation heat transfer coefficient	"

			Section of Program
HT COEFF	=	Total heat transfer coefficient	Calcualte Factor
WALL THICKNESS	=	Wall thickness	"
RHO WALL	=	Density of refractory	"
WALL SPHT	=	Specific heat of refractory	"
AIR PERIOD	=	Time taken for one passage to travel across air duct	"
GAS PERIOD	=	Time taken for one passage to travel across gas duct	"
K WALL	=	Thermal conductivity of refractory	"
XX	=	Dimensionless variable associated with Hausen coefficient	"
PHI	=	Thermal thickness used in Hausen calc.	"
HAUSEN COEFFICIENT	=	Hausen coefficient a modified overall heat transfer coefficient	"
FRICTION COEFF	=	Friction coefficient used in calculating the function factor	"
FRICTION FACTOR	=	Friction factor used to calculate the pressure charge	"
GAS TEMPERATURE	=	Inlet temperature of gas specified in Input File	PROGRAM
AIR TEMPERATURE	=	Inlet temperature of gas specified in Input File	"
GAS PRESSURE	=	Inlet pressure of gas specified in Input File	"
MEAN GAS FLUX DENSITY	=	Gas Flux density taking leakage into account	"

		PROGRAM
FH_2O	= Gas mass fraction of water vapour (leakage compensation)	"
FN2	= Gas mass fraction of nitrogen vapour (leakage compensation)	"
FO2	= Gas mass fraction of oxygen vapour (leakage compensation)	"
FCO2	= Gas mass fraction of carbon dioxide (leakage compensation)	"
FSO2	= Gas mass fraction of sulphur dioxide (leakage compensation)	"
KWALL	= Thermal conductivity of rotor material	"
BLOCKAGE	= Gas sectional area of rotor material as percentage of rotor area	"
PASSAGE LENGTH	= Rotor depth	"
Several leakage variables	= self explanatory	MATRIX TEMPERATURE
OLD GFD	= original gas flux density inlet duct	"
OLD AFD	= original air " " "	"
DH2O	= original gas mass fraction of waste vapour	"
DN 2	= original gas mass fraction of waste nitrogen	"
D O2	= original gas mass fraction of waste oxygen	"
DCO2	= Original gas mass fraction of carbon dioxide	"

DSO2	=	Original gas mass fraction of sulphur dioxide	"
OLD GAS TEMP	=	Original gas temperature	"
PI	=	3.1415926 535 898	"
MH2O	=	Gas mass fraction of water vapour (leakage compensated)	"
M N2	=	Gas mass fraction of nitrogen (leakage compensated)	"
M O2	=	Gas mass fraction of oxygen (leakage compensated)	"
M CO2	=	Gas mass fraction of carbon dioxide (leakage compensated)	"
M SO2	=	Gas mass fraction of sulphur dioxide (leakage compensated)	"
XSTEP	=	The number of axial steps in the rotor	
Z	=	Gas temperature	"
X	=	initial guess at bottom rotor temp.	"
AXIAL STEP	=	Actual length of section used in calculating temps down through passage	"
FRICTION MULTIPLIER	=	1/d term basic pressure loss factor	"
CRCA	=	Solid/Air capacity rate ratio	"
CRCG	=	Silid/gas capacity rate ratio	"
MAX DTHETA	=	factor used in relaxation	"
DELTA F	=	" "	"
RELAXATION FACTOR	=	Relaxation factor used to converge iteration	"

			MATRIX TEMPERATURE
GAS TSTEPS	=	No of time segments in gas period	"
AIR TSTEPS	=	No of time segments in air period	"
GAS TIMESTEP	=	Actual time of one gas time segment	"
AIR TIMESTEP	=	" " air "	"
CP AIR	=	Library Procedure to find specific heat of air	"
GAS FLUX DENSITY (I)	=	Gas flux density in any gas time step	"
GAS TEMP (I,J)	=	Gas temperature anywhere in matrix	"
AIR FLUX DENSITY (I)	=	Air flux density in any air time step	"
AIR TEMP (I,J)	=	Air temperature anywhere in matrix	"
S	=	Variation associated with temperature calculation Flow Cycles	
SS	=	" " "	"
T	=	" " "	"
CYCLE COUNT	=	Counts the no of iterations before the differences between the starting and finishing temps becomes countable	MATRIX TEMPERATURE
Z	=	gas temperature	Gas Temp Iter
Y	=	gas flux density less gas to ats leakage	"
GAS TEMPERATURE	=	inlet gas temperature compensated for leakage	"
AIR PREHEAT TEMP	=	mean air preheat temperature	"
H GAS	=	library procedure to predict enthalpy of a gas	"

GM COUNT	=	gas flow distribution count	Gas Cycle
GP COUNT	=	gas pressure equalisation count	"
RHO GAS	=	library procedure for calculation of density	Gas Pressure Equalisation
I	=	gas flow distribution iteration variable	"
J	=	gas pressure equalisation iteration variable	"
MUGAS	=	library programme for calc of viscosity of a gas	"
CP GAS	=	library programme for calc of specific heat of a gas	"
R	=	Gas Constant/molecular weight of the gas matrix	"
RHO FLU	=	density of the gas mixture for particular conditions	"
K	=	dimensonless variable	"
L	=	" "	"
GAS TEMPERATURE	=	change in gas temp from one axial step to the next	"
MATRIX TEMPERATURE	=	change in molecular temp from one axial step to the next	"
PRESSURE CHANGE	=	change of pressure from one axial step to the next	"
PRESSURE	=	pressure in axial step	"
GAS RHODX	=	variable associated with calc of gas carry over	"

PRESSURE DROP	=	drop in pressure from inlet press to press in axial step	"
GAS OUTLET TEMP	=	self explanatory	"
REF PRESSURE DROP	=	Pressure drop on first gas time step	"
PRESSURE DROP RATIO	=	Ratio of pressure drop to reference pressure drop	"

<u>DISK INPUT DATA FILE</u>

Rotor Speed (r.p.m.)

Air Flux Density (Kg/s m^2)

Gas Flux Density (Kg/s m^2)

Hot Face: air to gas leakage (Kg/s)

Cold Face: air to gas leakage (Kg/s)

High Pressure: air to atmosphere leakage (Kg/s)

Low Pressure: air to atmosphere leakage (Kg/s)

High Pressure: gas to atmosphere leakage (Kg/s)

Low Pressure: gas to atmosphere leakage (Kg/s)

Hole Length / Diameter Ratio

Air Port Area (m^2)

Air to Gas Port Area Ratio

Hole Diameter (m)

Wall Thickness (m)

Gas Inlet Temperature (^oC)

Air Inlet Temperature (^oC)

Air Inlet Pressure (N/m^2 abs)

Gas Inlet Pressure (N/m^2 abs)

Gas Mass Fraction of H_2O

Gas Mass Fraction of N_2

Gas Mass Fraction of O_2

Gas Mass Fraction of CO_2

Gas Mass Fraction of SO_2

Rotor Material: Density of Material

Thermal Conductivity of Material

Rotor Material - Alumina Bonded Silicon Carbide (ABSC)
Mullite (MAT. 2)
Sillimanite (MAT. 3)
Steel (MAT. 4)

"Stir" creates two output files "Leakage Data" and "Stir Output", both of which are utilised by "Mash"

Procedure "Do I read" this procedure tells the program which pieces of data to read from the data file. If the item is not "*" (which signifies repeated variable) then the command to read that item is given otherwise the item is recorded as being a repeated data item.

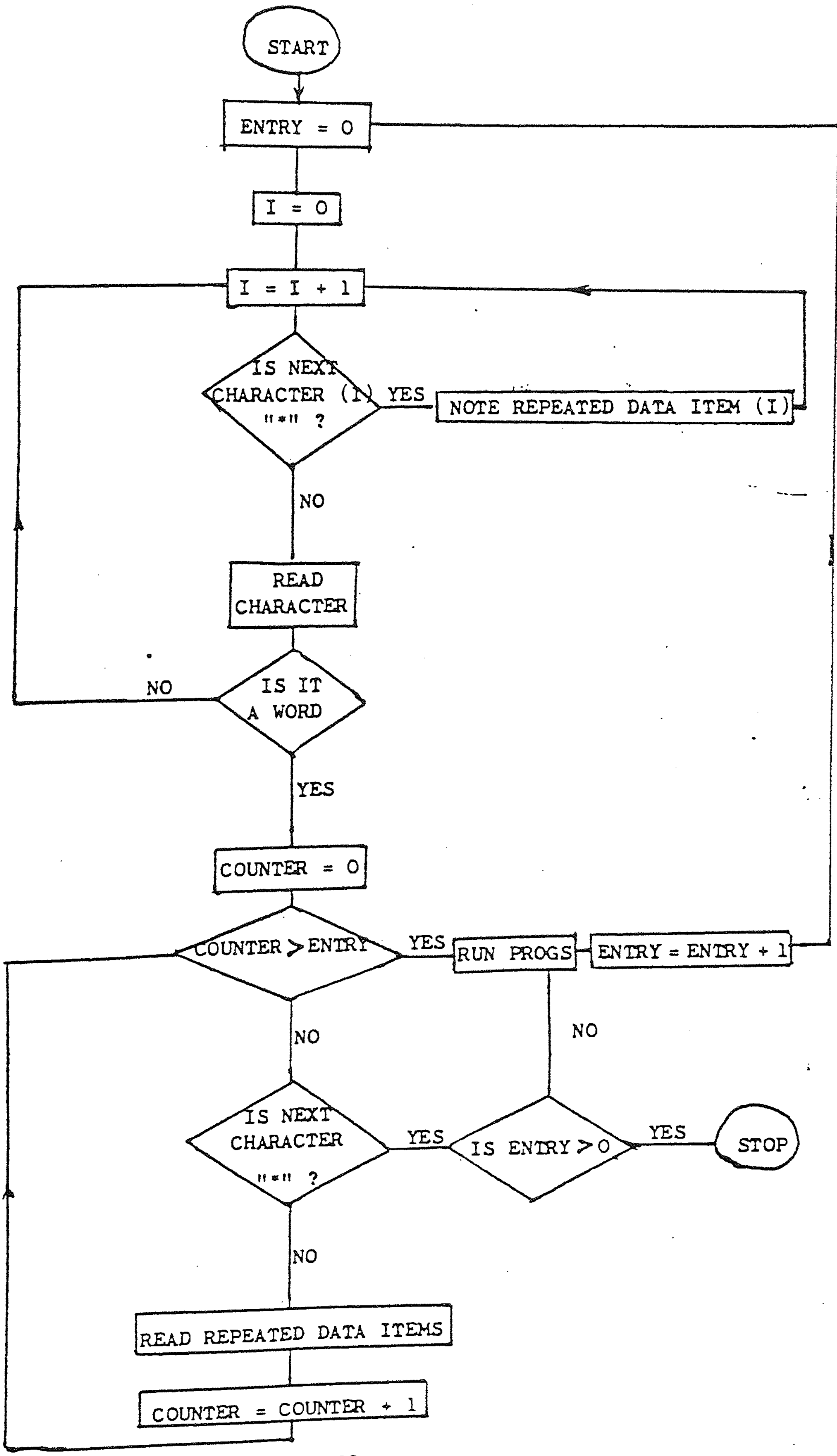
Procedure "Read Sets" after "Do I Read" has been called for all data items up to "Mullite" "Read Sets" is called to note whether the next character is "*". If this is so the data is terminated i.e. there is no (more) repeated data if "Mash" and "Pour" have not yet been called for that "Data Set" then the run proceeds, if mash and pour have been used then the run is over.

If the next character is not "*" then there is repeated data to read "Read Sets" uses information given by "Do I Read" as to which are the repeated data items and reads the first set of repeated data items on which subsequent calculation are performed the process is iterative and after as the results using the first set have been "Poured", a check is made again to set if the next character is "*" etc. see flow chart.

3. PROGRAM

Having obtained the correct data for the first run the program begins.

FLOW CHART FOR "DO I READ" AND "READ SETS"



BLOCKAGE = 1-1/(1+ WALL THICKNESS/HYDRAULIC DIAMETER)^2

PASSAGE LENGTH = LENGTH TO DIAMETER RATIO * HYDRAULIC DIAMETER

AIR PERIOD = 60/(ROTOR SPEED * (1+1/AIR TO GAS RATIO))

THE PROGRAM NOW OUTPUTS THE "STIR DATA FILE" TO THE LINE PRINTER FOR PURPOSES OF CROSS REFERENCING WITH COMPUTER RESULTS. ITS FORMAT DIFFERS FROM THAT OF THE ORIGINAL DATA FILE ONLY IN THAT THE CORRECT ITEMS OF REPEATED DATA HAVE BEEN SELECTED AND THE NAME OF THE MATERIAL IS NOW REPLACED BY ITS DENSITY AND THERMAL CONDUCTIVITY IN THAT ORDER.

LEAKAGE CALLS

AIR FLUX DENSITY = AIR FLUX DENSITY/(1 - BLOCKAGE)

GAS FLUX DENSITY = GAS FLUX DENSITY/(1 - BLOCKAGE)

OLD GFD = GAS FLUX DENSITY

Z= AIR TO GAS AREA RATIO/(AIR PORT AREA *(1 - BLOCKAGE)

GAS FLUX DENSITY = GAS FLUX DENSITY - HF GAS TO ATS LEAKAGE * Z

NEWFD = GAS FLUX DENSITY + HOT FACE AIR TO GAS LEAKAGE * Z

COMMENT: THE FIRST TWO OF THE ABOVE LINES CORRECTS FLUX DENSITIES BASED ON THE ENTIRE PORT AREA TO ONES BASED ON FREE FLOW AREA ONLY. THE FOLLOWING DETERMINE THE TRU FLUX DENSITIES TAKING LEAKAGE INTO ACCOUNT.

IF NEWFD GAS FLUX DENSITY THEN

BEGIN

THIS PART OF THE PROGRAM CHECKS WHETHER OR NOT AIR LEAKS INTO THE GAS STREAM AT THE HOT FACE; IF IT DOES THEN OBVIOUSLY THE GAS COMPOSITION IS ALTERED, IF IT DOES NOT IT IS LEFT UNCHANGED.

END

ELSE

BEGIN

END

GAS FLUX DENSITY = NEWFD

NEXT STEP:

OLD AFD = MR FLUX DENSITY

Y = 1/(AIR PORT AREA * (1 - BLOCKAGE))

AIR FLUX DENSITY = AIR FLUX DENSITY - (COLD FACE AIR TO GAS LEAKAGE + HF AIR TO ATS LEAKAGE)*Y

THESE LINES PERFORM THE SAME CORRECTIONS ON THE AIR FLOW AS WERE PREVIOUSLY CARRIED OUT ON THE GAS FLOW.

THE PROGRAM NOW CHECKS THAT NEITHER GAS NOR AIR FLOW HAS BEEN REDUCED BELOW ZERO IF THIS IS SO THEN THE RUN IS ABANDONED, ENTRY IS SET TO ONE (SEE FLOW CHART) AND THE NEXT SET OF DATA IS SOUGHT. OTHERWISE THE OUTPUT FILES FOR "MASH" ARE CREATED.

LEAKAGE DATA IS CREATED FIRST. ITS FORMAT IS

HOT FACE AIR TO GAS LEAKAGE = Z

COLD FACE AIR TO GAS LEAKAGE = Z

HF AIR TO ATS LEAKAGE = Y

LF AIR TO ATS LEAKAGE = Y

HF GAS TO ATS LEAKAGE = Z

LF GAS TO ATS LEAKAGE = Z

AIR TO GAS AREA RATIO

OLD GFD

OLD AFD

MH_2O

MN_2

$M\ O_2$

$M\ CO_2$

$M\ SO_2$

"STIR OUTPUT" IS THEN PRODUCED. ITS STRUCTURE WILL BE CLARIFIED BY REFERENCE TO "MASH"

