

ERRATA

Page Line

Throughout, 95% confidence limits refer to 95% limits of residual error, and do not refer to fiducial limits.

xi 1 for "are" read "rate"
14 delete "M/L"

xii 1,2,4 for "P" read "E/M"

12 eqn. 2.4 should read
$$\frac{\int_0^D G_{mm} D^3 dD}{\int_0^\infty G_n D^3 dD} = 0.5$$

15 8 add "where ε_v is the volume fraction of spray entering the reactor that has evaporated and"

15 20 for ref. (39) read (34) and delete (34) at end of line.

20 17 for ref. (14) read (13)

23 17 eqn. 2.34 should read " $\exp^{-k_{12} D^y}$ "

38 20,21 should read "... are based on the solution of a set of algebraic equations for first and second partial derivatives which have been..."

39 6 for "MURTAGH and" read "MURTAGH and "

48 28 should read "conditions for the flow system shown in Fig.3-2, and where the Laplace operator s is defined by equation 3.5"

52 25 for "250 μm " read "280 μm "

64 eqn. 4.5 should read " $z_{11} = m_{T1}/0.861$ "

65 eqn. 4.14 should read " $\varepsilon_{O_2} z_1 = 0.5(z_5 + z_8 - z_3) + z_6$ "

86 1 for "rpaid" read "rapid"

87 ordinate units should read " $\text{kg}/\text{m}^2 \text{s}$ "

94 eqn. 4.87 should read

$$Q_L^1 = \frac{F_8^1}{x_{O_2} (F_5^1 + F_7^1 - F_8^1)} \int_{298}^{T_8} C_{p(\text{air})} dT$$

128 9 for "constants" read "constraints"

139 * should read " $A = 2, 0, -2$ "

165 23 for "decreases" read "increases"

181 29 for "greater" read "smaller"

184 9 for "immdeiately" read "immediately"

199 12,17 for "13.8 kN/m^2 " read "170 kN/m^2 "

200 eqn. A2.5 should read " $\dot{m} \left(\frac{1}{\rho_g} + \frac{m_r \varepsilon_e}{\rho_v} \right) = v A_y$ "

Page	Line	
201	29	for ".0058" read ".0045"
218	4	for "2.84%" read "2.53%"
222		for "equivalence" read "equivalence"
239	22	add "The solids loss in the stack gas per kg. of fuel is
		$\frac{1.1244}{12 \times 60} \times \frac{1}{3.014} = .00052 \text{ kg solid/}$
		kg fuel. This corresponds to .052% of the fuel."

COMBUSTION OF RESIDUAL FUEL OIL:
MODELLING AND OPTIMISATION OF A TWO STAGE SYSTEM.

A thesis submitted
for the degree of
Doctor of Philosophy
in the Faculty of Engineering,
University of London.

by

Peter David Grout, B.A.Sc. (U.B.C.).

Department of Chemical Engineering
and Chemical Technology,
Imperial College of Science and Technology,
London, S.W.7.

July, 1970.

ABSTRACT.

A two stage spray combustion system burning residual fuel oil is mathematically modelled and optimised on the basis of experimental work performed on a pilot-scale furnace.

The object of this work is to determine optimal system operation on an economic basis in order to prevent high temperature corrosion by the elimination of excess air, and to reduce pollutant emission to acceptable levels. In particular, a solid emission of not more than 0.0040 kg solid/kg fuel is required.

The system consists of two high intensity combustion chambers in series, separated by a heat exchange section. Fuel is fed to the first chamber only. Combustion intensities of the order 10 MW/m^3 (10 Therm/hr ft^3) are achieved in each chamber at a firing rate of 4 g/s (31.7 lb./hr.). Both air and steam atomisation are investigated.

System operating limits are determined, in particular the thermal stability (blowout) limit of the second stage at or near stoichiometric conditions is measured. Experimental measurements of the maximum interstage heat removal using spray water cooling at blowout conditions are made, and then interpreted on a theoretical basis to determine the stability limits for industrial conditions where cooling is achieved by indirect means (without spray water).

Optimisation of the system is achieved on an economic basis using a constrained non-linear minimisation technique.

LIST OF CONTENTS.