FALSE

FALSE

TRUE

MATERIAL INDICATOR

GAS TEMPERATURE

AIR TEMPERATURE

GAS PRESSURE

AIR PRESSURE

GAS FLUX DENSITY

AIR FLUX DENSITY

FH_2O

FN_2

FO_2

FCO_2

FSO_2

KWALL

RHOWALL

WALL THICKNESS

HYDRAULIC DIAMETER

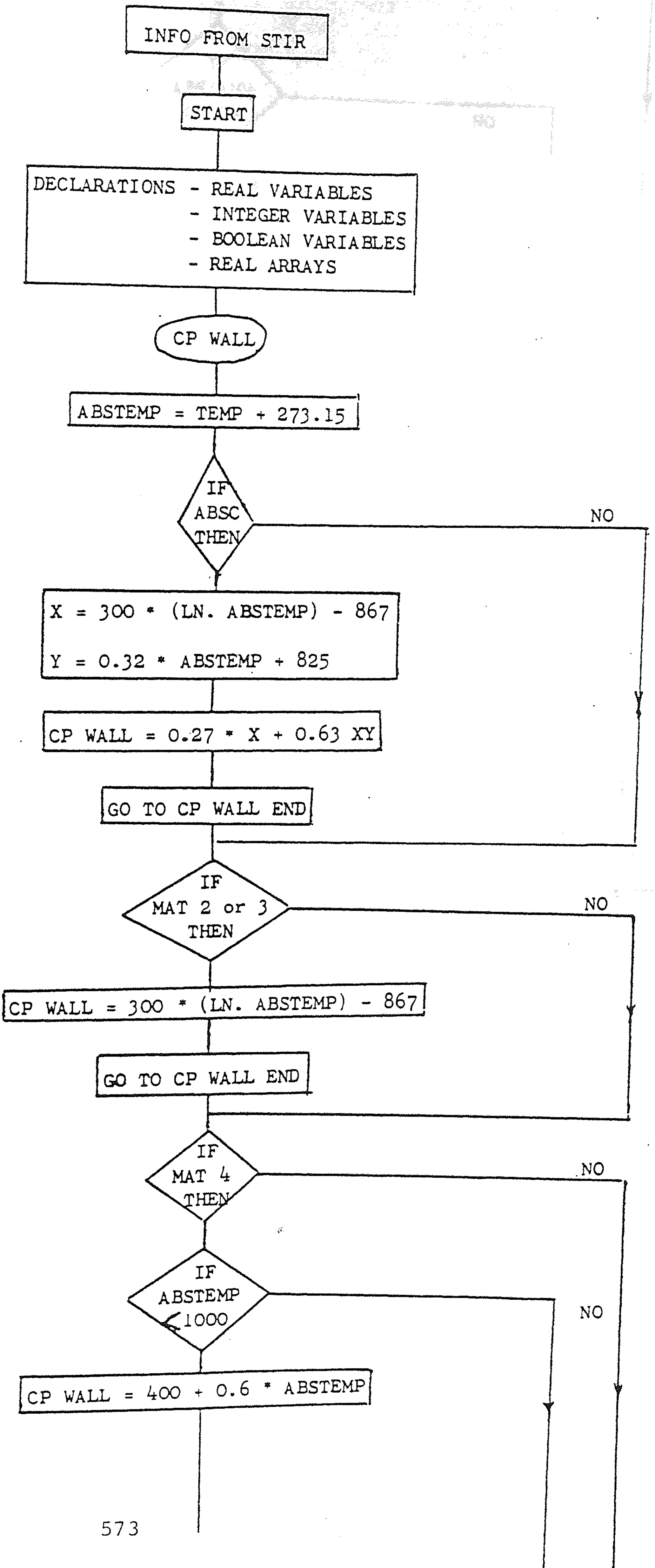
BLOCKAGE

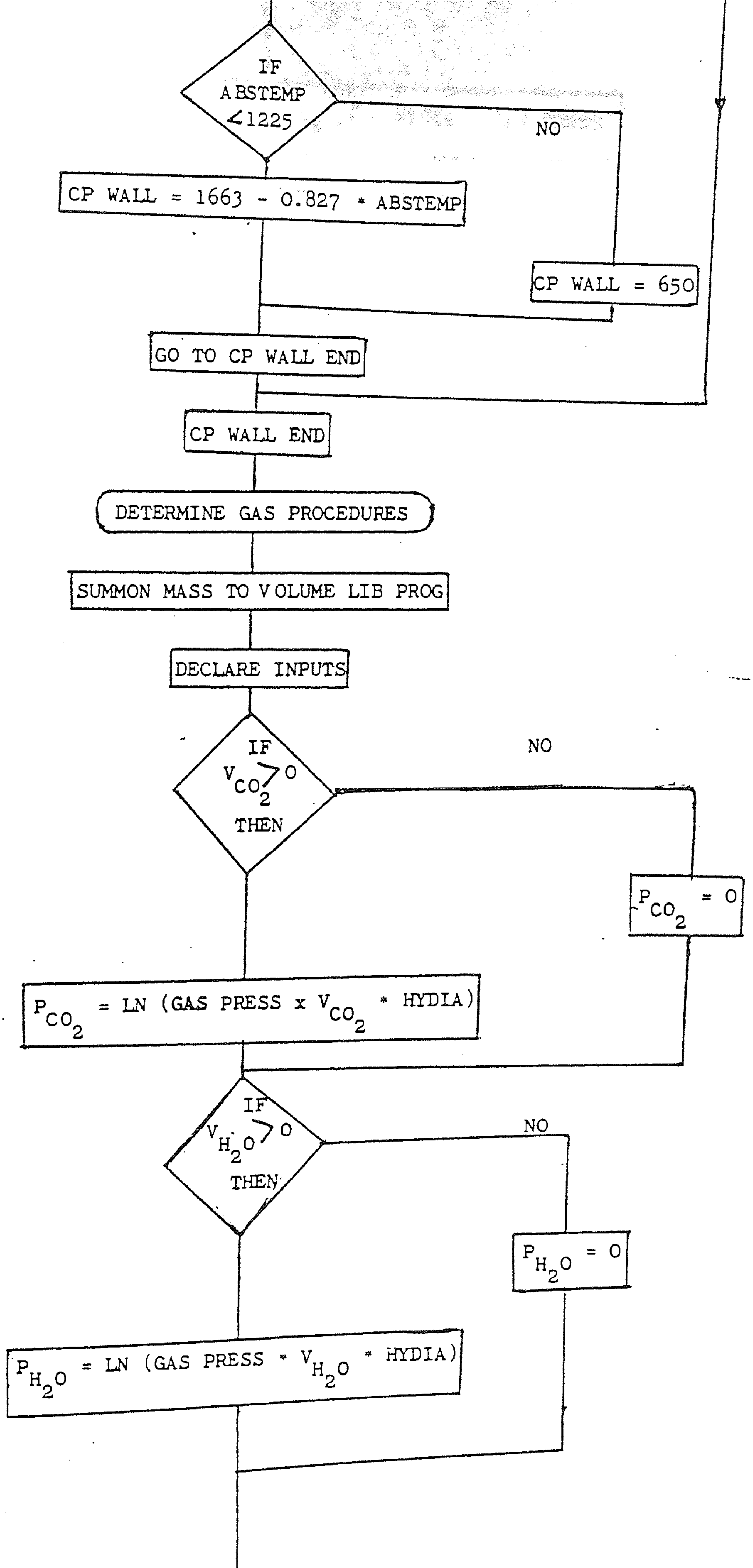
AIR PERIOD/AIR TO GAS PREHEAT

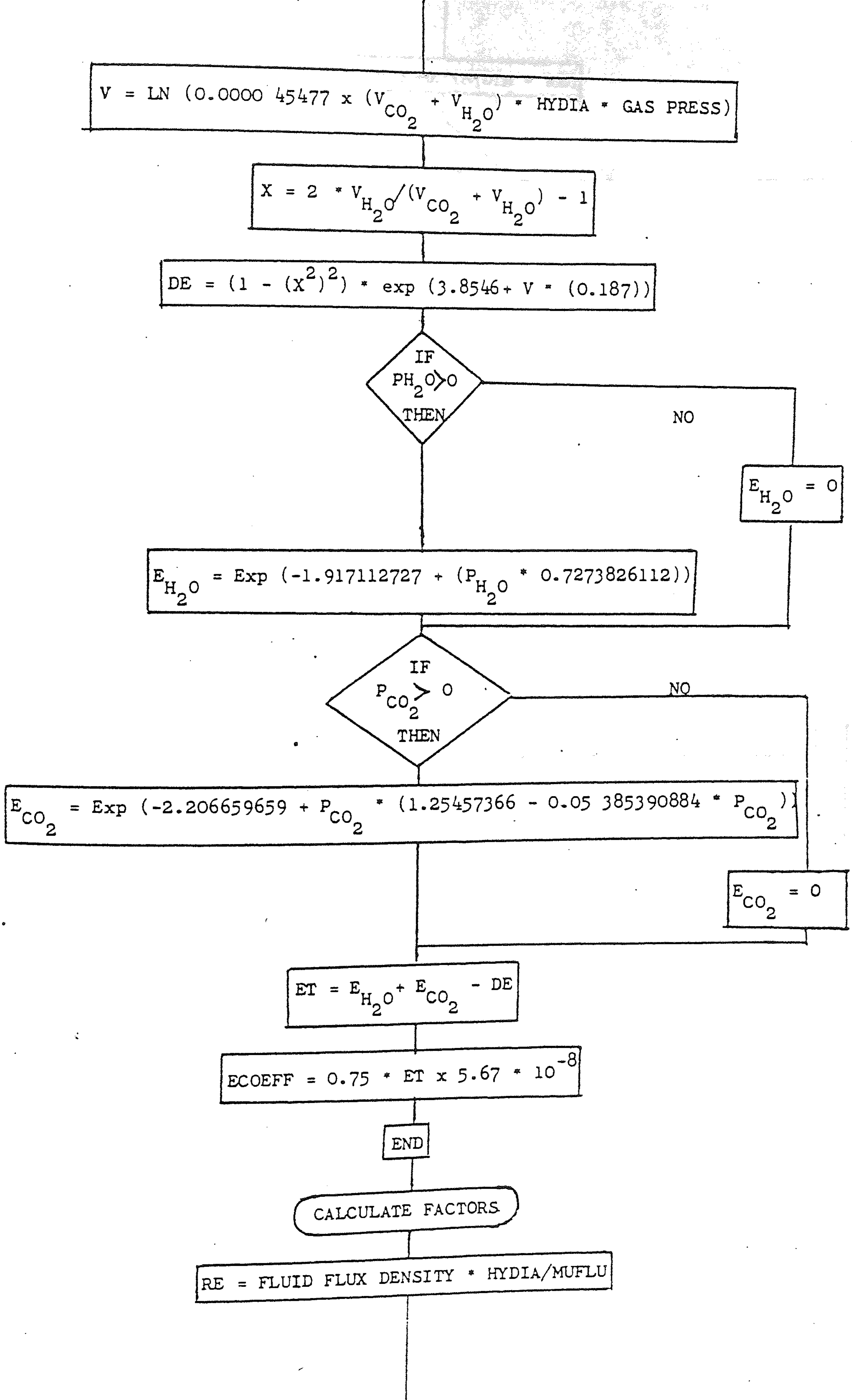
AIR PERIOD

PASSAGE LENGTH

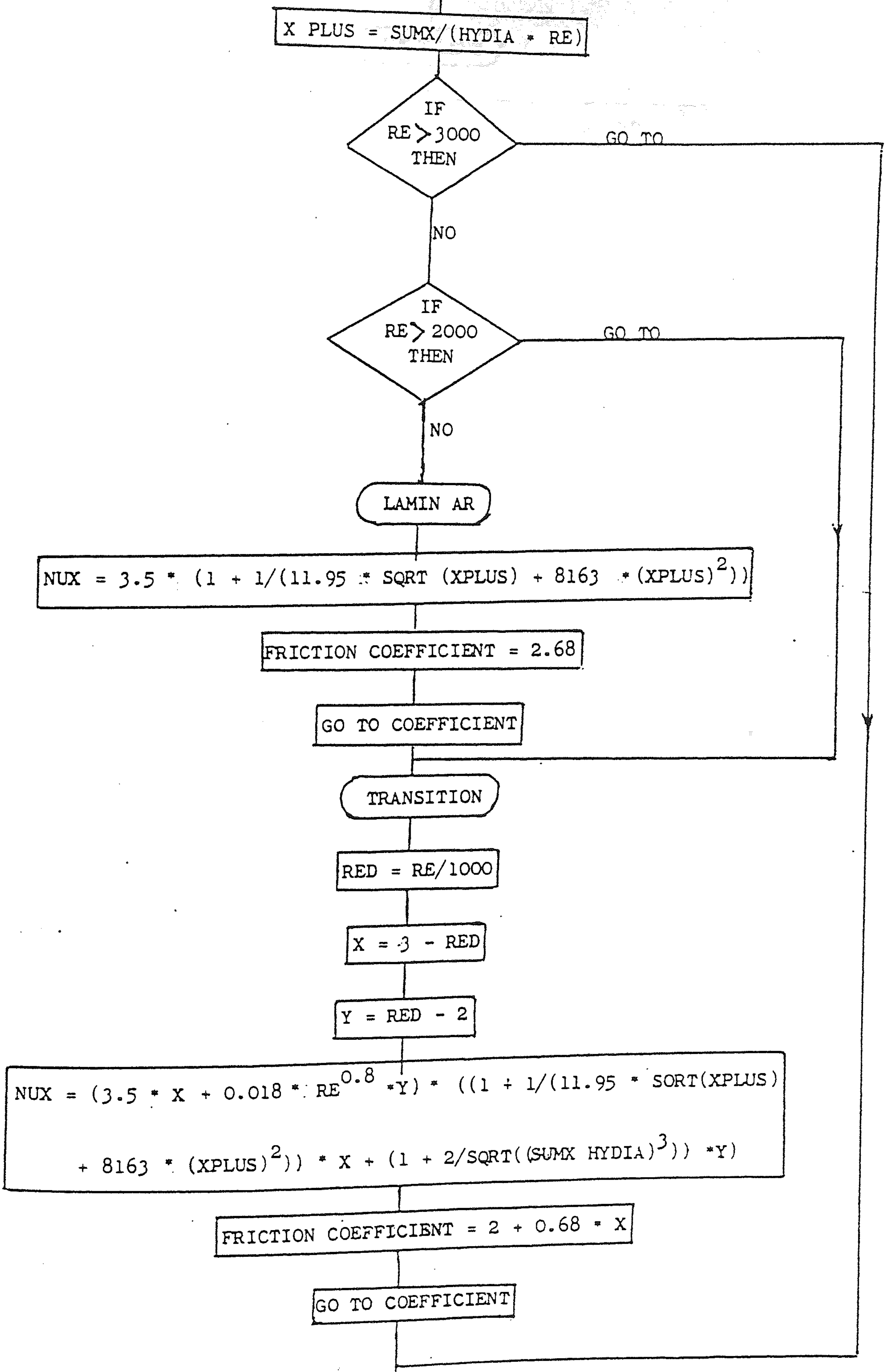
"MASH" IS NOW FOUND

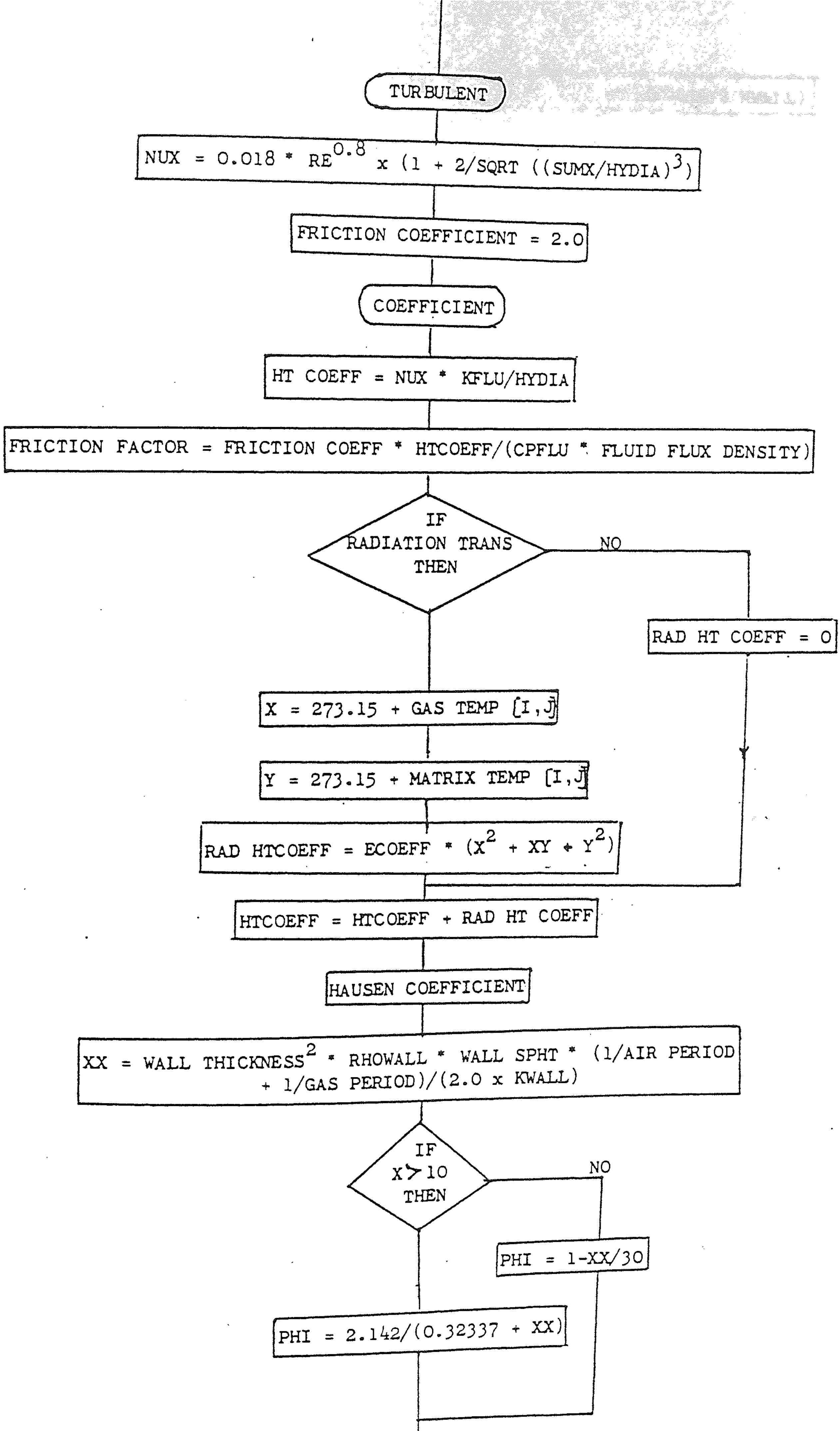

INFO FROM STIR
START
DECLARATIONS - REAL VARIABLES
- INTEGER VARIABLES
- BOOLEAN VARIABLES
- REAL ARRAYS
CP WALL
ABSTEMP = TEMP + 273.15
IF ABSC THEN
NO
X = 300 * (LN. ABSTEMP) - 867
Y = 0.32 * ABSTEMP + 825
CP WALL = 0.27 * X + 0.63 XY
GO TO CP WALL END
IF MAT 2 or 3 THEN
NO
CP WALL = 300 * (LN. ABSTEMP) - 867
GO TO CP WALL END
IF MAT 4 THEN
NO
IF ABSTEMP < 1000
NO
CP WALL = 400 + 0.6 * ABSTEMP

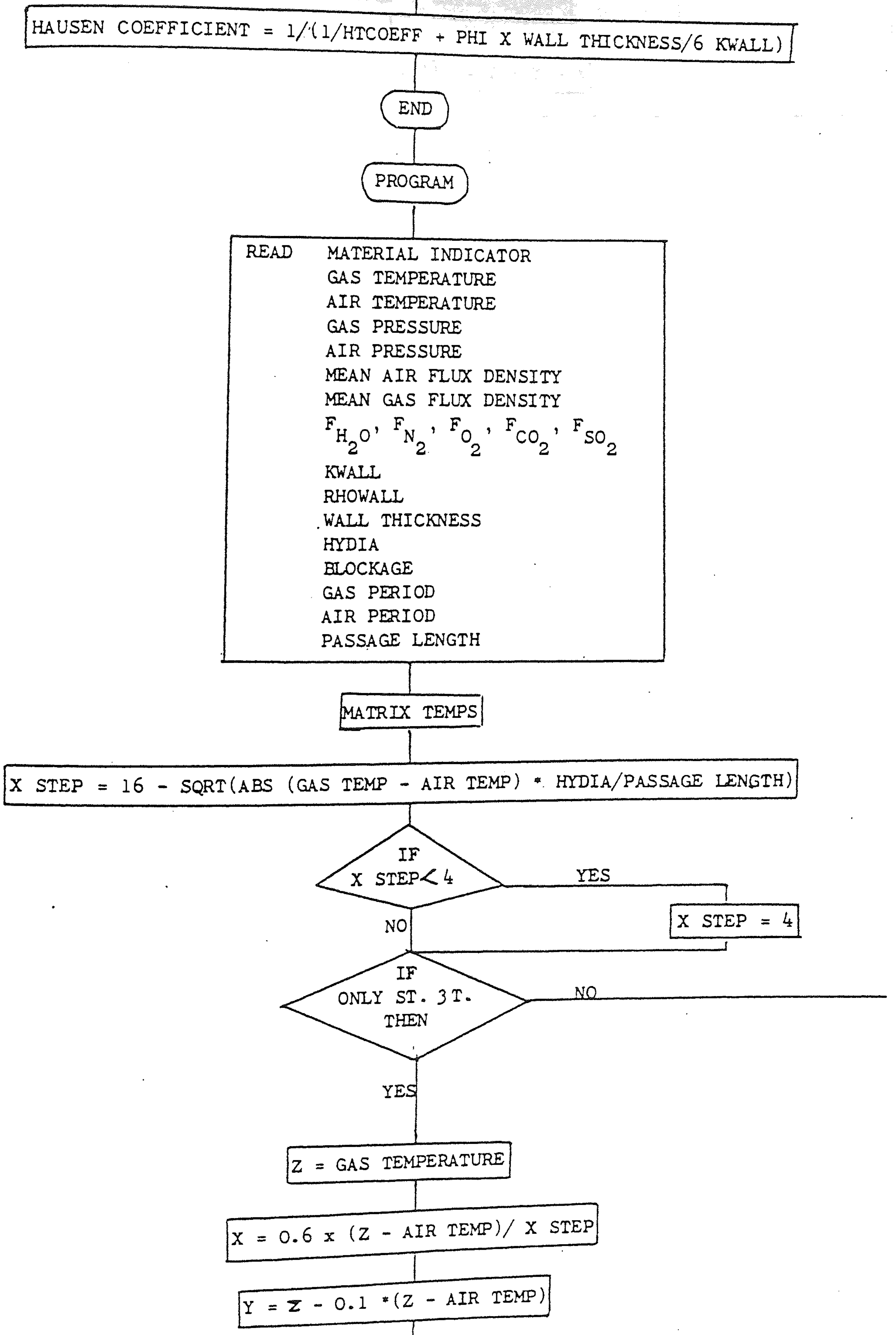

IF ABSTEMP <1225
NO
CP WALL = 1663 - 0.827 * ABSTEMP
CP WALL = 650
GO TO CP WALL END
CP WALL END
DETERMINE GAS PROCEDURES
SUMMON MASS TO VOLUME LIB PROG
DECLARE INPUTS
IF $V_{CO_2} > 0$ THEN
NO
$P_{CO_2} = 0$
P_{CO_2} = LN (GAS PRESS x V_{CO_2} * HYDIA)
IF $V_{H_2O} > 0$ THEN
NO
$P_{H_2O} = 0$
P_{H_2O} = LN (GAS PRESS * V_{H_2O} * HYDIA)

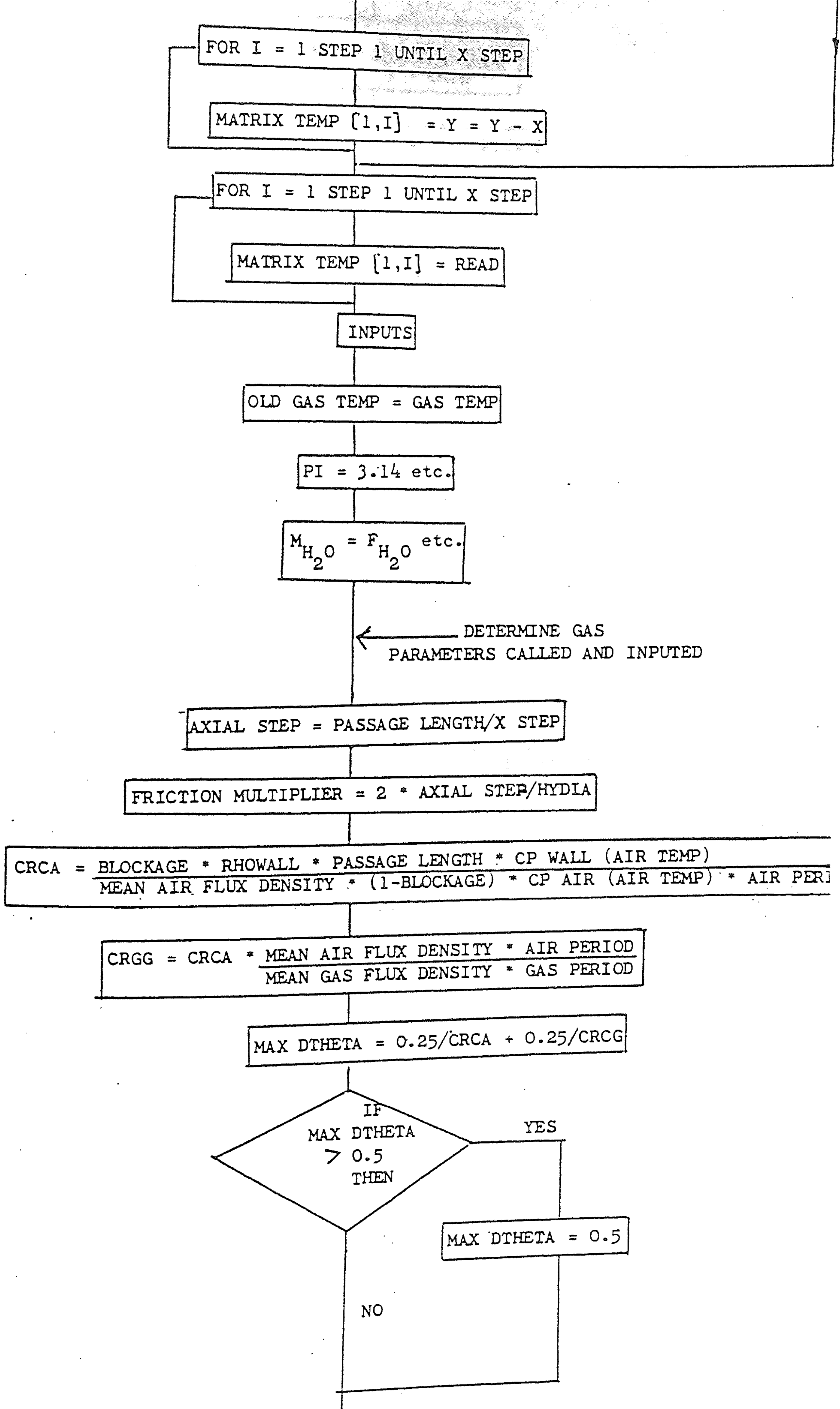


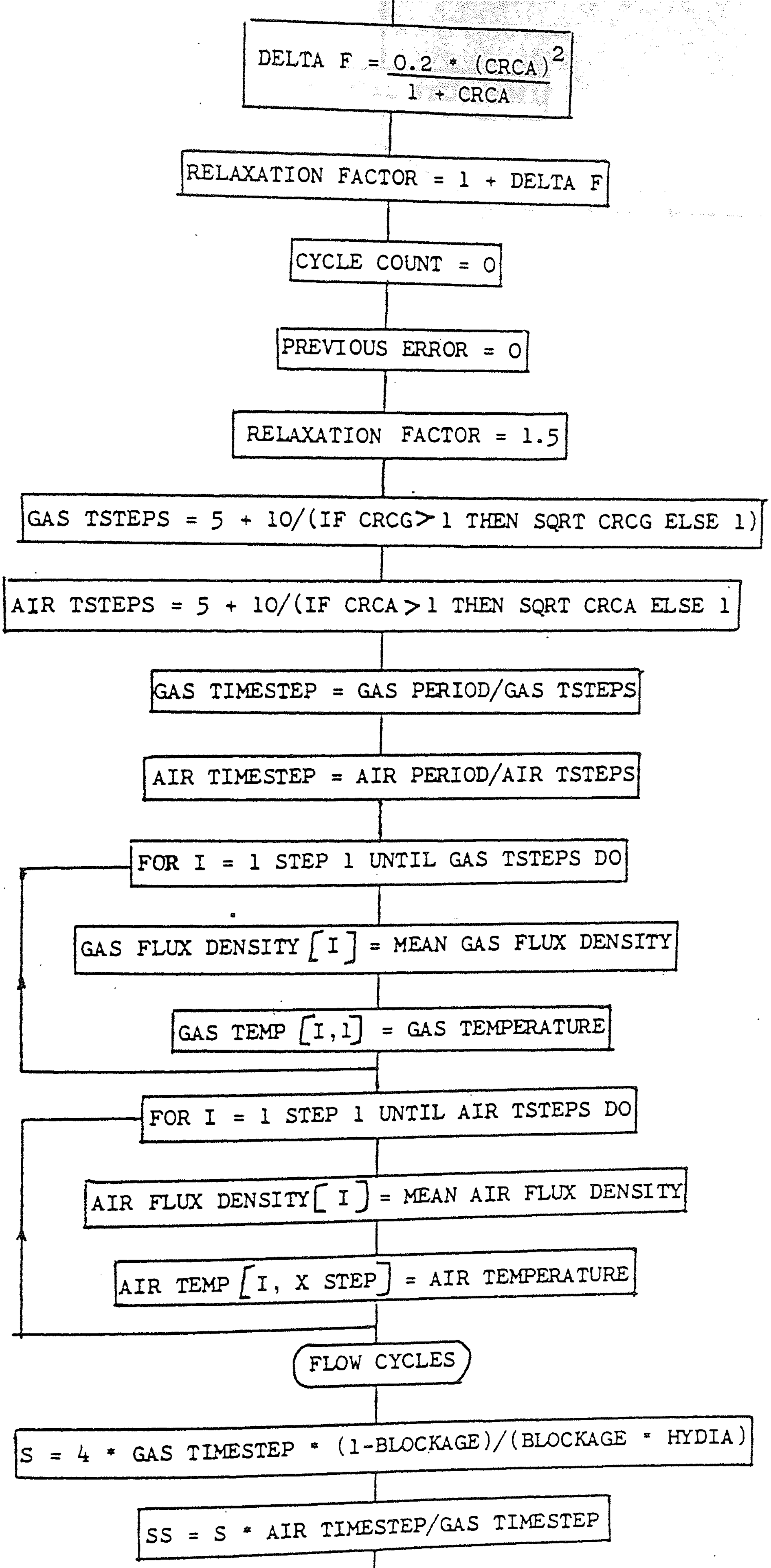
$V = LN\ (0.0000\ 45477 \times (V_{CO_2} + V_{H_2O}) * HYDIA * GAS\ PRESS)$
$X = 2 * V_{H_2O}/(V_{CO_2} + V_{H_2O}) - 1$
$DE = (1 - (X^2)^2) * \exp\ (3.8546 + V * (0.187))$
IF
$PH_2O > 0$
THEN
NO
$E_{H_2O} = 0$
$E_{H_2O} = Exp\ (-1.917112727 + (P_{H_2O} * 0.7273826112))$
IF
$P_{CO_2} > 0$
THEN
NO
$E_{CO_2} = Exp\ (-2.206659659 + P_{CO_2} * (1.25457366 - 0.05\ 385390884 * P_{CO_2}))$
$E_{CO_2} = 0$
$ET = E_{H_2O} + E_{CO_2} - DE$
$ECOEFF = 0.75 * ET \times 5.67 * 10^{-8}$
END
CALCULATE FACTORS
RE = FLUID FLUX DENSITY * HYDIA/MUFLU

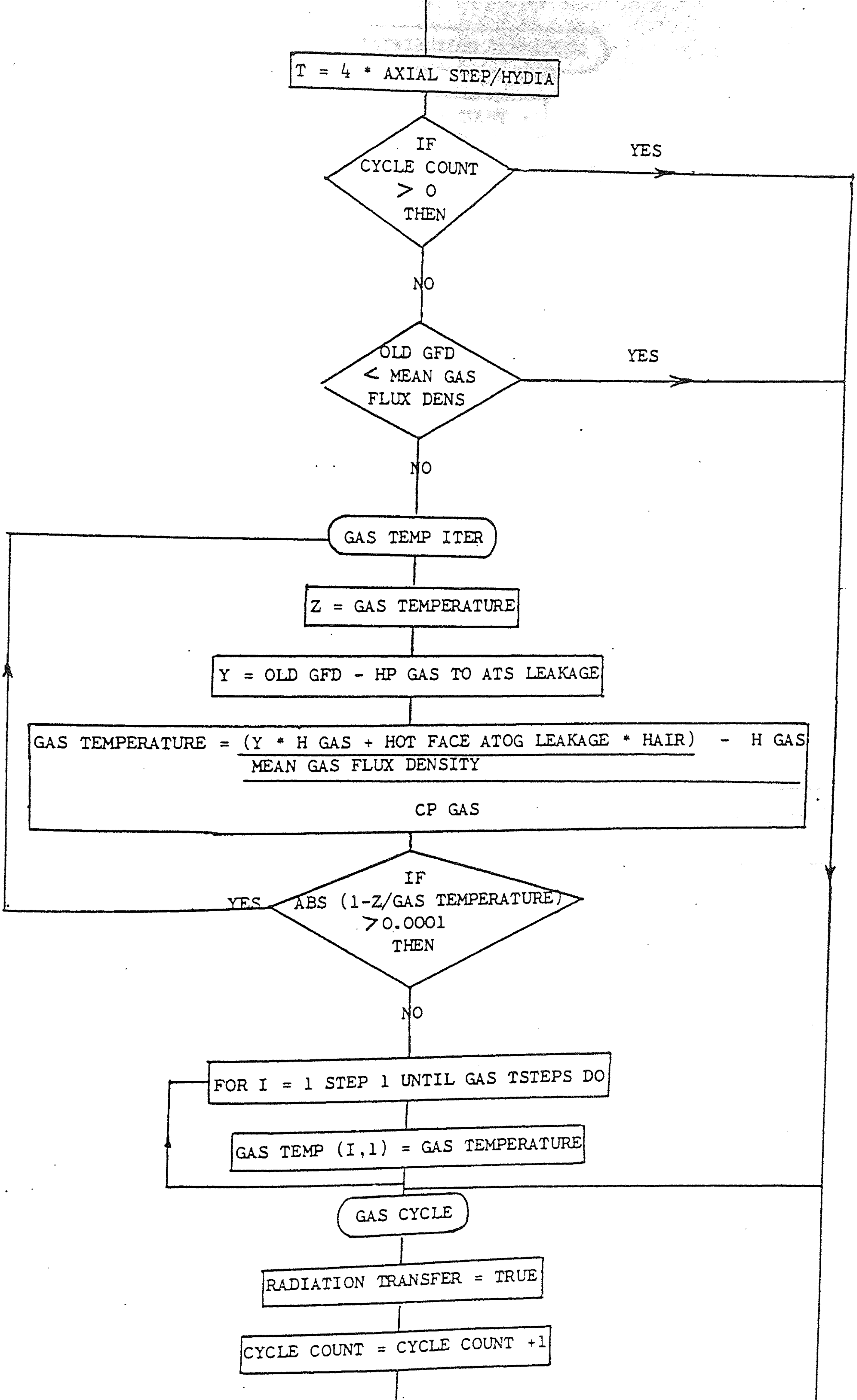

X PLUS = SUMX/(HYDIA * RE)
IF RE > 3000 THEN
GO TO
NO
IF RE > 2000 THEN
GO TO
NO
LAMIN AR
NUX = 3.5 * (1 + 1/(11.95 * SQRT (XPLUS) + 8163 * (XPLUS)^2))
FRICTION COEFFICIENT = 2.68
GO TO COEFFICIENT
TRANSITION
RED = RE/1000
X = 3 - RED
Y = RED - 2
NUX = (3.5 * X + 0.018 * RE^0.8 *Y) * ((1 + 1/(11.95 * SORT(XPLUS) + 8163 * (XPLUS)^2)) * X + (1 + 2/SQRT((SUMX HYDIA)^3)) *Y)
FRICTION COEFFICIENT = 2 + 0.68 * X
GO TO COEFFICIENT


TURBULENT
NUX = 0.018 * RE^0.8 x (1 + 2/SQRT ((SUMX/HYDIA)^3)
FRICTION COEFFICIENT = 2.0
COEFFICIENT
HT COEFF = NUX * KFLU/HYDIA
FRICTION FACTOR = FRICTION COEFF * HTCOEFF/(CPFLU * FLUID FLUX DENSITY)
IF RADIATION TRANS THEN
NO
RAD HT COEFF = 0
X = 273.15 + GAS TEMP [I,J]
Y = 273.15 + MATRIX TEMP [I,J]
RAD HTCOEFF = ECOEFF * (X^2 + XY + Y^2)
HTCOEFF = HTCOEFF + RAD HT COEFF
HAUSEN COEFFICIENT
XX = WALL THICKNESS^2 * RHOWALL * WALL SPHT * (1/AIR PERIOD + 1/GAS PERIOD)/(2.0 x KWALL)
IF X > 10 THEN
NO
PHI = 1-XX/30
PHI = 2.142/(0.32337 + XX)


HAUSEN COEFFICIENT = 1/(1/HTCOEFF + PHI X WALL THICKNESS/6 KWALL)
END
PROGRAM
READ MATERIAL INDICATOR
GAS TEMPERATURE
AIR TEMPERATURE
GAS PRESSURE
AIR PRESSURE
MEAN AIR FLUX DENSITY
MEAN GAS FLUX DENSITY
F_{H_2O}, F_{N_2}, F_{O_2}, F_{CO_2}, F_{SO_2}
KWALL
RHOWALL
WALL THICKNESS
HYDIA
BLOCKAGE
GAS PERIOD
AIR PERIOD
PASSAGE LENGTH
MATRIX TEMPS
X STEP = 16 - SQRT(ABS (GAS TEMP - AIR TEMP) * HYDIA/PASSAGE LENGTH)
IF X STEP < 4
YES
X STEP = 4
NO
IF ONLY ST. 3 T. THEN
NO
YES
Z = GAS TEMPERATURE
X = 0.6 x (Z - AIR TEMP)/ X STEP
Y = Z - 0.1 *(Z - AIR TEMP)


FOR I = 1 STEP 1 UNTIL X STEP
MATRIX TEMP [1,I] = Y = Y - X
FOR I = 1 STEP 1 UNTIL X STEP
MATRIX TEMP [1,I] = READ
INPUTS
OLD GAS TEMP = GAS TEMP
PI = 3.14 etc.
$M_{H_2O} = F_{H_2O}$ etc.
DETERMINE GAS
PARAMETERS CALLED AND INPUTED
AXIAL STEP = PASSAGE LENGTH/X STEP
FRICTION MULTIPLIER = 2 * AXIAL STEP/HYDIA
CRCA = BLOCKAGE * RHOWALL * PASSAGE LENGTH * CP WALL (AIR TEMP) / MEAN AIR FLUX DENSITY * (1-BLOCKAGE) * CP AIR (AIR TEMP) * AIR PERI
CRGG = CRCA * MEAN AIR FLUX DENSITY * AIR PERIOD / MEAN GAS FLUX DENSITY * GAS PERIOD
MAX DTHETA = 0.25/CRCA + 0.25/CRCG
IF MAX DTHETA > 0.5 THEN
YES
MAX DTHETA = 0.5
NO


DELTA F = 0.2 * (CRCA)2 / 1 + CRCA
RELAXATION FACTOR = 1 + DELTA F
CYCLE COUNT = 0
PREVIOUS ERROR = 0
RELAXATION FACTOR = 1.5
GAS TSTEPS = 5 + 10/(IF CRCG>1 THEN SQRT CRCG ELSE 1)
AIR TSTEPS = 5 + 10/(IF CRCA>1 THEN SQRT CRCA ELSE 1
GAS TIMESTEP = GAS PERIOD/GAS TSTEPS
AIR TIMESTEP = AIR PERIOD/AIR TSTEPS
FOR I = 1 STEP 1 UNTIL GAS TSTEPS DO
GAS FLUX DENSITY [I] = MEAN GAS FLUX DENSITY
GAS TEMP [I,1] = GAS TEMPERATURE
FOR I = 1 STEP 1 UNTIL AIR TSTEPS DO
AIR FLUX DENSITY [I] = MEAN AIR FLUX DENSITY
AIR TEMP [I, X STEP] = AIR TEMPERATURE
FLOW CYCLES
S = 4 * GAS TIMESTEP * (1-BLOCKAGE)/(BLOCKAGE * HYDIA)
SS = S * AIR TIMESTEP/GAS TIMESTEP



T = 4 * AXIAL STEP/HYDIA
IF CYCLE COUNT > 0 THEN
YES
NO
OLD GFD < MEAN GAS FLUX DENS
YES
NO
GAS TEMP ITER
Z = GAS TEMPERATURE
Y = OLD GFD - HP GAS TO ATS LEAKAGE
GAS TEMPERATURE = (Y * H GAS + HOT FACE ATOG LEAKAGE * HAIR) / MEAN GAS FLUX DENSITY - H GAS / CP GAS
IF ABS (1-Z/GAS TEMPERATURE) >0.0001 THEN
YES
NO
FOR I = 1 STEP 1 UNTIL GAS TSTEPS DO
GAS TEMP (I,1) = GAS TEMPERATURE
GAS CYCLE
RADIATION TRANSFER = TRUE
CYCLE COUNT = CYCLE COUNT +1

GAS FLOW DISTRIBUTION ITERATION

GM COUNT = GM COUNT +1

FOR I = STEP 1 UNTIL GAS TSTEPS DO

GAS PRESSURE EQUALISATION ITERATION

GP COUNT = GP COUNT +1

SUM X = -0.5 * AXIAL STEP

$$\text{PRESSURE} = \text{GAS PRESSURE} - \frac{\text{BLOCKAGE} * \text{GAS FLUX DENSITY (I)}^2}{4 * \text{RHOGAS}}$$

FOR J = 1 STEP 1 UNTIL X STEP DO

SUM X = SUM X + AXIAL STEP

MUFLU = MU GAS

CP FLU = CP GAS

$$R = \frac{8314.4}{(V_{H_2O} * 18.016 + V_{N_2} * 28.164 + V_{O_2} * 32.0 + V_{CO_2} * 44.01 + V_{SO_2} * 64.07)}$$

K FLU = MUFLU * (CPFLUE + 1.25*R)

$$\text{RHOFLU} = \frac{\text{PRESSURE}}{R * (273.15 + \text{GAS TEMP } [I,J])}$$

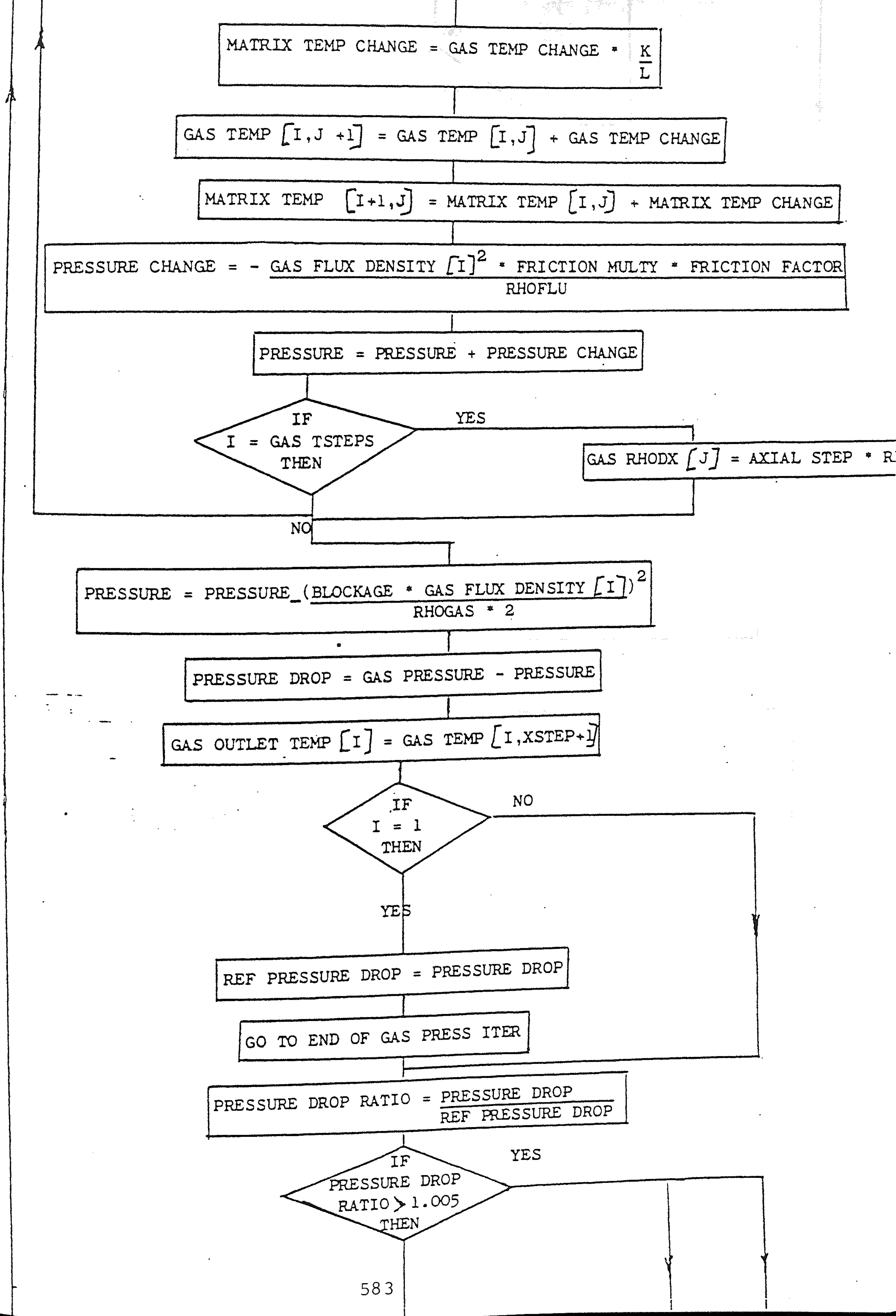
WALL SPHT = CP WALL (MATRIX TEMP (J+J))

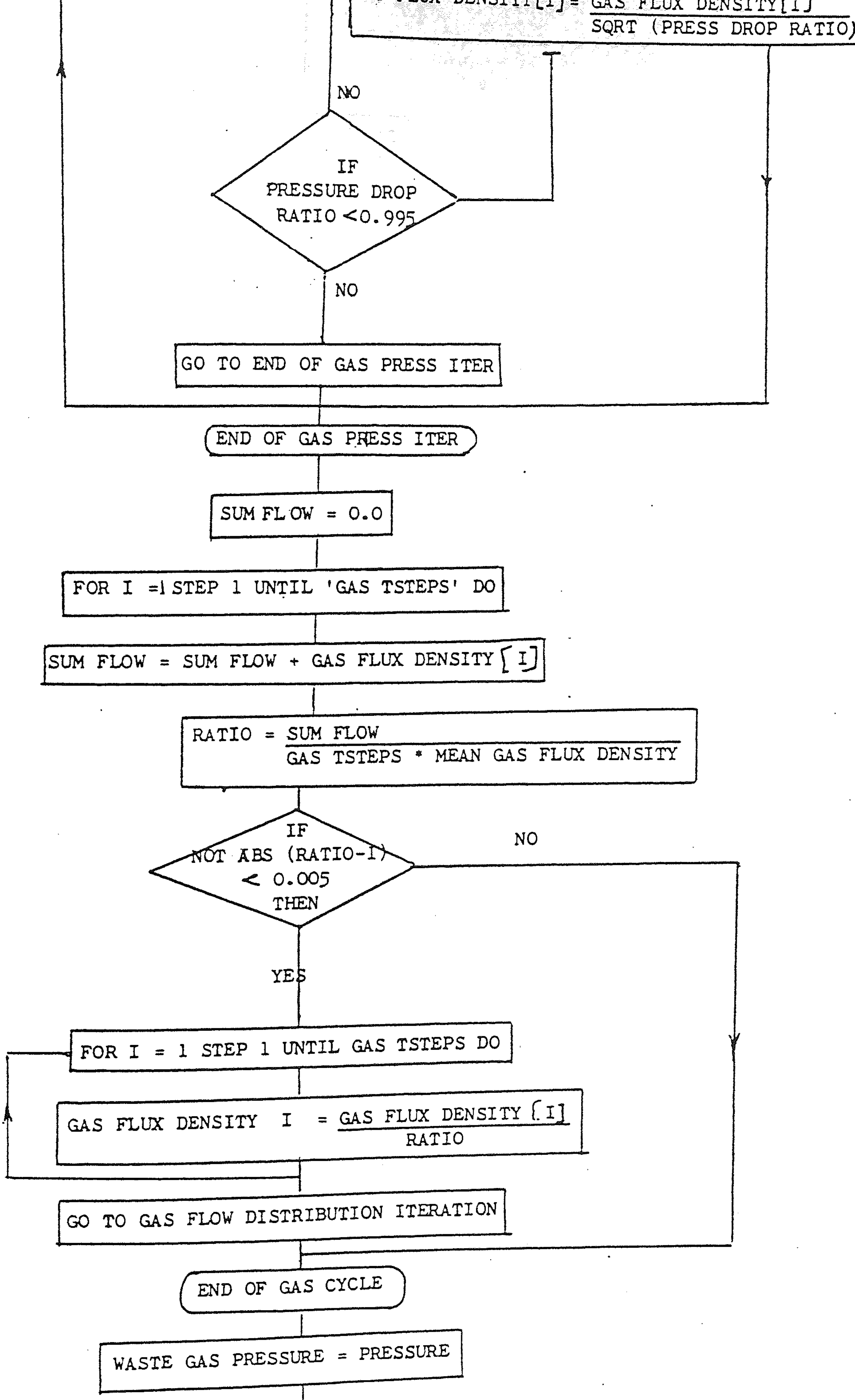
CALCULATE FACTORS (GAS FLUX DENSITY [I])

$$K = \frac{S * \text{HAUSEN HT COEFF}}{\text{RHOWALL} * \text{WALL SPHT}}$$

$$L = \frac{T * \text{HAUSEN HT COEFF}}{\text{GAS FLUX DENSITY } [I] * \text{CPFLU}}$$

$$\text{GAS TEMP CHANGE} = \frac{L * (\text{MATRIX TEMP } [I,J] - \text{GAS TEMP } [I,J])}{1 + \frac{L + K}{2}}$$


MATRIX TEMP CHANGE = GAS TEMP CHANGE * K/L
GAS TEMP [I,J +1] = GAS TEMP [I,J] + GAS TEMP CHANGE
MATRIX TEMP [I+1,J] = MATRIX TEMP [I,J] + MATRIX TEMP CHANGE
PRESSURE CHANGE = - GAS FLUX DENSITY [I]^2 * FRICTION MULTY * FRICTION FACTOR / RHOFLU
PRESSURE = PRESSURE + PRESSURE CHANGE
IF I = GAS TSTEPS THEN
YES
GAS RHODX [J] = AXIAL STEP * R
NO
PRESSURE = PRESSURE _ (BLOCKAGE * GAS FLUX DENSITY [I])^2 / RHOGAS * 2
PRESSURE DROP = GAS PRESSURE - PRESSURE
GAS OUTLET TEMP [I] = GAS TEMP [I,XSTEP+1]
IF I = 1 THEN
NO
YES
REF PRESSURE DROP = PRESSURE DROP
GO TO END OF GAS PRESS ITER
PRESSURE DROP RATIO = PRESSURE DROP / REF PRESSURE DROP
IF PRESSURE DROP RATIO > 1.005 THEN
YES


FLUX DENSITY[I] = GAS FLUX DENSITY[I] / SQRT (PRESS DROP RATIO)
NO
IF PRESSURE DROP RATIO <0.995
NO
GO TO END OF GAS PRESS ITER
END OF GAS PRESS ITER
SUM FLOW = 0.0
FOR I =1 STEP 1 UNTIL 'GAS TSTEPS' DO
SUM FLOW = SUM FLOW + GAS FLUX DENSITY [I]
RATIO = SUM FLOW / GAS TSTEPS * MEAN GAS FLUX DENSITY
IF NOT ABS (RATIO-1) < 0.005 THEN
NO
YES
FOR I = 1 STEP 1 UNTIL GAS TSTEPS DO
GAS FLUX DENSITY I = GAS FLUX DENSITY [I] / RATIO
GO TO GAS FLOW DISTRIBUTION ITERATION
END OF GAS CYCLE
WASTE GAS PRESSURE = PRESSURE

PRINT G.M. COUNT
G.P. COUNT

AM COUNT = AP COUNT = 0

AIR CYCLE

RADIATION TRANSFER = FALSE

AIRFLOW DISTRIBUTION ITERATION

AM COUNT = AM COUNT +1

FOR I = 1 STEP 1 UNTIL AIR TSTEPS DO

AIR PRESSURE EQUALISATION ITERATION

AP COUNT = AP COUNT +1

H = GAS TSTEPS +I

SUMX = -0.5 * AXIAL STEP

PRESSURE = AIR PRESSURE

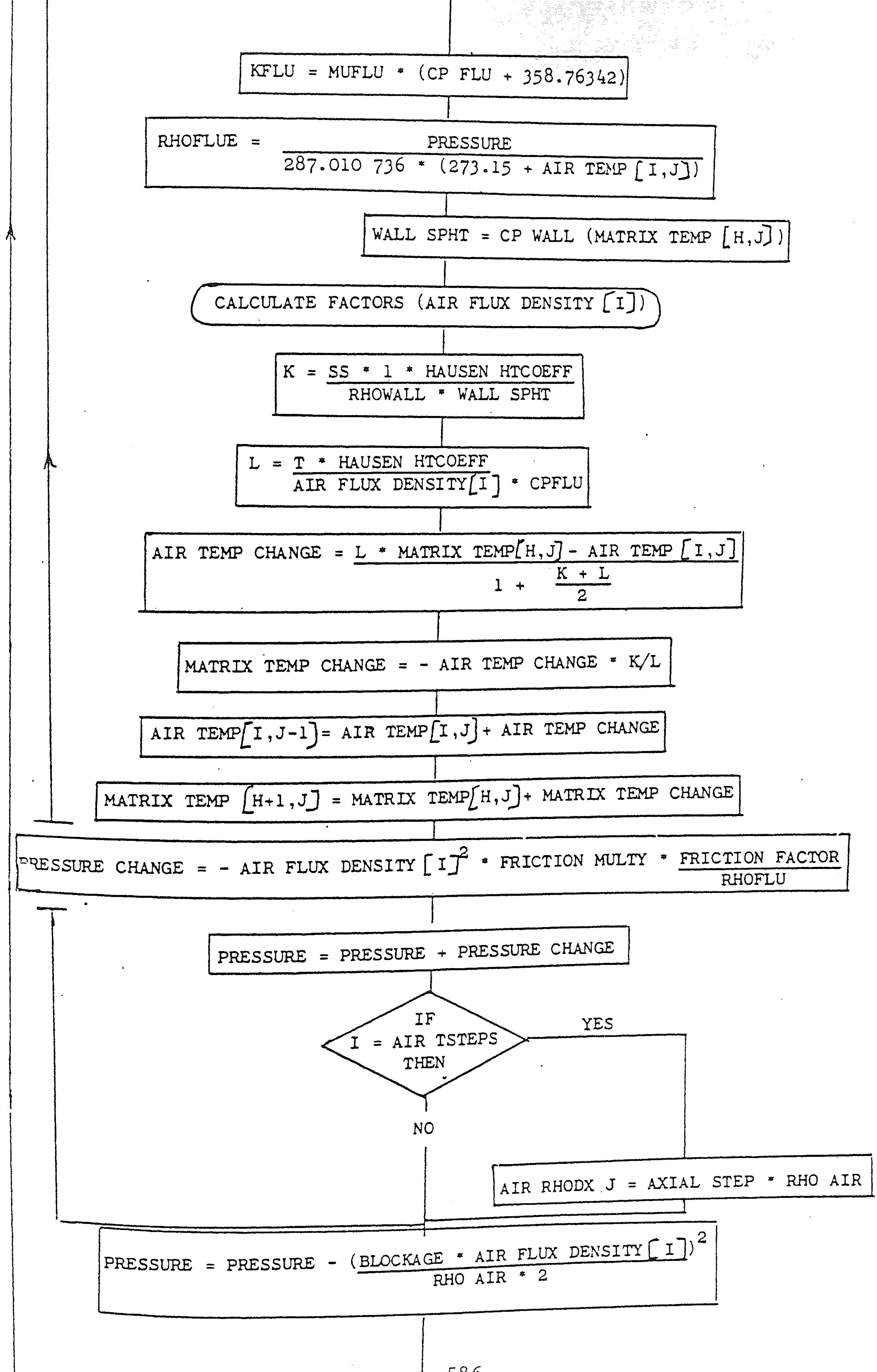
$$\text{PRESSURE} = \text{PRESSURE} - \frac{\text{BLOCKAGE} * \text{AIR FLUX DENSITY}\,[\text{I}]^2}{4 * \text{RHO AIR}}$$

FOR J = XSTEP STEP -1 UNTIL 1 DO

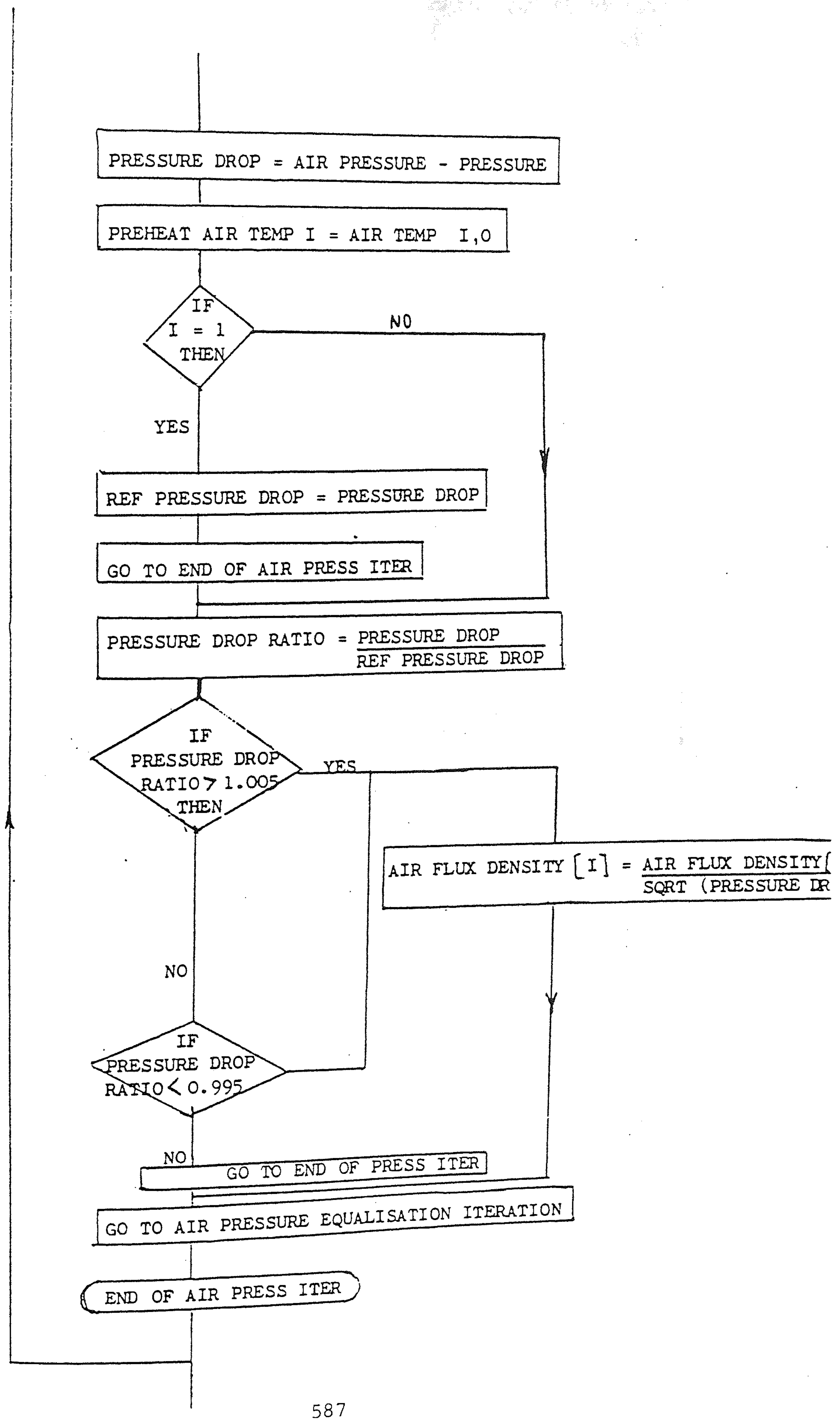
SUMX = SUMX + AXIAL STEP

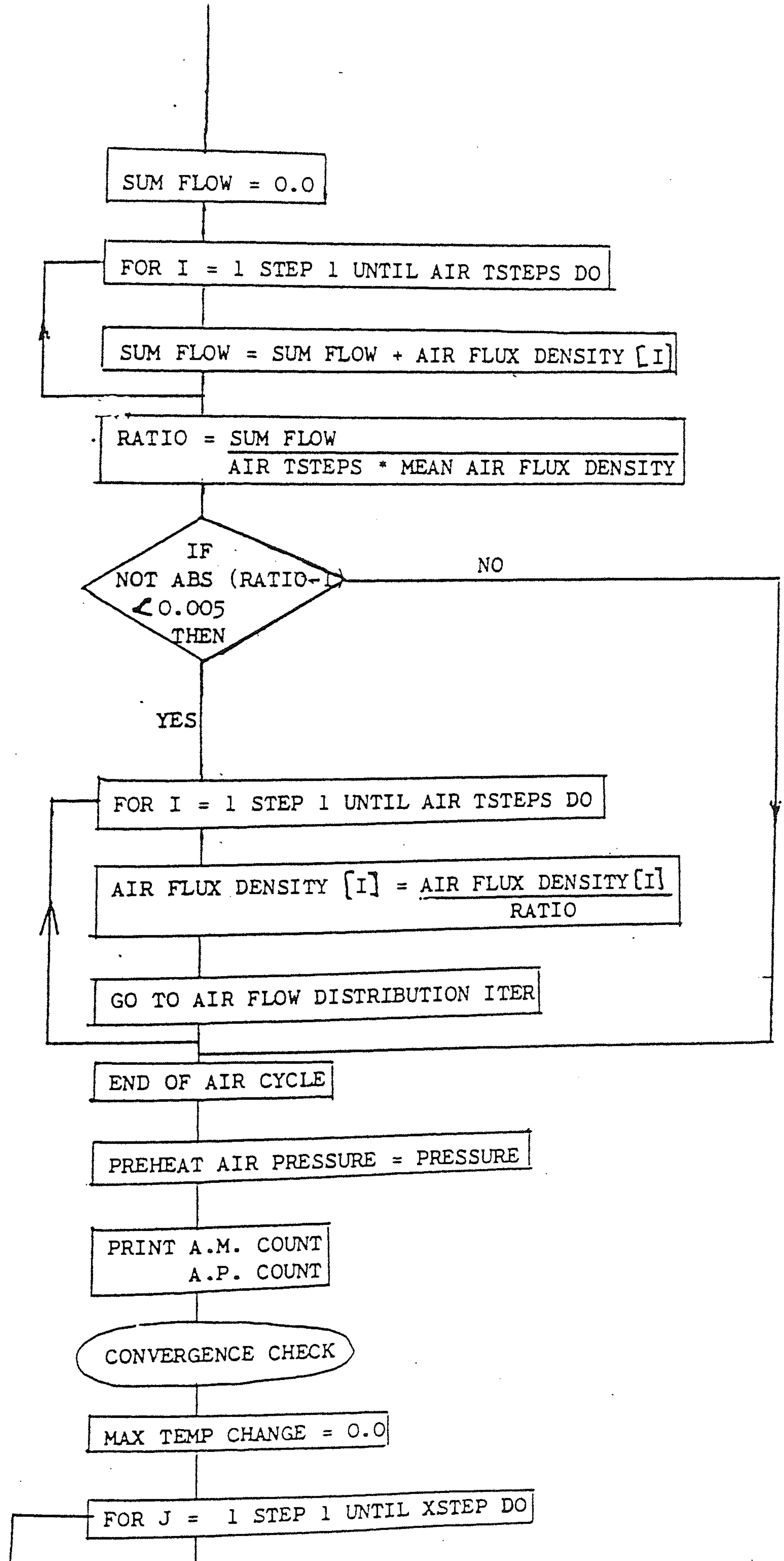
MUFLU = MU AIR

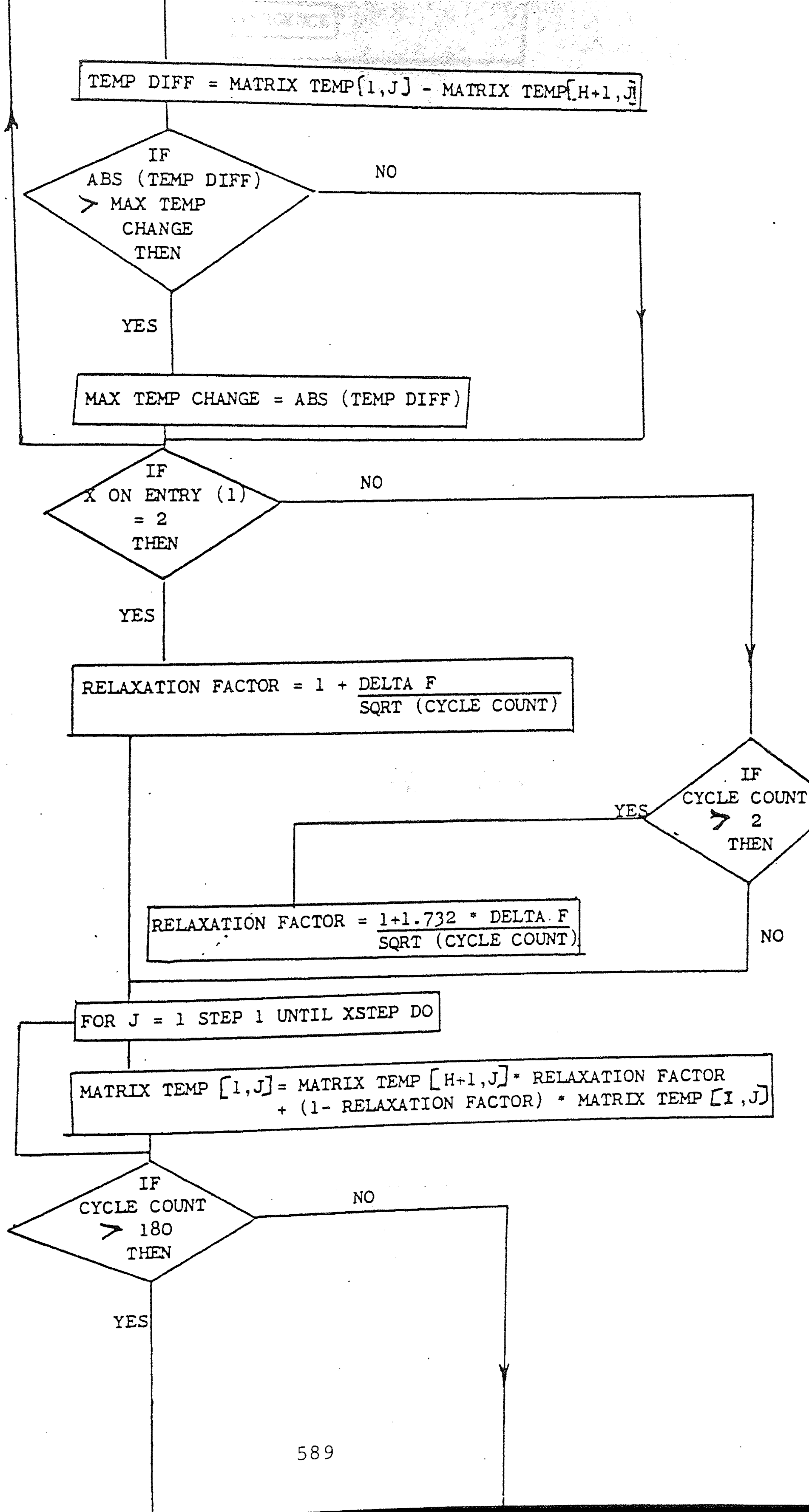
CPFLU = CP AIR

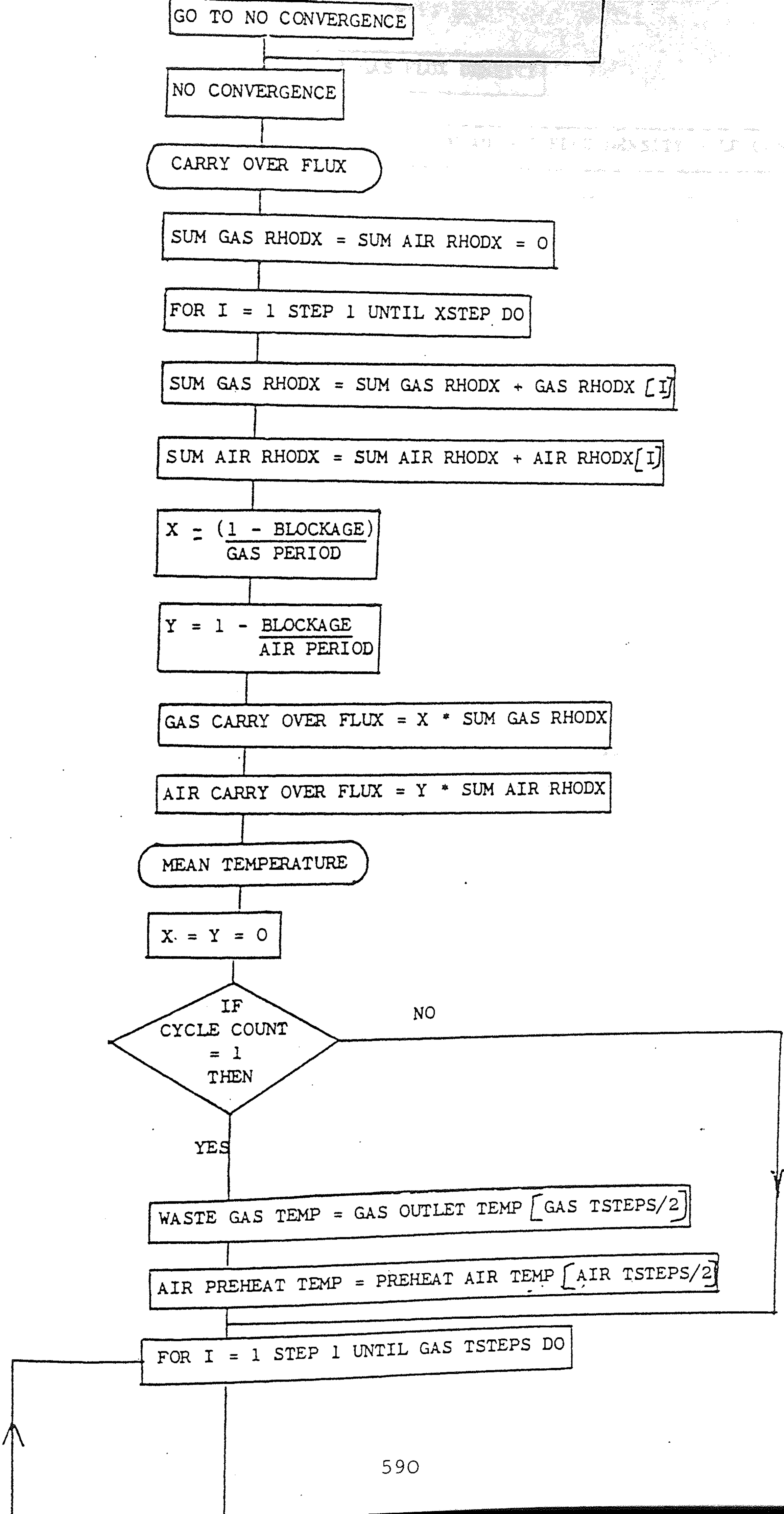


KFLU = MUFLU * (CP FLU + 358.76342)
RHOFLUE = PRESSURE / 287.010 736 * (273.15 + AIR TEMP [I,J])
WALL SPHT = CP WALL (MATRIX TEMP [H,J])
CALCULATE FACTORS (AIR FLUX DENSITY [I])
K = SS * 1 * HAUSEN HTCOEFF / RHOWALL * WALL SPHT
L = T * HAUSEN HTCOEFF / AIR FLUX DENSITY[I] * CPFLU
AIR TEMP CHANGE = L * MATRIX TEMP[H,J] - AIR TEMP [I,J] / 1 + K + L / 2
MATRIX TEMP CHANGE = - AIR TEMP CHANGE * K/L
AIR TEMP[I,J-1]= AIR TEMP[I,J]+ AIR TEMP CHANGE
MATRIX TEMP [H+1,J] = MATRIX TEMP[H,J]+ MATRIX TEMP CHANGE
PRESSURE CHANGE = - AIR FLUX DENSITY [I]2 * FRICTION MULTY * FRICTION FACTOR / RHOFLU
PRESSURE = PRESSURE + PRESSURE CHANGE
IF I = AIR TSTEPS THEN
YES
NO
AIR RHODX J = AXIAL STEP * RHO AIR
PRESSURE = PRESSURE - (BLOCKAGE * AIR FLUX DENSITY [I])2 / RHO AIR * 2


PRESSURE DROP = AIR PRESSURE - PRESSURE
PREHEAT AIR TEMP I = AIR TEMP I,0
IF I = 1 THEN
NO
YES
REF PRESSURE DROP = PRESSURE DROP
GO TO END OF AIR PRESS ITER
PRESSURE DROP RATIO = PRESSURE DROP / REF PRESSURE DROP
IF PRESSURE DROP RATIO > 1.005 THEN
YES
AIR FLUX DENSITY [I] = AIR FLUX DENSITY[/ SQRT (PRESSURE DR
NO
IF PRESSURE DROP RATIO < 0.995
NO
GO TO END OF PRESS ITER
GO TO AIR PRESSURE EQUALISATION ITERATION
END OF AIR PRESS ITER


SUM FLOW = 0.0
FOR I = 1 STEP 1 UNTIL AIR TSTEPS DO
SUM FLOW = SUM FLOW + AIR FLUX DENSITY [I]
RATIO = SUM FLOW / AIR TSTEPS * MEAN AIR FLUX DENSITY
IF NOT ABS (RATIO-1) < 0.005 THEN
NO
YES
FOR I = 1 STEP 1 UNTIL AIR TSTEPS DO
AIR FLUX DENSITY [I] = AIR FLUX DENSITY [I] / RATIO
GO TO AIR FLOW DISTRIBUTION ITER
END OF AIR CYCLE
PREHEAT AIR PRESSURE = PRESSURE
PRINT A.M. COUNT
A.P. COUNT
CONVERGENCE CHECK
MAX TEMP CHANGE = 0.0
FOR J = 1 STEP 1 UNTIL XSTEP DO


TEMP DIFF = MATRIX TEMP[1,J] - MATRIX TEMP[H+1,J]
IF ABS (TEMP DIFF) > MAX TEMP CHANGE THEN
NO
YES
MAX TEMP CHANGE = ABS (TEMP DIFF)
IF X ON ENTRY (1) = 2 THEN
NO
YES
RELAXATION FACTOR = 1 + DELTA F / SQRT (CYCLE COUNT)
IF CYCLE COUNT > 2 THEN
YES
NO
RELAXATION FACTOR = 1+1.732 * DELTA F / SQRT (CYCLE COUNT)
FOR J = 1 STEP 1 UNTIL XSTEP DO
MATRIX TEMP [1,J] = MATRIX TEMP [H+1,J] * RELAXATION FACTOR + (1- RELAXATION FACTOR) * MATRIX TEMP [I,J]
IF CYCLE COUNT > 180 THEN
NO
YES


GO TO NO CONVERGENCE
NO CONVERGENCE
CARRY OVER FLUX
SUM GAS RHODX = SUM AIR RHODX = 0
FOR I = 1 STEP 1 UNTIL XSTEP DO
SUM GAS RHODX = SUM GAS RHODX + GAS RHODX [I]
SUM AIR RHODX = SUM AIR RHODX + AIR RHODX[I]
X = (1 - BLOCKAGE) / GAS PERIOD
Y = 1 - BLOCKAGE / AIR PERIOD
GAS CARRY OVER FLUX = X * SUM GAS RHODX
AIR CARRY OVER FLUX = Y * SUM AIR RHODX
MEAN TEMPERATURE
X = Y = 0
IF CYCLE COUNT = 1 THEN
NO
YES
WASTE GAS TEMP = GAS OUTLET TEMP [GAS TSTEPS/2]
AIR PREHEAT TEMP = PREHEAT AIR TEMP [AIR TSTEPS/2]
FOR I = 1 STEP 1 UNTIL GAS TSTEPS DO

X = X + H GAS * GAS FLUX DENSITY

MEAN GAS FLUX DENSITY = MEAN GAS FLUX DENSITY - LP GAS TO ATS L

Z = MEAN GAS FLUX DENSITY + COLD FACE ATOG LEAKAGE

$$X = \frac{\frac{X}{\text{GAS TSTEPS}} + \text{HAIR} * \text{COLD FACE ATOG LEAKAGE} - \text{HGAS} * \text{LPGAS TO ATS}}{Z}$$

IF Z > MEAN GAS FLUX DENSITY THEN

NO

YES

$$M_{H_2O} = M_{H_2O} \frac{* \text{ MEAN GAS FLUX DENSITY}}{Z}$$

$$M_{N_2} = \frac{(M_{N_2} - 0.764) * \text{MEAN GAS FLUX DENSITY}}{Z} + 0.764$$

$$M_{O_2} = \frac{(M_{O_2} - 0.232) * \text{MEAN GAS FLUX DENSITY}}{Z} + 0.232$$

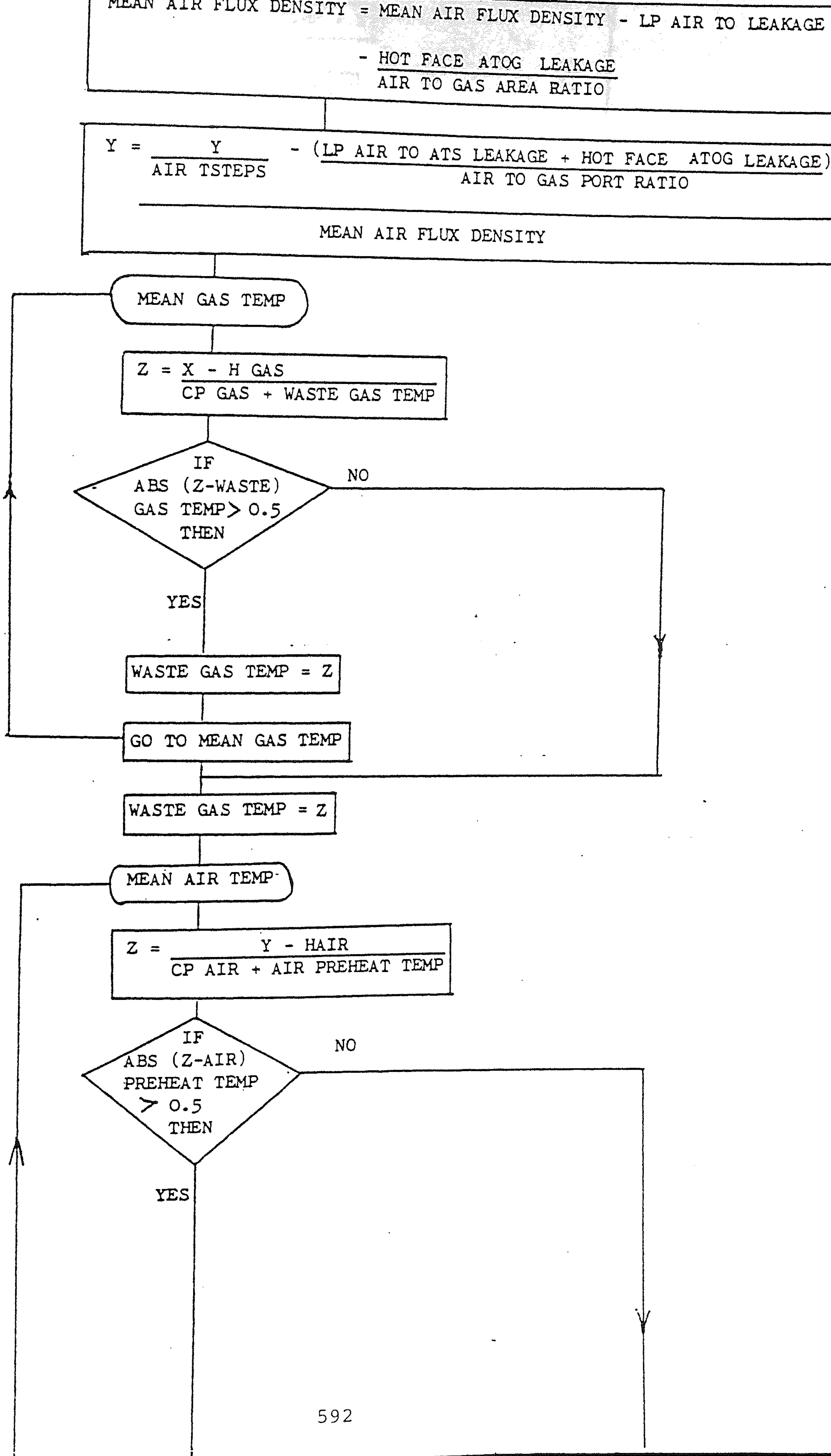
$$M_{CO_2} = \frac{(M_{CO_2} - 0.004) * \text{MEAN GAS FLUX DENSITY}}{Z} + 0.004$$

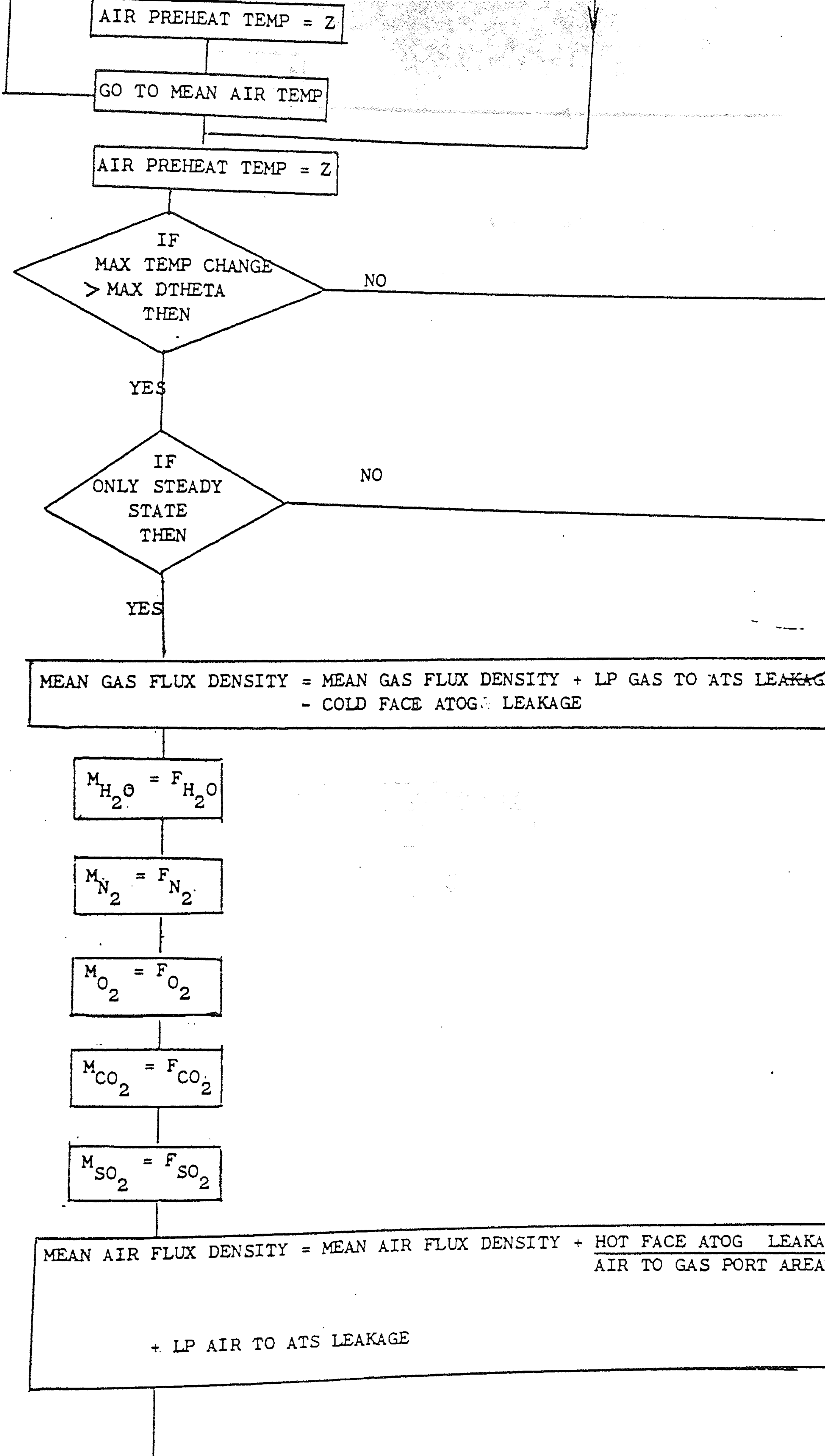
$$M_{SO_2} = M_{SO_2} \frac{* \text{ MEAN GAS FLUX DENSITY}}{Z}$$

MEAN FLUX DENSITY = Z

FOR I = 1 STEP 1 UNTIL AIR TSTEPS DO

Y = Y + HAIR * AIR FLUX DENSITY


MEAN AIR FLUX DENSITY = MEAN AIR FLUX DENSITY - LP AIR TO LEAKAGE
- HOT FACE ATOG LEAKAGE
AIR TO GAS AREA RATIO
Y = Y
AIR TSTEPS
- (LP AIR TO ATS LEAKAGE + HOT FACE ATOG LEAKAGE)
AIR TO GAS PORT RATIO
MEAN AIR FLUX DENSITY
MEAN GAS TEMP
Z = X - H GAS
CP GAS + WASTE GAS TEMP
IF
ABS (Z-WASTE)
GAS TEMP> 0.5
THEN
NO
YES
WASTE GAS TEMP = Z
GO TO MEAN GAS TEMP
WASTE GAS TEMP = Z
MEAN AIR TEMP
Z = Y - HAIR
CP AIR + AIR PREHEAT TEMP
IF
ABS (Z-AIR)
PREHEAT TEMP
> 0.5
THEN
NO
YES


AIR PREHEAT TEMP = Z
GO TO MEAN AIR TEMP
AIR PREHEAT TEMP = Z
IF MAX TEMP CHANGE > MAX DTHETA THEN
NO
YES
IF ONLY STEADY STATE THEN
NO
YES
MEAN GAS FLUX DENSITY = MEAN GAS FLUX DENSITY + LP GAS TO ATS LEAKAGE
- COLD FACE ATOG LEAKAGE
$M_{H_2O} = F_{H_2O}$
$M_{N_2} = F_{N_2}$
$M_{O_2} = F_{O_2}$
$M_{CO_2} = F_{CO_2}$
$M_{SO_2} = F_{SO_2}$
MEAN AIR FLUX DENSITY = MEAN AIR FLUX DENSITY + HOT FACE ATOG LEAKA
AIR TO GAS PORT AREA
+ LP AIR TO ATS LEAKAGE

GO TO FLOW CYCLES

EFFECTIVENESS

(AIR TEMPERATURE, AIR.PREHEAT T

AIR SIDE EFFECTIVENESS = $\frac{\text{H AIR}}{\text{H AIR}}$

(AIR TEMPERATURE OLD GAS TEMP)

(WASTE GAS TEMP, GAS TEMPERATUR

GAS SIDE EFFECTIVENESS = $\frac{\text{H GAS}}{\text{H GAS}}$

(AIR TEMPERATURE, OLD GAS TEMP)

AIR FLUX RATIO = $\frac{\text{MEAN AIR FLUX DENSITY}}{\text{OLD AFD}}$

GAS FLUX RATIO = $\frac{\text{MEAN GAS FLUX DENSITY}}{\text{OLD GFD}}$

ROTARY REG REFRACTORY UTILISATION

REFRACTORY UTILISATION = MEAN AIR FLUX DENSITY * (1-BLOCKAGE)* H
RHOWALL * BLOCKAGE * PASSAGE LENGTH * GA
AIR PER

OUTPUTS

4. POUR OUTPUT

The 'POUR' simply prints the results from 'MASH OUTPUT' into the parameters shown below. STIR is then called again and the whole process repeated for the repeated data items.

- STIR DATA
- Number of iterations before MASH was poured
- Matrix Temperatures (^{o}C) Gas Cycle
- Number of Time-steps
- Matrix Temperatures (^{o}C) Air Cycle
- Number of Time-steps
- Gas Temperature (^{o}C)
- Air Temperature (^{o}C)
- Waste Gas Temperature (^{o}C)
- Air Preheat Temperature (^{o}C)
- Gas Flux Density (kg/s.m^{2})
- Air Flux Density (kg/s.m^{2})
- Air Side Effectiveness
- Gas Side Effectiveness
- Air Flux Ratio
- Gas Flux Ratio
- Passage Length (M)
- Gas Carry-over (Kg/s.m^{2})
- Air Carry-over (kg/s.m^{2})
- Solid/Air 'Capacity Rate' Ratio
- Solid/Gas 'Capacity Rate' Ratio
- Passage Length/Hydraulic Dia.

- Refrectory Utilisation (W/kg)
- Gas Pressure (N/m^2)
- Waste Gas Pressure (N/m^2)
- Gas Pressure Drop (N/m^2)
- Air Pressure (N/m^2)
- Preheat Air Pressure (N/m^2)
- Air Pressure Drop (N/m^2)
- STIR DATA

5. THEORETICAL ASSUMPTIONS AND APPROXIMATIONS

Specific Heat:

In calculating specific heats the equation CP WALL = 300 LN (ABS TEMP) - 867 is used. It is recognised that the logrithmic curve is only an approximation and that the actual specific heat/temperature curves are slightly more complicated.

However, as can be seen from Fig. E1, over the operation range of $400^{o}C$ - $1400^{o}C$ silliminite, our selected material, offers good approximation values though the fit for mullite at high temperatures seems suspect.

Determination of Flow Regions:

The flow regions of any fluid can be categorised as follows:

		Reynold's Number (Re)
(a)	Laminar Flow	to 2000
(b)	Transition Flow	2000 to 3000
(c)	Turbulent Flow	3000 to -

Due to small size holes (6 mm) and low flow rates, the pilot rig offers $Re \sim 90$. Nuselt's number was calculated using an empirical formula:

$$Nu = \frac{h\,D}{k} = 3.5 \quad 1 + \frac{\quad}{11.95\sqrt{\frac{L}{DRe}} + 8163\left(\frac{L}{DRe}\right)^{2}}$$

$$= 3.5 \quad (1 + \alpha)$$

where L = SUMx distance down the hole

D = Hydraulic Diameter

Therefore for small Re ($<$ 100), Nu = 3.5 Comparing this with theoretical values (Fig.) we can see that our assumption is reasonable i.e. for laminar flow and for Prandtl No. of 0.7 - 0.74, a Nu value of 3.5 is found (Typical of dia-atomic ideal gases).

However, the MASH equstion differes considerably from the ESDU graph (Fig. E2) for Re = 500 - 2000, as illustrated in Fig. E3. Nu increases proportional to Re where as the ESDU graph shows an increase in Re value has little effect on Nu. This may be due to the fact that for a small hydraulic diameter hole and L/D value a turbulent flow condition exists at entry exit.

Nuselt's Equation $Nu = 0.023\ Re^{0.8}\ Pr^{0.33}$

For $Pr = 0.74$, $Nu = 0.018\ Re^{0.8}$

This is then multiplied by $\left[1 + \dfrac{2}{(L/D)^{3/2}} \right]$

which shows that Nu decreases and L/D increases to take account of entry effects.

The results thus produced correspond well with the results of ESDU theoretical values.

Transition flow boundary is always considered to be most uncertain area.

In MASH the equation used is

$$Nu = \left[3{,}5\ X + 0.018\ Re^{0.8}\ Y \right] * \left[(1+\alpha)\ X\ 1 + \frac{2}{(L/D)^{3/2}} \right]$$

where $X = 3 - \dfrac{Re}{1000}$ & $Y = \dfrac{Re}{1000} = 2$

From the above it can be seen that

(a) for Re closer to 2000 than 3000, $X > Y$ and therefore the dominant term is $3.5\ (1 + \alpha)$ i.e. laminar flow equation.

(b) for Re closer to 3000 than 2000, $Y > X$ and the dominant term is $0.018 \ Re^{0.8} \left[1 + \frac{2}{(L/D)^{3/2}} \right]$ i.e. trubulent flow conditions.

Therefore, it is accepted that the calculated values in the transitional flow could be prone to inaccuracies; but overall the method does offer the best results.

Friction Co-efficient:

For the three boundary conditions, the following friction coefficients have been assumed.

Laminar Flow:	2.68
Transition Flow:	$2 + 0.68 * (3 - \frac{Re}{1000})$
Turbulent Flow:	2

Heat Transfer Coefficients:

$$\text{Nusselt Number } (Nu) = \frac{hD}{k}$$

$$\text{Heat transfer coeff. } (h) = \frac{Nu * k}{D}$$

For the pilot rig, values of $h = 37 \ W \ m^{-2} \ {}^oK^{-1}$ and Nu 110 have been used.

Friction Factor (ff)

$$\text{Friction Factor} = \text{Friction Coeff.} * \frac{h}{Cp \nu \rho} \quad \text{(Dimensionless)}$$

$$= \frac{\text{Friction Coeff.} * Nu * k}{Cp * \nu\rho * D}$$

$$\text{where } h = \frac{Nu \ k}{D}$$

$$= \frac{\text{Friction Coeff.} * Nu * k}{Cp * Re * \mu}$$

$$\text{where } Re = \frac{D}{u}$$

$$= \frac{\text{Friction Coeff.} * Nu}{Pr * Re}$$

$$\text{Friction Factor (ff)} = 1.35 * \text{Friction Coeff.} * \frac{Nu}{Re}$$

For laminar flows this approximates to between 12/Re and 20/Re for the pilot plant which is in close approximation to the theoretical predicted values (GEC-Heat Transfer and Fluid Flow) of 16/Re. Fig. E4 surface roughness (ϵ) is considered to be negligible for the laminar flow conditions.

It is generally recognised that the values of friction factor for transition phase lie between the values of friction factor for laminar and turbulent flows. MASH calculations generally show an error of 3% from the published data (GEC-Heat Transfer & Fluid Flow).

Calculating the friction factor in the turbulent flow region for the pilot rig with a surface roughness (ϵ) of 2.4×10^{-4} (for smooth concrete pipes, cast metals, etc.), D of 0.00635m and thus of relative roughness (ϵ/D) of 0.038 the program predicts a value of 0.01 for Re = 4000. This, when compared with Rose & Cooper's value of 0.016 shows an error of 50%. However, it is accepted that the accuracies of friction factors is generally questionable.

For turbulent flows, a more accurate evaluation is obtained by using values of relative roughness ϵ/d for different materials in the Colebrook equation.

$$\frac{1}{\sqrt{C_f}} = 2.28 - 4 \, LN \left[\epsilon + \frac{4.68}{Re\sqrt{\epsilon f}} \right]$$

Overall the results thus obtained show good correlation with the theoretical results (Fig.)

Radiative Heat Transfer

The program assumes a radiative heat transfer on gas-side only; air side temperature being too low.

PCO_2 = LN (Gas Pressure * VCO_2 * HYDIA)

which is of the form LN (Partial Pressure * Mean beam length)

where mean beam length is the HYDIA.

Using the empirical equation

ϵ Tg can be given as

$$\epsilon_{H2O}\, Tg = \epsilon\, H2O = e^{(-1.9 + PH2O \times 0.727)}$$

$$\epsilon\, CO2\, Tg = \epsilon CO_2 = e^{(-2.2 + PCO2\,(1.25-0.54\, PCO2))}$$

$$\epsilon_i\, Tg = \epsilon CO_2 + \epsilon H_2O - DE$$

where Tg is the gas temperature

Radiative Heat Coefficient Ht (RAD HT COEFF)

$$= \epsilon \frac{(Tg^4 - Tw^4)}{Tg - Tw}$$

From the Program RHTC or Hr $= \frac{3}{4} \epsilon\, Tg\, (Tg^2 + Tg\, Tw + Tw^2)\sigma$

$$= \frac{3}{4} \epsilon \frac{(Tg^3 - Tw^3)}{Tg - Tw} Tg\sigma$$

$$= \frac{3}{4} \epsilon \frac{Tg^4 - Tg\, Tw^3\sigma}{Tg - Tw}$$

The results when compared with Hottel & Sarofim (Fig. E5) give quite a good correlation.

7. Temperature Calculation:

The temperature changes are calculated by the formula

$$\text{Gas Temperature} = \frac{L * (\text{Matrix Temp} - \text{Gas Temp})}{1 + \frac{L + K}{2}}$$

& Matrix Temperature = - Gas Temp * $\frac{K}{L}$

where k = $\frac{\text{4 * Gas Temp * (1- Blockage)* Hausen HT Coeff.}}{\text{(Blockage * HYDIA) * Rhowall * Wall SPHT}}$

= $\frac{\text{S * Hausen HT Coeff.}}{\text{Rhowall * Wall SPHT}}$

L = $\frac{\text{4 * Axial Step * Hausen HT Coeff.}}{\text{HYDIA * Gas Flux Density * CP FLU}}$

where $\frac{\text{4 * Axial Step}}{\text{HYDIA}}$ = $\frac{\text{Surface Area of Cyl. Sec}}{\text{X - sectional Area}}$

= $\frac{\text{Surface Area * Hausen HT Coeff.}}{\text{X-sectional Area * Gas Flux Density * CP FLU}}$

= $\frac{\text{T * Hausen HT Coeff.}}{\text{Gas Flux Density (I) * CP FLU}}$

S = $\frac{\text{4 * Gas Temp. Step * (1-Blockage)}}{\text{Bloakcage * HYDIA}}$

T = 4 * $\frac{\text{Axial Step}}{\text{HYDIA}}$

& where Bloakcage = 1 - $\frac{1}{\left(1+ \frac{\text{wall thickness}}{\text{hydraulic dia.}}\right)^2}$ =

1 - $\frac{\text{(HYDIA)}^2}{\text{(HYDIA+Wall thickness)}^2}$

8. Gas Pressure Equalisation Iteration:

Pressure change =
$\frac{\text{Gas Flux Density)}^2 \text{ * Friction Multiplier*Friction Factor}}{\text{RHO FLU}}$

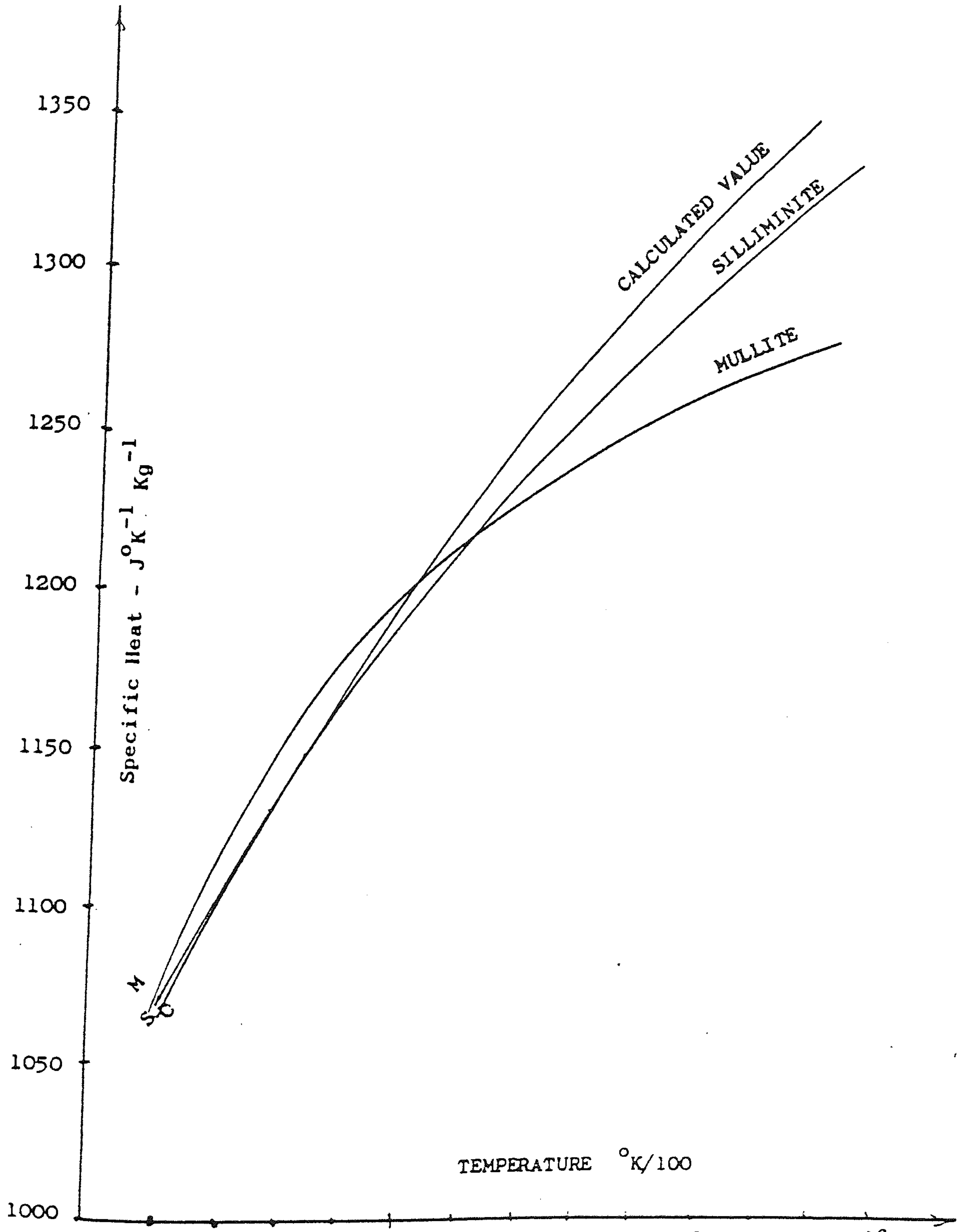
where Friction Multiplier = 2 * L/D

$$\text{Pressure charge} = \frac{(\rho v)^2 * 2 * \frac{L}{D} * \text{f.f.}}{\rho}$$

$$= \rho v^2 * 2 * \frac{L}{D} * \text{f.f.}$$

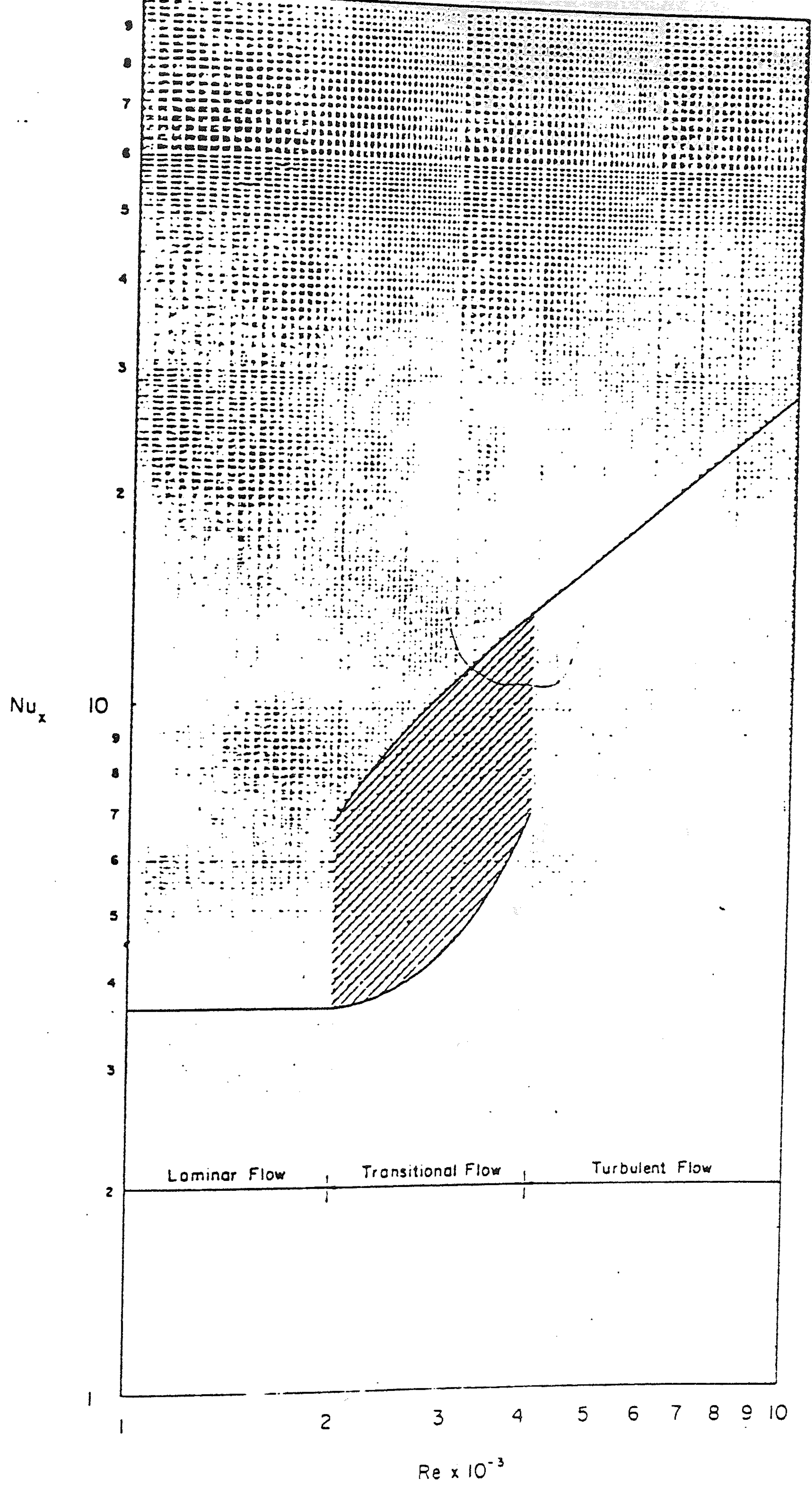
$$= \frac{L}{D} \rho \frac{v^2}{2} \quad (\text{for f.f.} = 4 - \text{D.S. Miller})$$

looks to be reasonably accurate.



TEMPERATURE/SPECIFIC HEAT RELATIONSHIP

FIGURE E.1



ESDU -NUSSELT NUMBER FOR TRANSITIONAL GAS FLOW (PR 0.7)

FIGURE E.2

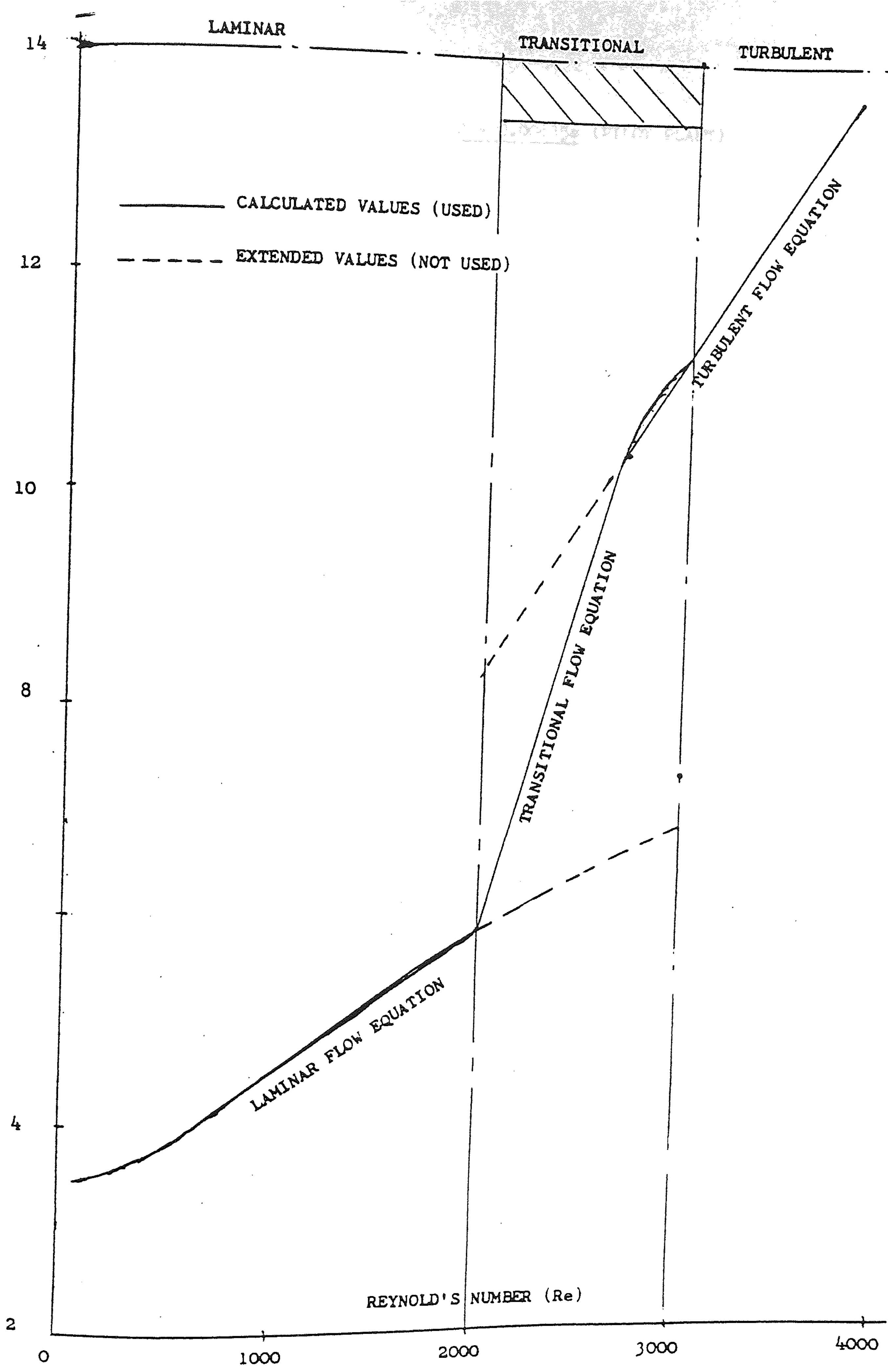
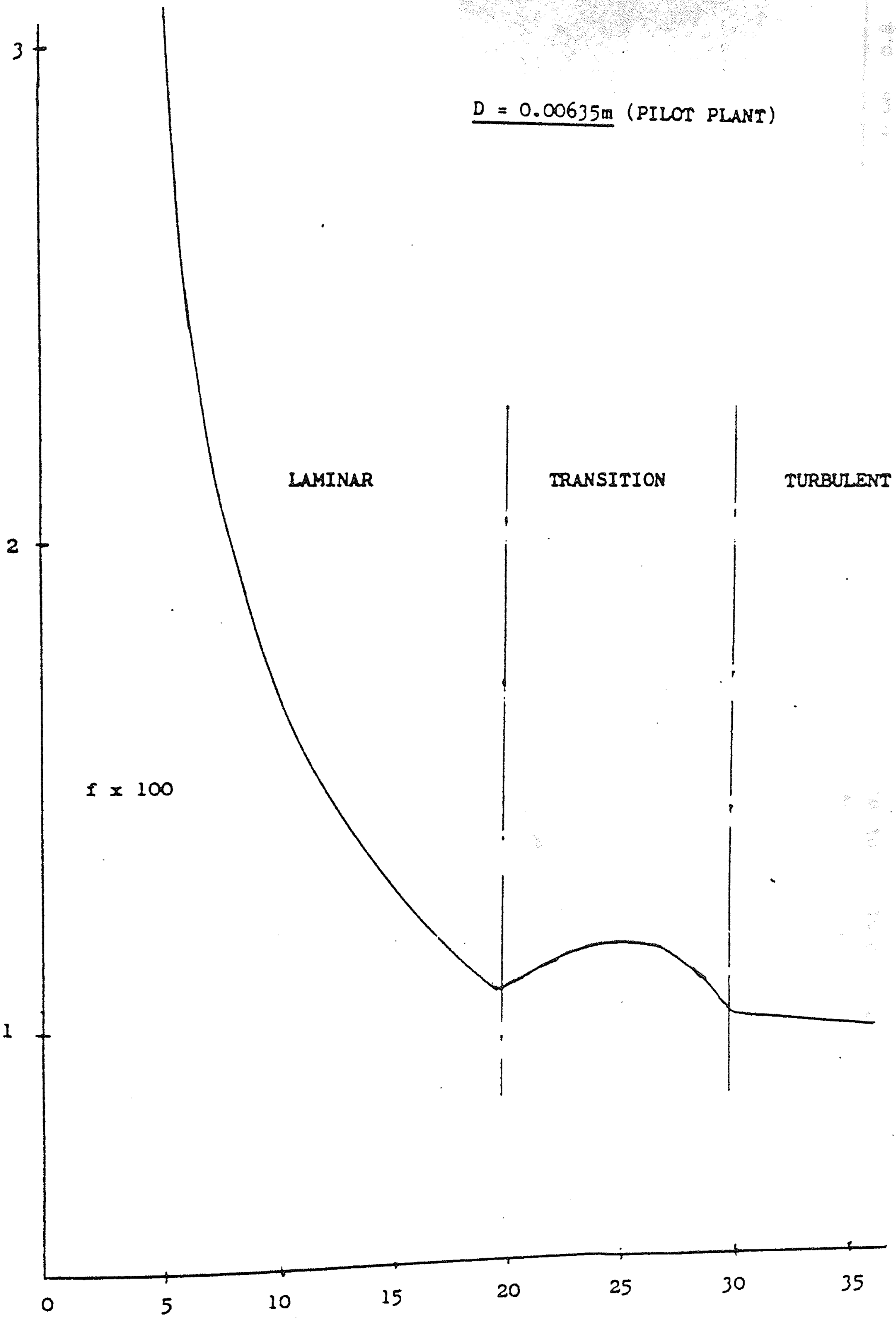


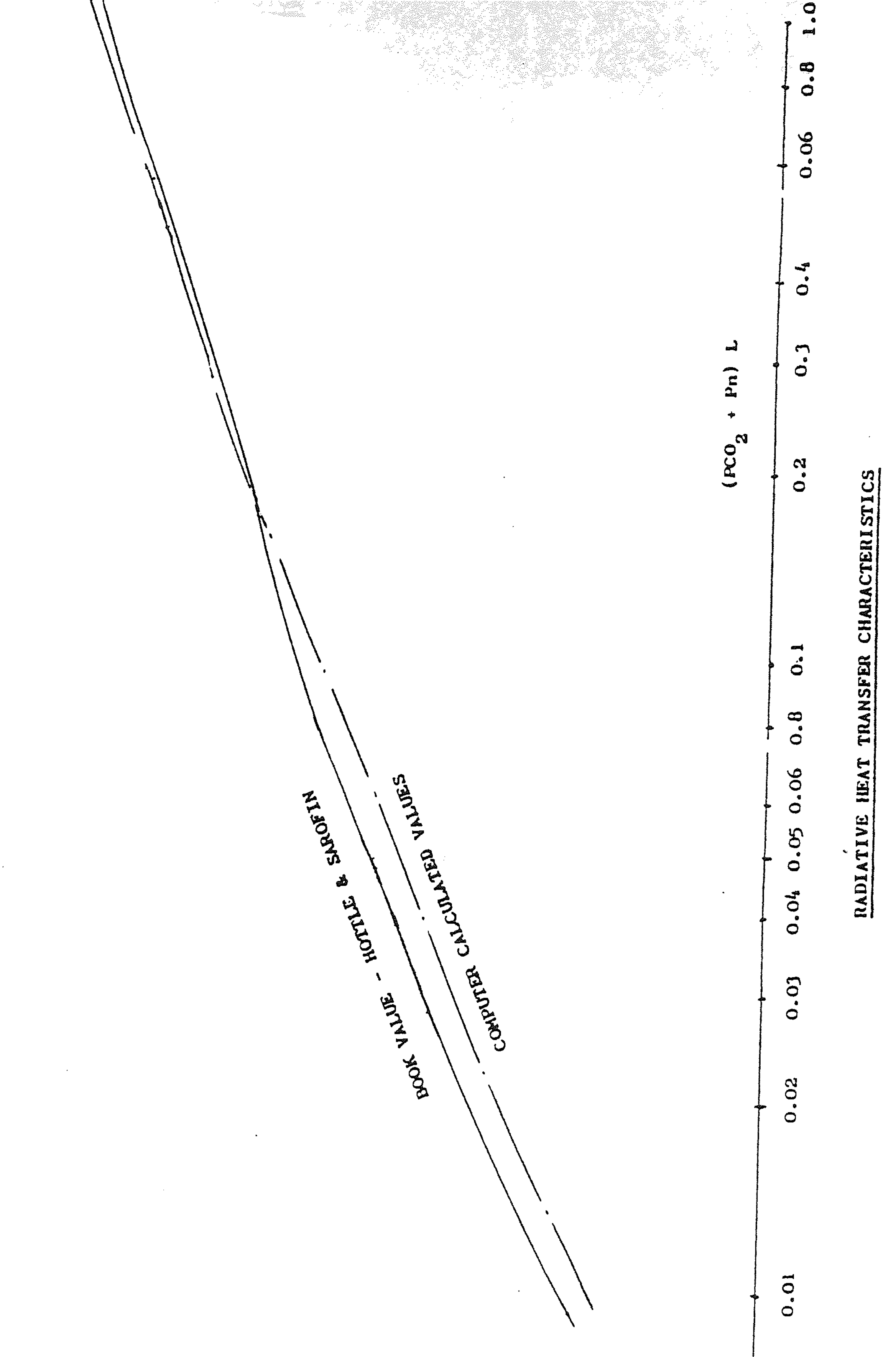
FIGURE E.3

GEC FLOW DIAGRAM



MOODY DIAGRAM (FRICTION FACTOR/REYNOLD NUMBER)

FIGURE E.4



RADIATION HEAT TRANSFER CHARACTERISTICS

FIGURE E.5

STIR DATA

```
      0.50000
      0.75000
      0.75000
      0.00040
      0.00040
      0.00060
      0.00060
      0.00000
      0.00000
     80.00000
      2.00000
      1.00000
      0.01200
      0.00700
   1200.00000
     30.00000
 104325.00000
 101300.00000
      0.05883
      0.73849
      0.14254
      0.06013
      0.00000
   2400.00000
      1.60000
```

1.0	11.0	2.0	30.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	8.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0
1.0	7.0	1.0	7.0

15 ITERATIONS BEFORE MASH WAS POURED

MATRIX TEMPS(DEG.C)   GAS CYCLE

GAS PERIOD =   1.00000   (MINS.)
NO OF TIMESTEPS =   7

AXIAL STATION	TIMESTEP 1	TIMESTEP 2	TIMESTEP 3	TIMESTEP 4	TIMESTEP 5	TIMESTEP 6	TIMESTEP
0.0400	1029.489	1033.105	1036.643	1040.099	1043.478	1046.782	1050.0
0.1200	939.257	942.422	945.558	948.660	951.732	954.773	957.7
0.2000	876.752	879.772	882.774	885.755	888.716	891.659	894.5
0.2800	821.399	824.333	827.256	830.163	833.057	835.938	838.8
0.3600	769.177	772.043	774.904	777.750	780.588	783.417	786.2
0.4400	718.859	721.667	724.472	727.265	730.051	732.830	735.5
0.5200	669.763	672.521	675.277	678.022	680.762	683.497	686.22
0.6000	621.053	623.775	626.494	629.205	631.911	634.613	637.30
0.6800	571.082	573.797	576.511	579.216	581.917	584.614	587.30
0.7600	516.032	518.813	521.592	524.360	527.124	529.884	532.63
0.8400	446.657	449.678	452.695	455.698	458.694	461.683	464.65
0.9200	336.201	339.967	343.719	347.447	351.158	354.852	358.52

610

MATRIX TEMPS   AIR CYCLE

AIR PERIOD =   1.00000   (MINS.)
NO OF TIMESTEPS =   7

AXIAL STATION	TIMESTEP 1	TIMESTEP 2	TIMESTEP 3	TIMESTEP 4	TIMESTEP 5	TIJESTEP 6	TIMESTEP
0.0400	1046.556	1043.112	1039.658	1036.197	1032.734	1029.263	1025.7
0.1200	954.688	951.603	948.506	945.402	942.294	939.177	936.0
0.2000	891.606	888.640	885.664	882.681	879.696	876.704	873.6
0.2800	835.897	832.999	830.095	827.186	824.277	821.362	818.4
0.3600	783.380	780.538	777.691	774.842	771.994	769.143	766.2
0.4400	732.795	730.002	727.207	724.411	721.619	718.825	716.0
0.5200	683.462	680.712	677.962	675.214	672.471	669.728	666.9
0.6000	634.576	631.858	629.141	626.428	623.721	621.016	618.3
0.6800	584.574	581.859	579.147	576.440	573.739	571.041	568.3
0.7600	529.838	527.057	524.280	521.509	518.745	515.986	513.1
0.8400	461.617	458.595	455.580	452.574	449.579	446.591	443.5
0.9200	354.701	350.918	347.160	343.429	339.728	336.054	332.3

```
GAS TEMPERATURE(DEG.C) =      1199.900
AIR TEMPERATURE(DEG.C) =        30.000
WASTE GAS TEMP(DEG.C) =       588.199
AIR PREHEAT TEMP(DEG.C) =       749.972
GAS FLUX DENSITY(KG/S.M2) =      0.75040
AIR FLUX DENSITY(KG/S.M2) =      0.74900
AIR SIDE EFFECTIVENESS =      0.59156
GAS SIDE EFFECTIVENESS =      0.55206
AIR FLUX RATIO =    0.999
GAS FLUX RATIO =    1.001
PASSAGE LENGTH(M) =      0.96000
GAS 'CARRY-OVER'(KG/S.M2) =      0.00197
AIR 'CARRY-OVER'(KG/S.M2) =      0.00375
SOLID/AIR 'CAPACITY RATE' RATIO =    25.98112
SOLID/GAS 'CAPACITY RATE' RATIO =    25.95688
PASSAGE LENGTH/HYDRAULIC DIA =        80.0
REFRACTORY UTILISATION(W/KG) =      207.994

GAS PRESSURE(N/M2) =      101300.000
WASTE GAS PRESSURE(N/M2) =      101243.219
GAS PRESSURE DROP(N/M2) =       56.78

AIR PRESSURE(N/M2) =      104325.000
PREHEAT AIR PRESSURE(N/M2) =      104303.582
AIR PRESSURE DROP(N/M2) =       21.42

STIR DATA
```

APPENDIX F

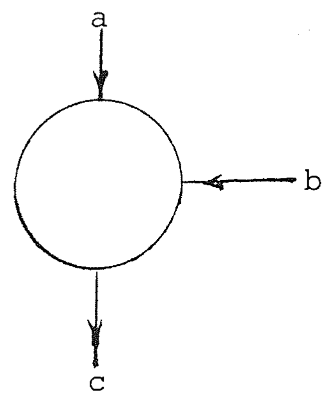
LEAKAGE TESTING USING ARGON AS TRACER GAS

Argon is fed in sufficiently high quantities to known gas flowrates to give a tracer gas concentration of 2.5% to 10%. The mixed gas having passed through mixing sections, ensuring complete mixing, (Fig. F1) is analysed using mass spectometer. The results are fed to the computer model JIMLAD.BAS developed from the calculations given below and interstream leakage are predicted.

The results obtained show a hogh correlation between the measured and estimated flows for cold air condition and satisfactory for the hot gases.

CALCULATION OF DUCT FLOWS:

The introduction of the tracer gas can be represented by:



a is the gases in the duct before introducing the tracer gas.

b is the tracer gas added to the duct flow.

c is the duct gases and tracer gas mixture in the duct after introducing the gas.

$\dot{m}_a$, $\dot{m}_b$, $\dot{m}_c$ are the mass flowrates at each point in kg/sec

TG_a, TG_b, TG_c are the mass concentrations of the tracer at each point.

We can say that,

$$\dot{m}_a + \dot{m}_b = \dot{m}_c$$

and $$\dot{m}_a TG_a + \dot{m}_B\ TG_b = \dot{m}_c TG_c$$

Now $TG_b = 1$ since the tracer gas is pure

$\dot{m}_b$, TG_a and TG_c are known

So,

$$\dot{m}_a TG_a + \dot{m}_b = (\dot{m}_a + \dot{m}_b)\ TG_C$$

$$\dot{m}_a\ (TG_c - TG_a) = \dot{m}_b\ (1 - TG_c)$$

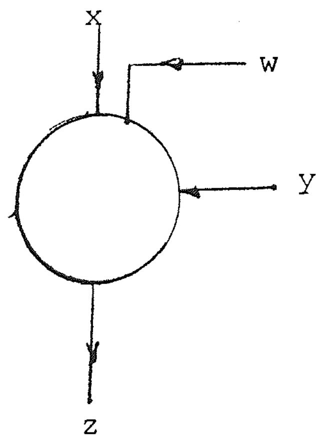
$$\dot{m}_a = \dot{m}_b\ \frac{(1 - TG_c)}{(TG_c - TG_a)}$$

therefore $\dot{m}_a$ can be calculated, and hence $\dot{m}_c$ using

$$\dot{m}_a + \dot{m}_b = \dot{m}_c$$

CALCULATION OF INTERSTREAM LEAKAGE FLOWRATE

The leakage of gases from the Main Side to the Waste Gas Side can be represented thus (making allowance for leakage into the system from the Seal Air);



x is the gases from the Waste Gas Inlet entering the Wast Gas Outlet Duct

w is the Seal Air leakage

y is the leakage from the Main Air Side to the Waste Gas Side

z is the gas mixture in the Waste Gas Outlet Duct.

$\dot{m}_x$, $\dot{m}_w$, $\dot{m}_y$, $\dot{m}_z$ are the mass flowrates at each point in kg/sec

TH_{x1}, TH_{w1}, TG_{y1}, TG_{z1} are the mass concentration of tracer gases at each point before introduction of tracer gas to the Main Air Inlet.

TG_{z2}, TG_{w2}, TG_{y2}, TG_{z2} are the mass concentration of tracer ga gas at each point with introduction of tracer gas to the Main Air Inlet

We can say that,

$$\dot{m}_x TH_{x1} + \dot{m}_w TG_{w1} + \dot{m}_y TG_{y1} = \dot{m}_z TG_{z1}$$

and $\dot{m}_x TG_{x2} + \dot{m}_w TG_{w2} + \dot{m}_y TG_{y2} = \dot{m}_z TG_{z2}$

now $TG_{x1} = TG_{x2}$, $TG_{w1} = TG_{w2}$ hence

$$\dot{m}_y (TG_{y2} - TG_{y1}) = \dot{m}_z (TG_{z2} - TG_{z1})$$

hence $$\dot{m}_y = \dot{m}_z \frac{(TG_{z2} - TG_{z1})}{(TG_{y2} - TG_{y1})}$$

now $\dot{m}_z$, TG_z, TG_{z2} TG_{y1}, TG_{y2} are known

hence $\dot{m}_y$ can be calculated

Input Name of File
to be used for Output

Input Readings of Argon
Introduced as Tracer Gas

Input Gas Analyses in Ducts before
and after Argon tracing (in % vol)

Calculate flowrates of
Argon into Ducts

Convert Volume analysis
to Mass analysis

Calculate all four Duct Flowrates from
simultaneous equations for Argon and Mass balances

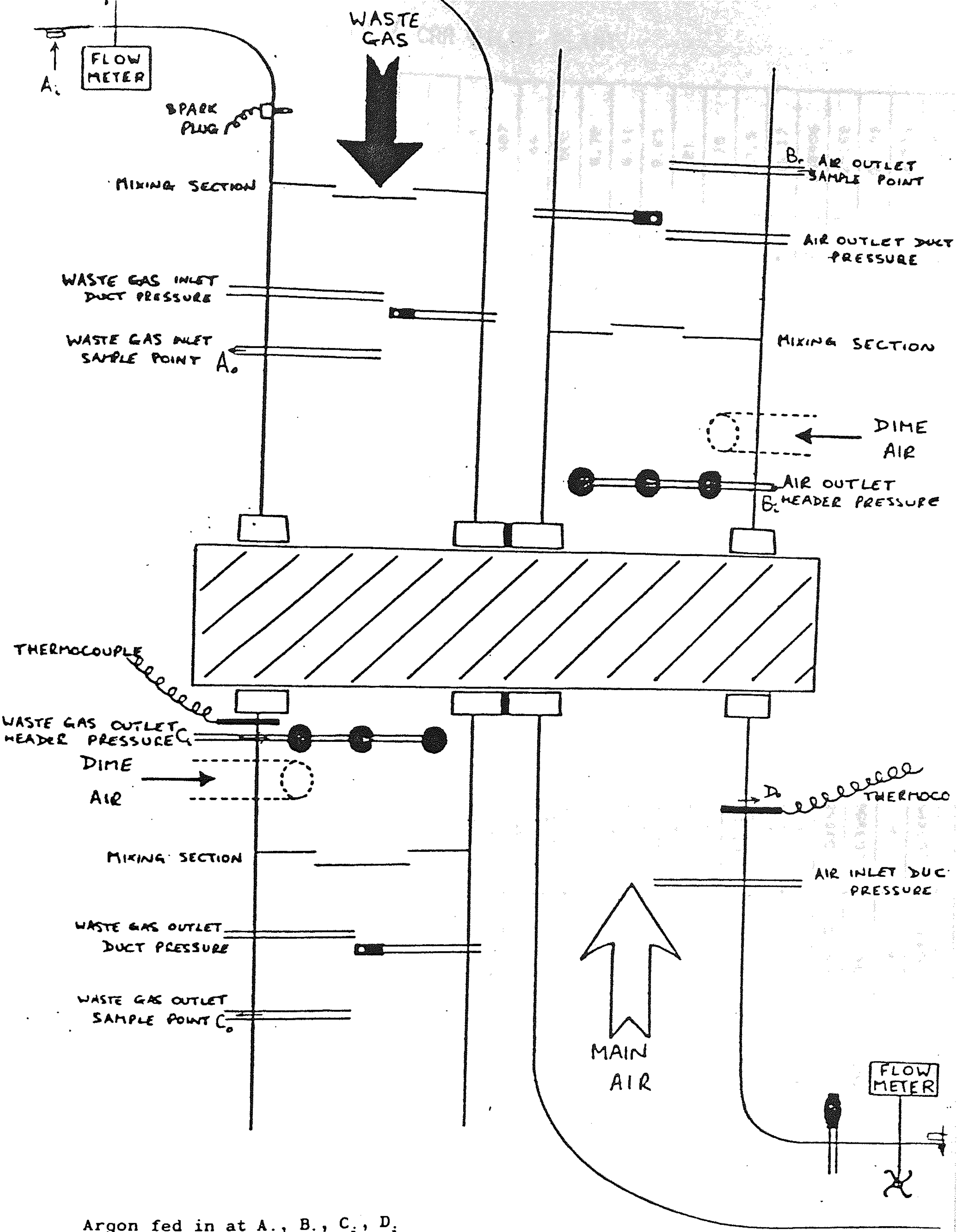
Give alphabetical names to each flow
in system

Calculate Interstream Leakage Flowrate from
simultaneous equations for Argon balances

Calculate Interstream Leakage Flowrate as
a Percentage of Main Air Inlet Flowrate

Calculate Seal Air Leakage from
simultaneous equations for Oxygen and Mass balances

Output Results in Pictorial Fashion



Argon fed in at A_i, B_i, C_i, D_i
Samples taken at A_o, B_o, C_o, D_o

LEAKAGE TESTING USING ARGON AS TRACER GAS

FIGURE F.1

Group	Parameter	RS500 Data 3881.1	RS500 Data 3881.2	RS500 Data 3881.3	RS500 Data 3881.4	RS850 Data 7981.1	RS850 Data 7981.2	RS850 Data 7981.3	RS850 Data 8981.1	RS850 Data 8981.2
	TEST NO.	3881.1	3881.2	3881.3	3881.4	7981.1	7981.2	7981.3	8981.1	8981.2
	ROTOR SPEED RPM	.5	1	.275	.075	.5	1	.125	.25	.5
WASTE GAS SIDE	COMBUSTION AIR FLOWRATE kg/sec	.045	.045	.045	.045	.0395	.0395	.0395	.0395	.0395
WASTE GAS SIDE	PRIMARY AIR FLOWRATE ft3/min	*	*	*	*	24.8	24.8	24.8	24	24
WASTE GAS SIDE	NATURAL GAS FLOWRATE kg/sec	.0007	.0007	.00067	.00068	.00109	.00108	.00109	.00102	.00105
WASTE GAS SIDE	GAS VALVE SETTING No. of turns	*	*	*	*	1.734	1.734	1.734	*	*
WASTE GAS SIDE	GAS FLUX DENSITY kg/sec m2	.568	.568	.567	.567	.504	.504	.504	.503	.504
WASTE GAS SIDE – TEMP.	WASTE GAS INLET C	575	585	585	*	885	885	884	865	870
WASTE GAS SIDE – TEMP.	WASTE GAS OUTLET HEADER C	*	*	*	*	365	360	375	360	360
WASTE GAS SIDE – TEMP.	WASTE GAS OUTLET C	275	290	290	300	320	320	335	335	320
WASTE GAS SIDE – PRESSURE	WASTE GAS INLET mb	*	*	*	*	.81	.75	.75	.78	.75
WASTE GAS SIDE – PRESSURE	WASTE GAS OUTLET HEADER mb	*	*	*	*	.31	.31	.31	.28	.28
WASTE GAS SIDE – PRESSURE	WASTE GAS OUTLET mb	*	*	*	*	.09	.08	.08	.03	.03
	INTERSTREAM PRESSURE DIFF. mb	*	*	*	*	5.41	5.47	5.47	6.07	6.1
MAIN AIR SIDE	MAIN AIR INLET FLOWRATE kg/sec	.043	.043	.043	.043	.04	.04	.04	.04	.04
MAIN AIR SIDE	AIR FLUX DENSITY kg/sec m2	.534	.534	.534	.534	.497	.497	.497	.497	.497
MAIN AIR SIDE – TEMP.	MAIN AIR INLET C	41	41	41	41	44.5	46.5	46.5	42.3	44
MAIN AIR SIDE – TEMP.	MAIN AIR OUTLET C	355	370	370	295	500	510	500	490	500
MAIN AIR SIDE – PRESSURE	MAIN AIR INLET mb	*	*	*	*	6.22	6.22	6.22	6.79	6.72
MAIN AIR SIDE – PRESSURE	MAIN AIR OUTLET HEADER mb	*	*	*	*	6.22	6.22	6.22	6.85	6.41
MAIN AIR SIDE – PRESSURE	MAIN AIR OUTLET mb	*	*	*	*	6.22	6.22	6.22	6.85	6.85
	AMBIENT TEMPERATURE C	*	*	*	*	16	16	16	21	21
	TORQUE AMMETER READING Amps	*	*	*	*	.65	.66	.63	.8	.75
	INTERSTREAM LEAKAGE %	9.78	9.57	12.61	13.15	15.34	15.47	14.61	17.28	17.9
	THERMAL EFFICIENCY %	*	*	*	*	26.82	32.03	21.41	34.38	36.17
TRACER GAS METHOD	WASTE GAS INLET FLOWRATE kg/sec	*	*	*	*	.03979	.0371	.04387	.03392	.03496
TRACER GAS METHOD	WASTE GAS OUTLET FLOWRATE kg/sec	.04158	.03926	.03825	.03865	.04534	.04479	.04528	.04898	.05165
TRACER GAS METHOD	MAIN AIR INLET FLOWRATE kg/sec	.03983	.04054	.04103	.03996	.0384	.04165	.03962	.03953	.03793
TRACER GAS METHOD	MAIN AIR OUTLET FLOWRATE kg/sec	*	*	*	*	.02223	.02422	.01957	.02403	.02574
TRACER GAS METHOD	INTERSTREAM LEAKAGE FLOW kg/sec	.00575	.0056	.0043	.00429	.00648	.00704	.00638	.0075	.0075

RS1050 Data

			TEST NO.	231081.1	231081.2	281081.3	281081.4	281081.5
			ROTOR SPEED RPM	.5	1	.25	.25	.148
WASTE GAS SIDE			COMBUSTION AIR FLOWRATE kg/sec	.066	.066	.0715	.0705	.0705
			PRIMARY AIR FLOWRATE ft3/min	27	27	32	30	30
			NATURAL GAS FLOWRATE kg/sec	.00193	.00193	.00209	.00204	.00208
			GAS VALVE SETTING No. of turns	2	2	2.016	2.016	2.016
			GAS FLUX DENSITY kg/sec m2	.844	.844	.914	.901	.902
	TEMP.		WASTE GAS INLET C	1050	1055	1057	1060	1061
			WASTE GAS OUTLET HEADER C	570	600	615	620	580
			WASTE GAS OUTLET C	560	550	550	565	574
	PRESSURE		WASTE GAS INLET mb	5.4	5.4	5.2	5.3	5
			WASTE GAS OUTLET HEADER mb	3.4	3.2	3.1	3.3	302
			WASTE GAS OUTLET mb	1.4	1.4	1.3	1.4	1.4
			INTERSTREAM PRESSURE DIFF. mb	6.8	6.6	3.3	6.4	7.2
MAIN AIR SIDE			MAIN AIR INLET FLOWRATE kg/sec	.073	.073	.073	.071	.071
			AIR FLUX DENSITY kg/sec m2	.907	.807	.907	.882	.882
	TEMP.		MAIN AIR INLET C	31	31	35	38	39
			MAIN AIR OUTLET C	600	630	615	610	602
	PRESSURE		MAIN AIR INLET mb	12.2	12	8.8	11.8	11.2
			MAIN AIR OUTLET HEADER mb	11.8	11.4	8.7	10.9	10.8
			MAIN AIR OUTLET mb	12.2	12	8.7	11.7	11.3
			AMBIENT TEMPERATURE C	11	12	16	17	17
			TORQUE AMMETER READING Amps	.85	.8	.9	.85	.8
			INTERSTREAM LEAKAGE %	20.74	22.78	19.23	20.53	20.6
			THERMAL EFFICIENCY %	13.54	13.45	17.98	9.93	11.45
TRACER GAS METHOD			WASTE GAS INLET FLOWRATE kg/sec	.06555	.06622	.0656	.06712	.06825
			WASTE GAS OUTLET FLOWRATE kg/sec	.08195	.08198	.08236	.09007	.0847
			MAIN AIR INLET FLOWRATE kg/sec	.06727	.06926	.06323	.06477	.06524
			MAIN AIR OUTLET FLOWRATE kg/sec	.01813	.01742	.02369	.01357	.01604
			INTERSTREAM LEAKAGE FLOW kg/sec	.01501	.0169	.01317	.01428	.01443

RS1100 Data

9282.1	9282.2	9282.3
1	.5	.25
.046	.048	.046
17	17	17
.00146	.00148	.00146
1.875	1.875	1.875
.5897	.5896	.5896
1105	1110	1110
570	580	575
520	520	520
1.494	1.494	1.494
.747	.747	.747
.249	.249	.249
3.856	3.856	3.856
.042	.042	.042
.5217	.5217	.5217
36	38	39
605	600	580
5.35	5.35	5.35
5.35	5.35	5.35
5.35	5.35	5.35
15	14	14
1.1	.8	.7
23.81	22.02	23.06
8.69	8.83	7.89
.04229	.04293	.04225
.06557	.06272	.05823
.03799	.03918	.03948
.00797	.00835	.00762
.01022	.00998	.01046

P500 Data

		TEST NO.	6881.1	6881.2	6881.3	6881.4	6881.5	6881.5	7881.6	7881.7	7881.8	7881.9
		ROTOR SPEED RPM	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
WASTE GAS SIDE		COMBUSTION AIR FLOWRATE kg/sec	.04	.04	.039	.039	.04	.04	.04	.04	.04	.039
WASTE GAS SIDE		PRIMARY AIR FLOWRATE ft3/min	23	23	23	23	23	23	23	23	23	23
WASTE GAS SIDE		NATURAL GAS FLOWRATE kg/sec	.00059	.00059	.00058	.00058	.0006	.0006	.0006	.00062	.00061	.00059
WASTE GAS SIDE		GAS VALVE SETTING No. of turns	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
WASTE GAS SIDE		GAS FLUX DENSITY kg/sec m2	.504	.504	.492	.492	.504	.504	.504	.505	.504	.492
WASTE GAS SIDE	TEMP.	WASTE GAS INLET C	560	560	563	560	562	561	560	560	560	562
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET HEADER C	*	295	282	233	220	237	*	*	*	*
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET C	*	270	262	230	215	230	235	225	235	242
WASTE GAS SIDE	PRESSURE	WASTE GAS INLET mb	.93	1.87	3.11	2.92	2.86	2.24	.84	.78	.81	.81
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET HEADER mb	*	1.56	2.68	2.74	2.43	1.99	.56	.5	.5	.56
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET mb	.31	1.25	2.24	2.37	2.05	1.74	.37	.31	.31	.31
		INTERSTREAM PRESSURE DIFF. mb	5.3	4.9	3.7	.88	.16	1.48	3.06	.47	1.59	2.69
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	.043	.042	.042	.042	.043	.043	.043	.043	.043	.042
MAIN AIR SIDE		AIR FLUX DENSITY kg/sec m2	.534	.522	.522	.522	.534	.534	.534	.534	.534	.522
MAIN AIR SIDE	TEMP.	MAIN AIR INLET C	42.5	42	43	44	45	43	40	42.5	42.5	42
MAIN AIR SIDE	TEMP.	MAIN AIR OUTLET C	362.5	365	371	335	325	339	360	320	340	355
MAIN AIR SIDE	PRESSURE	MAIN AIR INLET mb	6.2	6.5	6.6	3.9	2.6	3.6	3.9	1.3	2.2	3.4
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET HEADER mb	6.2	6.8	7.1	4	2.9	3.9	4	1.4	2.5	3.6
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET mb	6.2	6.8	6.8	3.9	2.7	3.7	3.9	1.2	2.4	3.5
		AMBIENT TEMPERATURE C	*	*	20	*	20.1	21	17	18	19	20
		TORQUE AMMETER READING Amps	*	*	.7	.7	.7	.8	*	*	*	*
		INTERSTREAM LEAKAGE %	11.88	6.68	2.52	.83	.41	.55	9.22	.87	4.52	8.53
		THERMAL EFFICIENCY %	*	*	*	*	*	*	*	*	*	*
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	*	*	*	*	*	*	*	*	*	*
TRACER GAS METHOD		WASTE GAS OUTLET FLOWRATE kg/sec	.04679	.03759	.03061	.0233	.02244	.0259	.04343	.03611	.04265	.04365
TRACER GAS METHOD		MAIN AIR INLET FLOWRATE kg/sec	.04003	.0388	.0397	.03886	.03911	.03973	.04138	.04072	.0407	.03983
TRACER GAS METHOD		MAIN AIR OUTLET FLOWRATE kg/sec	*	*	*	*	*	*	*	*	*	*
TRACER GAS METHOD		INTERSTREAM LEAKAGE FLOW kg/sec	.00521	.00285	.0011	.00036	.00018	.00024	.00417	.00039	.00201	.00372

P850 Data

		TEST NO.	8981.1	8981.2	11981	11981.1	11981	14981.1	14981.2	14981.3
		ROTOR SPEED RPM	.5	.5	.5	.5	.5	.5	.5	.5
WASTE GAS SIDE		COMBUSTION AIR FLOWRATE kg/sec	.04	.04	.025	.025	.025	.074	.072	.068
WASTE GAS SIDE		PRIMARY AIR FLOWRATE ft3/min	22	22	22	22	22	58	58	58
WASTE GAS SIDE		NATURAL GAS FLOWRATE kg/sec	.00106	.00105	*	.0007	*	.00169	.00174	.00168
WASTE GAS SIDE		GAS VALVE SETTING No. of turns	1.7422	1.7422	*	*	*	*	*	*
WASTE GAS SIDE		GAS FLUX DENSITY kg/sec m2	.51	.51	*	.319	*	.94	.916	.866
WASTE GAS SIDE	TEMP.	WASTE GAS INLET C	870	865	900	900	902	910	904 4.7	905
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET HEADER C	385	405	430	440	360	510	520	510
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET C	345	365	400	400	325	460	465	480
WASTE GAS SIDE	PRESSURE	WASTE GAS INLET mb	.81	.99	.93	.93	.62	2.37	2.49	2.93
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET HEADER mb	.34	.44	.5	.44	.25	1.43	1.49	.31
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET mb	.062	.093	.44	.12	.06	.59	.59	.72
		INTERSTREAM PRESSURE DIFF. mb	12	17.2	8.8	8.8	.9	3.5	7.5	8.5
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	.04	.04	.04	.04	.04	.078	.078	.076
MAIN AIR SIDE		AIR FLUX DENSITY kg/sec m2	.497	.497	.497	.497	.497	.969	.969	.944
MAIN AIR SIDE	TEMP.	MAIN AIR INLET C	43.5	45	48	48	47	41	40	40
MAIN AIR SIDE	TEMP.	MAIN AIR OUTLET C	530	490	430	425	515	485	515	540
MAIN AIR SIDE	PRESSURE	MAIN AIR INLET mb	12.8	17.7	9.7	9.7	1.5	8.3	8.3	11.8
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET HEADER mb	13.2	17.9	9.7	9.7	1.5	5.7	8.3	13
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET mb	12.8	18.2	9.7	9.7	1.5	5.9	8.1	11.5
		AMBIENT TEMPERATURE C	22	24	23	22	22	21	23	24.5
		TORQUE AMMETER READING Amps	.7	.7	.75	.75	.75	.75	.75	.75
		INTERSTREAM LEAKAGE %	25.73	33.02	*	29.28	*	9.4	12.76	18.93
		THERMAL EFFICIENCY %	19.83	8.19	*	6.68	*	36.38	27.78	23.46
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	.03781	.03692	*	.03469	*	.07527	.08614	.07762
TRACER GAS METHOD		WASTE GAS OUTLET FLOWRATE kg/sec	.05019	.05622	*	.05084	*	.07923	.09468	.08407
TRACER GAS METHOD		MAIN AIR INLET FLOWRATE kg/sec	.03884	.0386	*	.03844	*	.08295	.08025	.07749
TRACER GAS METHOD		MAIN AIR OUTLET FLOWRATE kg/sec	.01428	.00626	*	.00596	*	.05983	.04865	.03521
TRACER GAS METHOD		INTERSTREAM LEAKAGE FLOW kg/sec	.01099	.01402	*	.0124	*	.00823	.01082	.0157

P1050 Data

		TEST NO.	271081.1	271081.2	271081.3	271081.4	271081.5	271081.6
WASTE GAS SIDE		ROTOR SPEED RPM	.5	.5	.5	.5	.5	.5
		COMBUSTION AIR FLOWRATE kg/sec	.07	.07	.07	.07	.07	.071
		PRIMARY AIR FLOWRATE ft3/min	29	29.5	29.5	30	30	30
		NATURAL GAS FLOWRATE kg/sec	.00206	.002	.0021	.0021	.0021	.0021
		GAS VALVE SETTING No. of turns	2	2	2	2	2	2
		GAS FLUX DENSITY kg/sec m2	.895	.895	.895	.895	.896	.908
	TEMP.	WASTE GAS INLET C	1040	1051	1050	1045	1047	1044
		WASTE GAS OUTLET HEADER C	600	615	580	585	555	560
		WASTE GAS OUTLET C	540	570	547	545	535	525
	PRESSURE	WASTE GAS INLET mb	4.1	4.2	3.5	3.2	3.2	2.5
		WASTE GAS OUTLET HEADER mb	2	1.9	1.7	1.7	2.8	2.2
		WASTE GAS OUTLET mb	.87	.93	.75	.75	2	1.56
		INTERSTREAM PRESSURE DIFF. mb	9.7	10.6	7.7	5.9	3.3	.6
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	.072	.07	.07	.07	.071	.072
		AIR FLUX DENSITY kg/sec m2	.894	.87	.87	.87	.882	.894
	TEMP.	MAIN AIR INLET C	35	39	42	42	39	38
		MAIN AIR OUTLET C	589	585	610	600	580	551
	PRESSURE	MAIN AIR INLET mb	13.9	14.8	11.3	9.2	6.7	3.9
		MAIN AIR OUTLET HEADER mb	13.8	13.9	11.3	9.2	6.6	3.5
		MAIN AIR OUTLET mb	13.8	14.8	11.2	9.1	6.5	3.1
		AMBIENT TEMPERATURE C	15	18	19	18	16	15
		TORQUE AMMETER READING Amps	.9	.75	.7	.75	.8	.8
		INTERSTREAM LEAKAGE %	28.27	34.38	20.81	18.6	11.35	*
		THERMAL EFFICIENCY %	11.83	7.32	15.98	20.82	28.44	47.21
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	.06496	.06465	.06604	.06453	.06585	.065
		WASTE GAS OUTLET FLOWRATE kg/sec	.09976	.1097	.08613	.08273	.07229	*
		MAIN AIR INLET FLOWRATE kg/sec	.06437	.06289	.06264	.06713	.06612	.06652
		MAIN AIR OUTLET FLOWRATE kg/sec	.01592	.01002	.02136	.02755	.03993	.06891
		INTERSTREAM LEAKAGE FLOW kg/sec	.01995	.02327	.01415	.01345	.00815	.00872

P1100 Data

15282.1	15282.2	15282.3	15282.4
.5	.5	.5	.5
.0715	.0715	.072	.072
25	*	*	*
.00189	.0019	.00185	.00193
1.96875	1.96875	1.96875	1.96875
.91164	.91177	.91736	.91841
1095	1100	1090	1106
580	610	625	630
555	560	580	592
2.615	3.113	3.548	3.611
1.432	1.805	2.054	2.117
.747	.934	1.121	1.121
.065	3.117	5.79	8.092
.069	.068	.068	.068
.857	.845	.845	.845
36	34	35	35
580	610	625	635
3.24	6.35	9.462	11.703
2.86	6.23	9.338	11.703
2.68	6.23	9.338	11.703
19	16	13	13
.82	.9	.8	.9
1.98	10.22	18.21	24.62
28.6	15.99	10.21	5.1
.07498	.08177	.07743	.07785
.08519	.08699	.08666	.09869
.06525	.0633	.06427	.06349
.04746	.02748	.01598	.00805
.0014	.00706	.01278	.01712

P1150 Data

		TEST NO.	9382.1	9382.2	9382.3	9382.4
		ROTOR SPEED RPM	.5	.5	.5	.5
WASTE GAS SIDE		COMBUSTION AIR FLOWRATE kg/sec	.065	.065	.064	.063
WASTE GAS SIDE		PRIMARY AIR FLOWRATE ft3/min	31	31	31	31
WASTE GAS SIDE		NATURAL GAS FLOWRATE kg/sec	.00193	.00195	.00192	.00191
WASTE GAS SIDE		GAS VALVE SETTING No. of turns	1.9375	1.9375	1.9375	1.9375
WASTE GAS SIDE		GAS FLUX DENSITY kg/sec m2	.83147	.83171	.81891	.80634
WASTE GAS SIDE	TEMP.	WASTE GAS INLET C	1140	1140	1140	1140
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET HEADER C	540	580	600	625
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET C	512	539	557	575
WASTE GAS SIDE	PRESSURE	WASTE GAS INLET mb	2	2.8	2.9	3.2
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET HEADER mb	1.1	1.4	1.7	2.1
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET mb	.56	.75	.87	1.1
		INTERSTREAM PRESSURE DIFF. mb	.1	2.4	6.1	9.5
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	.067	.065	.063	.062
MAIN AIR SIDE		AIR FLUX DENSITY kg/sec m2	.832	.807	.783	.77
MAIN AIR SIDE	TEMP.	MAIN AIR INLET C	35	36	36.5	37
MAIN AIR SIDE	TEMP.	MAIN AIR OUTLET C	600	635	660	525
MAIN AIR SIDE	PRESSURE	MAIN AIR INLET mb	2.6	5.2	9.1	12.7
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET HEADER mb	2.2	5	9	12.7
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET mb	2.1	5	9	12.7
		AMBIENT TEMPERATURE C	14	14	14	14
		TORQUE AMMETER READING Amps	.9	.9	.9	.9
		INTERSTREAM LEAKAGE %	2.57	10.27	20.47	23.42
		THERMAL EFFICIENCY %	32.29	22.97	11.81	1.48
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	.07409	.07302	.07352	.06915
TRACER GAS METHOD		WASTE GAS OUTLET FLOWRATE kg/sec	.06105	.08754	.07469	.07915
TRACER GAS METHOD		MAIN AIR INLET FLOWRATE kg/sec	.06203	.08022	.05691	.05637
TRACER GAS METHOD		MAIN AIR OUTLET FLOWRATE kg/sec	.05385	.03551	.01762	.00268
TRACER GAS METHOD		INTERSTREAM LEAKAGE FLOW kg/sec	.00176	.00683	.01308	.01488

P1200 Data

22382.1	22382.2	22382.3	22382.4	22382
.5	.5	.5	.5	.5
.07	.07	.07	.07	.07
25.5	25.5	25.5	25.5	26
.00238	.00236	.00235	.00234	.002
2.031	2.031	2.031	2.031	2.03
.8989	.8969	.8988	.8987	.898
1220	1220	1210	1210	1200
635	652	670	675	620
588	598	615	628	575
3.05	3.42	3.67	3.86	2.68
1.62	1.99	2.12	2.3	1.49
.68	.93	.99	1.12	.68
1.68	4.3	7.29	8.47	-.07
.0675	.065	.065	.064	.065
.839	.807	.807	.795	.807
41	41	41	40	40
665	687	683	680	657
5.23	7.97	10.83	12.33	3.36
4.86	7.72	10.83	12.33	2.86
4.73	7.72	10.96	12.33	2.61
18	17	17	16	16
.9	.8	.9	.8	.8
9.89	13.75	17.78	27.48	2.87
29.3	18.35	9.21	6.18	37.21
.06376	.06487	.06506	.06387	.06379
.07051	.07493	.07715	.07915	.06546
.0638	.06137	.06059	.05984	.08286
.0412	.02523	.01269	.00841	.05192
.00682	.00925	.01193	.01827	.00198

FD500 Data

Side	Group	Test No.	10881.1	10881.2	11881.3	11881.4
		ROTOR SPEED RPM	.5	.5	.5	.5
WASTE GAS SIDE		COMBUSTION AIR FLOWRATE kg/sec	.088	.088	.042	.025
WASTE GAS SIDE		PRIMARY AIR FLOWRATE ft3/min	56	56	30	30
WASTE GAS SIDE		NATURAL GAS FLOWRATE kg/sec	.00112	.00113	.00068	.00041
WASTE GAS SIDE		GAS VALVE SETTING No. of turns	1.789	1.789	.406	.406
WASTE GAS SIDE		GAS FLUX DENSITY kg/sec m2	1.107	1.107	.53	.316
WASTE GAS SIDE	TEMP.	WASTE GAS INLET C	555	555	585	520
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET HEADER C	405	315	260	225
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET C	340	295	240	190
WASTE GAS SIDE	PRESSURE	WASTE GAS INLET mb	2.6	2.9	3.1	3.6
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET HEADER mb	1.75	1.9	2.75	3.1
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET mb	.75	.8	2.2	2.6
		INTERSTREAM PRESSURE DIFF. mb	.7	2.9	.8	-.2
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	.086	.086	.042	.024
MAIN AIR SIDE		AIR FLUX DENSITY kg/sec m2	1.068	1.068	.522	.298
MAIN AIR SIDE	TEMP.	MAIN AIR INLET C	47	47	44	44
MAIN AIR SIDE	TEMP.	MAIN AIR OUTLET C	310	300	350	*
MAIN AIR SIDE	PRESSURE	MAIN AIR INLET mb	3.6	5.9	3.8	3.25
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET HEADER mb	3.5	6.6	3.9	3.5
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET mb	3.3	5.8	3.9	3.4
		AMBIENT TEMPERATURE C	25	25	25	25
		TORQUE AMMETER READING Amps	*	*	*	*
		INTERSTREAM LEAKAGE %	3.36	9.59	4.78	2.89
		THERMAL EFFICIENCY %	*	*	*	*
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	*	*	*	*
TRACER GAS METHOD		WASTE GAS OUTLET FLOWRATE kg/sec	.092	.10785	.03051	.01415
TRACER GAS METHOD		MAIN AIR INLET FLOWRATE kg/sec	.07871	.07873	.04127	.02418
TRACER GAS METHOD		MAIN AIR OUTLET FLOWRATE kg/sec	*	*	*	*
TRACER GAS METHOD		INTERSTREAM LEAKAGE FLOW kg/sec	.00277	.00792	.00216	.00081

FD1150 Data

Test No.	8382.1	8382.2	8382.3	8382.4	8382.5	10382.1	10382.2	10382.3
ROTOR SPEED RPM	.5	.5	.5	.5	.5	.5	.5	.5
COMBUSTION AIR FLOWRATE kg/sec	.043	.043	.043	.043	.043	.065	.065	.065
PRIMARY AIR FLOWRATE ft3/min	20	20	22	22	22	30	32	32
NATURAL GAS FLOWRATE kg/sec	.00153	.00153	.00152	.00152	.00151	.00199	.002009	.001971
GAS VALVE SETTING No. of turns	1.875	1.875	1.875	1.875	1.875	1.9375	1.9375	1.9375
GAS FLUX DENSITY kg/sec m2	.5532	.5531	.553	.553	.553	.8322	.8324	.8319
TEMP. WASTE GAS INLET C	1145	1145	1145	1145	1140	1120	1120	1110
TEMP. WASTE GAS OUTLET HEADER C	555	435	395	370	348	625	550	505
TEMP. WASTE GAS OUTLET C	475	385	355	335	317	575	503	458
PRESSURE WASTE GAS INLET mb	1.3	1.2	1.1	1.1	1.1	2.8	2.4	2.2
PRESSURE WASTE GAS OUTLET HEADER mb	.53	.53	.5	.5	.5	1.37	1.31	1.31
PRESSURE WASTE GAS OUTLET mb	.12	.12	.12	.12	.19	.65	.68	.68
INTERSTREAM PRESSURE DIFF. mb	3.9	3.2	3.1	2.9	2.5	2.4	2.1	1.5
MAIN AIR INLET FLOWRATE kg/sec	.0425	.07	.089	.1035	.127	.047	.0735	.1035
AIR FLUX DENSITY kg/sec m2	.53	.87	1.11	1.29	1.58	.58	.91	1.29
TEMP. MAIN AIR INLET C	35	34.5	34.5	34.5	34.5	37	37	37
TEMP. MAIN AIR OUTLET C	645	530	430	370	317	715	593	460
PRESSURE MAIN AIR INLET mb	5.1	4.6	4.9	4.9	4.9	5	4.9	4.9
PRESSURE MAIN AIR OUTLET HEADER mb	5.1	4.3	4.5	4.4	4.2	4.9	4.5	4.2
PRESSURE MAIN AIR OUTLET mb	5.2	4.4	4.2	4	3.6	5	4.5	307
AMBIENT TEMPERATURE C	13	14	12	12	11	15	13	13
TORQUE AMMETER READING Amps	.7	.78	.75	.75	.8	.8	.7	.8
INTERSTREAM LEAKAGE %	25.11	15.44	14.44	12.98	9.84	12.91	8.25	9.36
THERMAL EFFICIENCY %	12.54	38.11	39.67	41.27	43.87	18.77	32.69	40.42
WASTE GAS INLET FLOWRATE kg/sec	.03864	.03933	.03973	.04039	.04179	.06029	.05843	.05935
WASTE GAS OUTLET FLOWRATE kg/sec	.0506	.05117	.05322	.05509	.05527	.06553	.06535	.06514
MAIN AIR INLET FLOWRATE kg/sec	.03958	.08717	.0815	.09269	.11496	.04378	.06594	.09381
MAIN AIR OUTLET FLOWRATE kg/sec	.01029	.03953	.05246	.06572	.08576	.02065	.04292	.07073
INTERSTREAM LEAKAGE FLOW kg/sec	.01105	.01121	.01282	.01293	.01199	.00642	.00594	.00944

FD850 DATA

TEST NO.	22981.1	22981.2	22981.3	22981.4	21981.1	16981.1	16981.2	15981.3	18981.1	18981.3	21981.1	21981.2	21981.3
ROTOR SPEED RPM	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
COMBUSTION AIR FLOWRATE kg/sec	.031	.031	.031	.031	.108	.071	.07	.07	.0685	.0685	.0425	.043	.0425
PRIMARY AIR FLOWRATE ft3/min	24	24	24	24	69	58	58	58	62	62	25	25	25
NATURAL GAS FLOWRATE kg/sec	.00106	.00105	.00106	.00105	.00268	.00177	.00172	.00173	.0018	.00179	.00128	.0013	.00127
GAS VALVE SETTING No. of turns	1.625	1.625	1.625	1.625	*	*	*	*	1.875	1.875	1.7656	1.7656	1.7656
GAS FLUX DENSITY kg/sec m2	.398	.398	.398	.398	1.3749	.903	.889	.889	.872	.872	.543	.549	.543
TEMP. WASTE GAS INLET C	905	910	910	910	906	910	910	905	895	901	880	901	905
TEMP. WASTE GAS OUTLET HEADER C	270	230	165	135	590	533	415	450	523	600	525	416	320
TEMP. WASTE GAS OUTLET C	255	215	160	140	544	486	380	420	483	535	470	404	910
PRESSURE WASTE GAS INLET mb	1.1	1.1	1	1	8.1	3	2.7	2.8	3.7	4.5	1.7	1.7	2
PRESSURE WASTE GAS OUTLET HEADER mb	.75	.7	.7	.7	6.5	1.75	1.55	1.75	2.7	3.2	1	1.1	1.2
PRESSURE WASTE GAS OUTLET mb	.5	.5	.5	.5	4.2	.8	.7	.75	1.9	2.2	.5	.7	.6
INTERSTREAM PRESSURE DIFF. mb	4.05	4.1	3.7	3.35	6.5	12.4	10.2	13.4	1.9	8.6	1.2	7.4	12.8
MAIN AIR INLET FLOWRATE kg/sec	.033	.044	.072	.1115	.1125	.076	.146	.108	.0575	.055	.025	.043	.08
AIR FLUX DENSITY kg/sec m2	.413	.55	.9	1.394	1.406	.95	1.825	1.35	.719	.688	.313	.538	1
TEMP. MAIN AIR INLET C	39	43	44.5	44	38	*	*	44	41	41	36.5	44	46.5
TEMP. MAIN AIR OUTLET C	465	455	305	225	452	551	325	450	560	535	500	481	460
PRESSURE MAIN AIR INLET mb	5.2	5.35	5.1	5.2	15.3	15.9	14.3	16.2	5.7	12.95	3.9	9.1	14.9
PRESSURE MAIN AIR OUTLET HEADER mb	5.2	5.35	5	4.85	14.9	15.7	13.7	16.2	5.6	13.1	3.9	9.2	14.9
PRESSURE MAIN AIR OUTLET mb	5.15	5.2	4.7	4.35	14.6	15.4	12.9	16.2	5.6	13.1	3.9	9.1	14.8
AMBIENT TEMPERATURE C	18	19	20	20	20	23	22	23	20	20	18	20	21
TORQUE AMMETER READING Amps	.8	.7	.7	.7	.7	.75	.75	.75	.75	.75	.8	.8	.7
INTERSTREAM LEAKAGE %	24.95	19.21	12.87	7.66	6.73	22.03	11.47	22.93	4.13	18.49	12.41	*	22.1
THERMAL EFFICIENCY %	15.6	35.11	55.19	78.56	25.92	12.89	*	33.34	28.44	5.39	7.1	10.26	23.58
TRACER GAS METHOD WASTE GAS INLET FLOWRATE kg/sec	.02264	.02247	.02142	.02248	.10718	.07632	*	.07685	.06757	.06681	.04034	.04025	.04215
TRACER GAS METHOD WASTE GAS OUTLET FLOWRATE kg/sec	.03168	.03177	.03201	.03234	.12245	.08924	.08767	.089	.06062	.07182	.04444	*	.05781
TRACER GAS METHOD MAIN AIR INLET FLOWRATE kg/sec	.032	.0453	.07224	.11122	.10587	.07369	.1774	.10884	.05608	.05158	.025	.04268	.07904
TRACER GAS METHOD MAIN AIR OUTLET FLOWRATE kg/sec	.00829	.01918	.04593	.09941	.06557	.01847	*	.06114	.03541	.00704	.00592	.00924	.02355
TRACER GAS METHOD INTERSTREAM LEAKAGE FLOW kg/sec	.00908	.00935	.00981	.00882	.00747	.0175	.02102	.02618	.00248	.01067	.00947	0	.01837

FD1050 Data

TEST NO.	91181.1	91181.2	91181.3	101181.4	101181.5	131181.1	131181.2	131181.3	281081.1	281081.2	281081.3	281081.4
ROTOR SPEED RPM	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
COMBUSTION AIR FLOWRATE kg/sec	.042	.042	.042	.041	.042	.102	.1	.1	.07	.07	.07	.07
PRIMARY AIR FLOWRATE ft3/min	14	15	14	15	15	41	41	41	30	30	30	29
NATURAL GAS FLOWRATE kg/sec	.00147	.00149	.00149	.00143	.00147	.00274	.00251	.00257	.00212	.00218	.00214	.00214
GAS VALVE SETTING No. of turns	1.844	1.844	1.844	1.844	1.844	2.125	2.125	2.125	2.018	2.018	2.018	2.018
GAS FLUX DENSITY kg/sec m2	.54	.54	.54	.527	.54	1.301	1.273	1.274	.896	.896	.896	.896
TEMP. WASTE GAS INLET C	1045	1045	1045	1048	1055	1023	1024	1020	1060	1065	1060	1060
TEMP. WASTE GAS OUTLET HEADER C	510	390	370	325	445	635	645	632	620	580	550	540
TEMP. WASTE GAS OUTLET C	450	350	320	298	405	595	600	580	565	520	495	485
PRESSURE WASTE GAS INLET mb	1.56	1.4	1.37	1.68	1.93	5.5	6.7	6.7	4.2	4.1	4.1	3.4
PRESSURE WASTE GAS OUTLET HEADER mb	.5	.5	.5	.56	.75	3.5	4.3	4.2	2.1	2.1	2.1	2
PRESSURE WASTE GAS OUTLET mb	.12	.12	.12	.19	.19	1.81	2.24	2.24	1.06	.87	.93	.87
INTERSTREAM PRESSURE DIFF. mb	7.1	7.8	6.8	7.4	8.2	-1.6	5.6	5.4	9.5	9.5	9.8	10.8
MAIN AIR INLET FLOWRATE kg/sec	.045	.078	.104	.133	.063	.13	.125	.146	.0705	.093	.11	.129
AIR FLUX DENSITY kg/sec m2	.559	.968	1.292	1.652	.783	1.615	1.553	1.814	.876	1.155	1.368	1.602
TEMP. MAIN AIR INLET C	36.5	38.5	38	36	36	38	38	37	36.5	39	39.5	38
TEMP. MAIN AIR OUTLET C	490	465	375	324	515	445	480	426	620	555	495	430
PRESSURE MAIN AIR INLET mb	8.7	9.3	8.7	10.1	11.1	6	13.3	13.7	13.7	14.7	14.8	14.7
PRESSURE MAIN AIR OUTLET HEADER mb	8.7	9.2	8.8	9.8	10.8	5	12.9	12.9	13.6	14.6	14.2	13.8
PRESSURE MAIN AIR OUTLET mb	8.7	9.2	8.2	9.1	10.1	3.9	12.3	12.1	13.7	13.8	13.9	14.2
AMBIENT TEMPERATURE C	14	16	15	12	22	16	16	16	15	17	17	16
TORQUE AMMETER READING Amps	.85	.8	.75	.85	.8	.8	.75	.75	.7	.7	.8	.8
INTERSTREAM LEAKAGE %	41.23	24.06	21.41	19.13	36.04	5.55	13.41	6.94	23.81	22.97	19.65	16.38
THERMAL EFFICIENCY %	5.14	30	60.09	49.13	13.07	31.72	23.66	30.02	13.5	21.23	33.31	36.27
METHOD WASTE GAS INLET FLOWRATE kg/sec	.03682	.03686	.03686	.03876	.03789	.12854	.12694	.11817	.0634	.06507	.06319	.08346
METHOD WASTE GAS OUTLET FLOWRATE kg/sec	.06294	.0623	.07256	.06259	.06973	.11019	.11476	.11602	.0888	.09195	.0949	.09029
METHOD MAIN AIR INLET FLOWRATE kg/sec	.04222	.07078	.09136	.13805	.05467	.14176	.13083	.16142	.06415	.0851	.10168	.13148
METHOD MAIN AIR OUTLET FLOWRATE kg/sec	.0492	.0306	.07816	.07924	.0123	.11287	.07603	.1023	.01717	.03163	.05449	.0888
INTERSTREAM LEAKAGE FLOW kg/sec	.01946	.0183	.02065	.02735	.02146	.00817	.01826	.0116	.01661	.0208	.02102	.02237

<u>*FD1100 Data*</u>

	TEST NO.	1382.1	1382.2	1382.3	1382.4	2382.1	2382.2	2382.3	2382.4	16282.1	16282.2	16282.3	16282.4
	ROTOR SPEED RPM	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5	.5
	COMBUSTION AIR FLOWRATE kg/sec	.072	.072	.072	.073	.098	.098	.0985	.0985	.0425	.0425	.0425	.5
	PRIMARY AIR FLOWRATE ft3/min	24	24	24	24	42	42	42.5	42.5	19	19	.0425	.0425
	NATURAL GAS FLOWRATE kg/sec	.00214	.00216	.00214	.00217	.0028	.0028	.00284	.00287	.001436	.00146	18	18
	GAS VALVE SETTING No. of turns	2	2	2	2	2.125	2.125	2.125	2.125	1.797	1.797	*	.001028
	GAS FLUX DENSITY kg/sec m2	.921	.9213	.921	.9338	1.2522	1.2522	1.2589	1.2593	.5458	.5461	1.797	1.797
TEMP.	WASTE GAS INLET C	1100	1100	1095	1085	1095	1100	1090	1090	1095	1100	*	.5407
TEMP.	WASTE GAS OUTLET HEADER C	624	573	534	495	660	620	603	565	560	455	1100	1105
TEMP.	WASTE GAS OUTLET C	573	525	483	453	620	585	565	530	505	405	400	370
PRESSURE	WASTE GAS INLET mb	3.4	3.1	3.1	3.1	6.2	5.7	5.7	5.5	1.307	1.432	355	330
PRESSURE	WASTE GAS OUTLET HEADER mb	2	1.87	1.81	1.93	4	3.8	3.9	3.6	.498	.623	1.368	1.307
PRESSURE	WASTE GAS OUTLET mb	1	.87	.87	.93	2.2	2.1	2	1.9	.249	.311	.56	.56
	INTERSTREAM PRESSURE DIFF. mb	5.6	5.4	4.9	4.5	6.5	5.9	6.3	4.1	1.806	3.797	.311	.374
	MAIN AIR INLET FLOWRATE kg/sec	.0685	.086	.1035	.133	.105	.133	.15	.2005	.038	.063	3.424	3.051
	AIR FLUX DENSITY kg/sec m2	.8509	1.068	1.286	1.652	1.3	1.65	1.86	2.49	.472	.783	.0875	.1065
TEMP.	MAIN AIR INLET C	39	38	38	38	36	36.5	37	37	32	33	1.087	1.322
TEMP.	MAIN AIR OUTLET C	634	578	483	347	565	480	430	355	630	575	33	32
PRESSURE	MAIN AIR INLET mb	9.1	8.7	8.7	8.6	13.1	12.6	13.2	12.6	3.237	5.354	455	385
PRESSURE	MAIN AIR OUTLET HEADER mb	9	8.5	8.2	8	12.7	12.1	12.5	12	2.988	5.105	5.229	5.105
PRESSURE	MAIN AIR OUTLET mb	9	8.5	8	7.6	12.7	11.6	12	9.6	3.119	5.229	4.856	4.607
	AMBIENT TEMPERATURE C	15	14	14	13	15	15	15	15	14	13	4.793	4.358
	TORQUE AMMETER READING Amps	.8	.8	.8	.8	.65	.65	.7	.7	.85	.75	11	11
	INTERSTREAM LEAKAGE %	15.56	*	8.24	6.25	7.15	5.01	5.08	4.04	17.3	20.43	.7	.7
	THERMAL EFFICIENCY %	9.39	*	27.92	25.8	15.07	22.51	25.33	32.93	11.34	20.91	*	13.86
TRACER GAS METHOD	WASTE GAS INLET FLOWRATE kg/sec	.06704	*	.06669	.06808	.09162	.09079	.09174	.09136	.04247	.04207	*	41.75
TRACER GAS METHOD	WASTE GAS OUTLET FLOWRATE kg/sec	.07604	*	.07921	.08082	.09636	.09876	.10025	.09968	.04581	.05226	*	.04109
TRACER GAS METHOD	MAIN AIR INLET FLOWRATE kg/sec	.06462	*	.1025	.11959	.09852	.12391	.13905	.18883	.03691	.06314	*	.05205
TRACER GAS METHOD	MAIN AIR OUTLET FLOWRATE kg/sec	.01285	*	.05116	.06954	.03168	.05907	.07211	.11617	.00997	.02005	*	.10532
TRACER GAS METHOD	INTERSTREAM LEAKAGE FLOW kg/sec	.01088	*	.00889	.00781	.00757	.00662	.00754	.00803	.00701	.01391	*	.06165
													.01549

			16382.1	16382.2	16382.3	16382.4
		TEST NO.	16382.1	16382.2	16382.3	16382.4
		ROTOR SPEED RPM	.5	.5	.5	.5
WASTE GAS SIDE		COMBUSTION AIR FLOWRATE kg/sec	.043	.043	.043	.043
		PRIMARY AIR FLOWRATE ft3/min	22	22	22	22
		NATURAL GAS FLOWRATE kg/sec	.00153	.00155	.00155	.00154
		GAS VALVE SETTING No. of turns	1.953	1.953	1.953	1.953
		GAS FLUX DENSITY kg/sec m2	.553	.553	.553	.553
	TEMP.	WASTE GAS INLET C	1190	1190	1190	1190
		WASTE GAS OUTLET HEADER C	583	467	410	380
		WASTE GAS OUTLET C	505	455	350	321
	PRESSURE	WASTE GAS INLET mb	1.62	1.37	1.31	1.25
		WASTE GAS OUTLET HEADER mb	.75	.62	.62	.56
		WASTE GAS OUTLET mb	.25	.25	.25	.25
		INTERSTREAM PRESSURE DIFF. mb	3.73	3.11	3.05	2.36
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	.04	.072	.103	.126
		AIR FLUX DENSITY kg/sec m2	.497	.894	1.28	1.565
	TEMP.	MAIN AIR INLET C	33	35	36	36
		MAIN AIR OUTLET C	667	545	415	345
	PRESSURE	MAIN AIR INLET mb	5.48	4.79	4.86	4.86
		MAIN AIR OUTLET HEADER mb	5.35	4.36	3.98	4.11
		MAIN AIR OUTLET mb	5.35	4.48	4.36	3.61
		AMBIENT TEMPERATURE C	12	13	12	12
		TORQUE AMMETER READING Amps	.8	.8	.8	.9
		INTERSTREAM LEAKAGE %	32.09	17.88	12.84	11.43
		THERMAL EFFICIENCY %	12.5	34.55	41.62	44.81
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	.04419	.04444	.04531	.04438
		WASTE GAS OUTLET FLOWRATE kg/sec	.05701	.05807	.05992	.06003
		MAIN AIR INLET FLOWRATE kg/sec	.03453	.06985	.10125	.12088
		MAIN AIR OUTLET FLOWRATE kg/sec	.01178	.04109	.06852	.08909
		INTERSTREAM LEAKAGE FLOW kg/sec	.01326	.01357	.01391	.01464

		TEST NO.	121083.1	181083.1	221183.1	11283.1	11283.2	21283.1	21283.2	21283
WASTE GAS SIDE		ROTOR SPEED RPM	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
		COMBUSTION AIR FLOWRATE kg/sec	0.094	0.101	0.088	0.091	0.089	0.094	0.091	0.09
		PRIMARY AIR FLOWRATE Kg/sec	0.033	0.050	0.036	0.030	0.030	0.033	0.033	0.03
		NATURAL GAS FLOWRATE kg/sec	0.0037	0.0051	0.0020	0.00245	0.0028	0.0031	0.00335	0.00
		GAS VALVE SETTING No. of turns	•	•	•	•	•	•	•	•
		GAS FLUX DENSITY kg/sec m2	1.214	1.318	1.118	1.161	1.140	1.206	1.172	1.1
	TEMP.	WASTE GAS INLET C	1320	1080	970	1010	1110	1190	1270	130
		WASTE GAS OUTLET HEADER C	•	•	•	•	•	•	•	•
		WASTE GAS OUTLET C	780	860	530	540	625	710	740	75
	PRESSURE	WASTE GAS INLET mb	7.9	8.7	3.7	4.4	8.0	7.9	8.9	8.
		WASTE GAS OUTLET HEADER mb	•	•	•	•	•	•	•	•
		WASTE GAS OUTLET mb	6.8	8.9	3.4	3.9	5.5	6.6	5.7	5.
		INTERSTREAM PRESSURE DIFF. mb	3.9	4.1	2.8	4.0	5.5	5.1	4.9	4.
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	0.098	0.099	0.088	0.094	0.098	0.090	0.090	0.09
		AIR FLUX DENSITY kg/sec m2	1.217	1.230	1.093	1.168	1.217	1.118	1.118	1.11
	TEMP.	MAIN AIR INLET C	46.5	47	48	44	43	43	43	43
		MAIN AIR OUTLET C	850	915	525	550	600	690	750	770
	PRESSURE	MAIN AIR INLET mb	13.5	13.5	7.7	9.6	13.5	12.5	14.0	13.0
		MAIN AIR OUTLET HEADER mb	•	•	•	•	•	•	•	•
		MAIN AIR OUTLET mb	11.8	12.8	6.5	8.4	13.5	13.0	13.8	13.5
		AMBIENT TEMPERATURE C	•	•	•	•	•	•	•	•
		TORQUE AMMETER READING Amps	1.5	1.4-1.8	0.8-1.4	0.8-1.8	0.8-1.6	0.7-1.7	0.7-1.7	0.7-1.
		INTERSTREAM LEAKAGE %	12.6	3.4	13.1	23.4	19.2	26.3	29.2	25.6
		THERMAL EFFICIENCY %	10.9	•	16.3	7.1	54.2	14.7	14.3	13.0
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	0.09855	0.10632	0.10054	0.10329	0.09833	0.10417	0.10044	0.1093
		WASTE GAS OUTLET FLOWRATE kg/sec	0.07503	0.08822	0.08516	0.09068	0.09083	0.08627	0.08764	0.0851
		MAIN AIR INLET FLOWRATE kg/sec	0.10959	0.09345	0.12366	0.13301	0.09120	0.12667	0.11590	0.1289
		MAIN AIR OUTLET FLOWRATE kg/sec	0.01772	0.04024	0.02997	0.01474	0.10624	0.02814	0.02585	0.0254
		INTERSTREAM LEAKAGE FLOW kg/sec	0.00988	0.00313	0.01166	0.02214	0.01817	0.02377	0.02649.	0.0224
			SILLIMANITE	SILLIMANITE	ALUMINA	ALUMINA	ALUMINA	ALUMINA	ALUMINA	ALUMINA
			SILLIMANITE	SILLIMANITE	SILLIMANITE	SILLIMANITE	SILLIMANITE	SILLIMANITE	SILLIMANITE	SILLIMANI

		TEST NO.	8283.1	9283.1	10283.1	6483.1	21483.1	51083.1	61083.1	101083.1
		ROTOR SPEED RPM	1.0	1.5	1.5	1.0	1.0	1.0	1.0	1.0
WASTE GAS SIDE		COMBUSTION AIR FLOWRATE kg/sec	0.119	0.102	0.100	0.071	0.060	0.088	0.092	0.128
WASTE GAS SIDE		PRIMARY AIR FLOWRATE Kg/sec	0.082	0.071	0.069	0.057	0.060	0.036	0.030	0.045
WASTE GAS SIDE		NATURAL GAS FLOWRATE kg/sec	0.00484	0.0040	•	0.0008	NIL	0.0020	0.00245	0.0048
WASTE GAS SIDE		GAS VALVE SETTING No. of turns	•	•	•	•	•	•	•	•
WASTE GAS SIDE		GAS FLUX DENSITY kg/sec m2	1.538	1.317	•	0.892	0.745	1.118	1.173	1.650
WASTE GAS SIDE	TEMP.	WASTE GAS INLET C	830	880	940	480	45	870	1010	1000
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET HEADER C	•	•	•	•	•	•	•	•
WASTE GAS SIDE	TEMP.	WASTE GAS OUTLET C	600	660	625	280	45	515	590	770
WASTE GAS SIDE	PRESSURE	WASTE GAS INLET mb	6.0	6.0	6.7	2.0	0.8	3.8	4.4	9.5
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET HEADER mb	•	•	•	•	•	•	•	•
WASTE GAS SIDE	PRESSURE	WASTE GAS OUTLET mb	4.6	5.2	5.6	2.2	0.7	2.7	3.2	8.1
		INTERSTREAM PRESSURE DIFF. mb	-0.2	2.2	3.1	4.2	0.3	4.1	5.2	3.5
MAIN AIR SIDE		MAIN AIR INLET FLOWRATE kg/sec	0.124	0.100	0.104	0.091	0.075	0.088	0.090	0.123
MAIN AIR SIDE		AIR FLUX DENSITY kg/sec m2	1.540	1.242	1.392	1.130	0.932	1.093	1.118	1.528
MAIN AIR SIDE	TEMP.	MAIN AIR INLET C	40	42	41.5	47	45	49	51.5	50
MAIN AIR SIDE	TEMP.	MAIN AIR OUTLET C	465	670	525	265	45	555	650	645
MAIN AIR SIDE	PRESSURE	MAIN AIR INLET mb	7.4	8.7	9.8	6.9	5.9	8.1	9.8	14.2
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET HEADER mb	•	•	•	•	•	•	•	•
MAIN AIR SIDE	PRESSURE	MAIN AIR OUTLET mb	5.8	8.2	9.3	6.2	1.1	7.9	9.6	13.0
		AMBIENT TEMPERATURE C	•	•	•	•	•	•	•	•
		TORQUE AMMETER READING Amps	•	•	•	0.9-1.2	•	0.9	0.85	1.0
		INTERSTREAM LEAKAGE %	8.6	27.2	36.8	24.4	7.7	0.1	20.0	4.2
		THERMAL EFFICIENCY %	40.8	9.9	•	15.5	NIL	12.0	10.1	23.0
TRACER GAS METHOD		WASTE GAS INLET FLOWRATE kg/sec	0.12384	0.10600	0.11488	0.08733	0.06948	0.11959	0.11220	0.13137
TRACER GAS METHOD		WASTE GAS OUTLET FLOWRATE kg/sec	0.17306	0.08681	0.08962	0.08824	0.07851	0.08057	0.09127	0.12215
TRACER GAS METHOD		MAIN AIR INLET FLOWRATE kg/sec	0.13328	0.12660	0.14009	0.13123	0.08748	0.17586	0.14231	0.15429
TRACER GAS METHOD		MAIN AIR OUTLET FLOWRATE kg/sec	0.09156	0.02209	0.00940	0.03155	0.06415	0.02458	0.01905	0.06476
TRACER GAS METHOD		INTERSTREAM LEAKAGE FLOW kg/sec	0.01540	0.02536	0.03528	0.02259	0.00650	0.00009	0.01942	0.00539
			SILLIMANITE	SILLIMANITE	SILLIMANITE	MULLITE	MULLITE	SILLIMANITE	SILLIMANITE	SILLIMANITE
			SILLIMANITE	SILLIMANITE	SILLIMANITE	SILIMANITE	SILLIMANITE	SILLIMANITE	SILLIMANITE	SILLIMANITE

TOP: PERIPHERAL SHOE WEAR

Shoe	Avg.Gap between shoe surface and ring mm	No.of Revs.	Avge. Wear mm	Weight Loss g	No.of Revs.	Avge. Wear mm	Weight Loss g	No.of Revs.	Avge. Wear. mm	Weight Loss g
1ET	0.00	3144	0.022	0.14	2070	0.145	0.89	-	-	-
1MT	0.28	5388	0.083	0.51	2070	0.14	0.08	-	-	-
1UT	0.10	3144	0.022	0.14	2070	0.008	0.05	-	-	-
2ET	0.00	3732	+0.004	+0.03	2070	0.006	0.04	9070	0.091	0.55
2MT	0.28	3144	0.052	0.32	2070	0.036	0.22	-	-	-
2UT	0.63	3144	0.039	0.24	2070	0.036	0.22	-	-	-
3ET	0.49	3732	0.022	0.13	2070	0.042	0.26	9070	0.073	0.44
3MT	1.16	5026	0.054	0.33	2070	0.047	0.29	-	-	-
3UT	0.63	5388	0.030	0.16	2070	0.035	0.21	-	-	-
4ET	0.03	3732	0.022	0.13	2070	0.079	0.48	9070	0.060	0.36
4MT	0.10	5206	0.061	0.37	2070	0.023	0.14	-	-	-
4UT	0.36	5388	0.068	0.42	2070	0.014	0.08	-	-	-
5ET	1.32	3384	0.033	0.20	2070	0.060	0.37	-	-	-
5MT	1.80	5388	0.064	0.39	2070	0.043	0.26	-	-	-
5UT	1.80	5388	0.060	0.37	2070	0.039	0.24	-	-	-
6ET	0.03	3144	+0.002	+0.01	2070	0.006	0.04	-	-	-
6MT	0.05	3384	+0.007	+0.02	2070	0.022	0.13	-	-	-
6UT	0.04	3144	+0.018	+0.11	2070	0.045	0.28	-	-	-

NOTE: Mean weight loss after 2070 revs = 0.238 g against Mullite ring
" " " " 3144 " = 0.116 g " High Al_2U_3 ring
" " " " 3364 " = 0.016 g " High Al_2U_3 ring
" " " " 3732 " = 0.082 g " Sillimanitering
" " " " 5206 " = 0.354 g " High Al_2O_3 ring
" " " " 5388 " = 0.372 g " High Al_2O_3 ring
" " " " 9070 " = 0.454 g " Sillimanite

TOP: PERIPHERAL SHOE WEAR

BOTTOM: PERIPHERAL SHOE WEAR

Shoe	No.of Revs	Avge. Wear mm	Weight Loss g	No.of Revs	Avge. Wear mm	Weight Loss g	No.of Revs	Avge. Wear mm	Weight loss g
1UB	2070	0.011	0.325	11140	0.192	0.105	-	-	-
1MB	2070	0.048	0.142	11140	0.072	0.039	-	-	-
1EB	2070	0.122	0.360	11140	0.184	0.101	-	-	-
2UB	2070	0.056	0.165	11140	0.094	0.052	-	-	-
2MB	2070	0.044	0.130	11140	0.042	0.023	-	--	-
2EB	2070	0.094	0.278	shoe replaced 25.11.83			3384	0.190	0.343
3UB	2070	0.062	0.183	11140	0.068	0.037	-	-	-
3MB	2070	0.096	0.283	11140	0.158	0.089	-	-	-
3EN	2070	0.138	0.407	11140	0.232	0.127	-	-	-
4UB	2070	0.098	0.289	11140	0.094	0.052	-	-	-
4MB	2070	0.112	0.330	shoe replaced 25.11.83			3384	0.066	0.119
4EB	2070	0.058	0.171	11140	0.024	0.013	-	-	-
5UB	2070	0.120	0.354	11140	0.316	0.173	-	-	-
5MB	2070	0.132	0.390	shoe replaced 25.11.83			3384	0.128	0.231
5EB	2070	0.122	0.360	11140	0.298	0.163	-	-	-
6UB	2070	0.098	0.289	11140	5.706	3.130 *	-	-	-
6MB	2070	0.110	0.325	11140	0.392	0.215	-	-	-
6EB	2070	0.034	0.100	11140	0.074	0.041	-	-	-
	Mean Weight Loss = 0.56 g			Mean Weight Loss = 0.98 g			Mean Weight Loss = 0.78 g		
	Stand.Dev = 0.094			Stand.Dev = 0.060			Stand.Dev = 0.091		

NOTE: * Suspected unfired shoe not included in mean and standard deviation calculations

. All wear against sillimanite rotor and wear rings.

BOTTOM: PERIPHERAL SHOE WEAR

RADIAL SHOE WEAR

TOP :

Shoe Area = 29.1 cm^2

Load on Shoe = 3.37 KN/m^2

Shoe	Revs. Done	Avge. Wear mm	Weight Loss g	Avge. Weight Loss g
AOET	5388	0.014	0.016	0.09
WGET	5388	0.013	0.015	
WGUT	3384	+0.009	+0.03	0.22
WGMT	3384	0.074	0.25	

BOTTOM :

Shoe Area = 29.1 cm^2

Load on shoe = 1.01 KN/m^2

B AIR 1	5206	0.005	0.031	0.104
B AIR 2	5206	0.023	0.141	
B AIR 3	5206	0.023	0.141	

RADIAL SHOE WEAR

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