	Page
Abstract.	i
List of Contents.	ii
List of Figures.	iv
List of Tables.	vii
Acknowledgements.	viii
Nomenclature.	ix
CHAPTER 1. INTRODUCTION.	1
CHAPTER 2. LITERATURE REVIEW.	4
2.1. System Criteria.	4
2.1.1. High Temperature Corrosion.	5
2.1.2. Emission of Pollutants.	6
2.1.3. Heat Generation and Transfer.	7
2.1.4. Furnace Control.	8
2.1.5. Economy.	9
2.2. System Models.	10
2.2.1. Fuel Oil Atomisation.	12
2.2.2. Fuel Spray Gasification.	13
2.2.3. Formation of Solids.	16
2.2.4. Combustion of Gases.	16
2.2.5. Combustion of Solids.	19
2.2.6. Thermal Stability.	23
2.2.7. Heat Transfer.	27
2.3. System Optimisation.	30
2.3.1. Single Stage Systems.	30
2.3.2. Multi-stage Systems.	34
2.3.3. Optimising Techniques.	37
2.4. System Implementation and Control.	39
CHAPTER 3. EXPERIMENTAL SYSTEM.	41
3.1. Data Available.	46
3.2. Data Required.	50
3.3. Equipment Required.	51
CHAPTER 4. DEVELOPMENT OF MATHEMATICAL MODELS.	56
4.1. Empirical Models.	57
4.2. Mechanistic Models.	58
4.3. Economic Factors.	58
4.4. Gas Composition and Temperature.	63
4.4.1. Equilibrium Model.	64
4.4.2. Reaction Rate Model.	67
4.5. Solids Formation.	69
4.5.1. Char Formation.	70
4.5.2. Soot Formation.	74
4.6. Solids Burning.	76
4.7. Heat Transfer.	86
4.7.1. Convection.	86
4.7.2. Radiation.	88
4.8. Combustion Stability.	90
4.8.1. Direct Cooling.	97
4.8.2. Indirect Cooling.	100
4.9. Solids Removal.	100

	iii.
	Page
CHAPTER 5. COMPARISON OF EXPERIMENTAL AND PREDICTED RESULTS.	102
5.1. Gas Composition and Temperature.	103
5.2. Solids Formation in the First Stage.	103
5.3. Heat Transfer.	108
5.4. Solids Burning in the Second Stage.	111
5.5. Thermal Stability.	114
CHAPTER 6. SYSTEM OPTIMISATION.	124
6.1. Objectives.	124
6.2. General Minimisation Program.	125
6.3. Variables.	127
6.4. Constraints.	128
6.5. Modifications.	135
CHAPTER 7. INTERPRETATION OF RESULTS.	137
7.1. Designed Model Experiments.	137
7.2. Gas Composition and Temperature.	138
7.3. Solids Formation.	146
7.4. Interstage Heat Transfer.	153
7.5. Solids Burning.	156
7.6. Combustion Intensity and Thermal Stability.	168
7.7. Air Atomisation.	173
7.8. Optimisation of the First Stage.	176
7.8.1. Minimum Solid Emission.	177
7.8.2. Economic Optimisation.	179
7.8.3. Combustion Volume and Intensity.	181
7.9. Optimisation of the Two Stage System.	183
7.9.1. Economic Optimisation.	184
7.9.2. Combustion Volume and Intensity.	187
7.9.3. System Control.	187
CHAPTER 8. CONCLUSIONS.	189
CHAPTER 9. RECOMMENDATIONS.	194
CHAPTER 10. APPENDICES.	196
10.1. Specification of Ministry of Defence (Navy) Reference Fuel Oil "Vanilla".	196
10.2. Design Calculations and Operating Procedure for the Spray Cooler Probe.	198
10.3. Integration of Drop Size Number Distribution Function and Calculation of D_{sm} , N_c , N_s and n .	206
10.4. Tabulated Results.	209
10.5. Analysis of Error.	215
10.6. Calculation of Equilibrium Constants from Thermodynamic Properties.	236
10.7. Determination of Stack Gas Solids and Collection Error.	238
REFERENCES.	242

LIST OF FIGURES.

Figure Number		Page
2-1.	Gasification and combustion of residual fuel oil.	11
2-2.	Comparison of carbon burning models for a 100 μm particle.	21
2-3.	Effect of inlet temperature on combustion stability.	26
3-1.	Schematic of experimental system.	42
3-2.	First stage chamber.	43
3-3.	Second stage chamber.	45
3-4.	Schematic of experimental subsections showing sampling points.	47
3-5.	Water cooled spray probe.	53
3-6.	Spray probe configuration.	54
4-1.	The effect of evaporation efficiency on total number of char particles formed.	73
4-2.	The effect of particle emissivity on reaction rate.	83
4-3.	Comparison of carbon burning models for a 1.0 μm particle.	84
4-4.	Comparison of carbon burning models for a 0.1 μm particle.	85
4-5.	The effect of oxygen concentration on reaction rate.	87
4-6.	Mass and enthalpy balances on the well mixed section of the second stage.	92
4-7.	Typical temperature-conversion diagram.	93
4-8.	The effect of water vapour mole fraction on gaseous reaction rates.	98
4-9.	The reaction rates of gaseous combustibles.	99
5-1.	Comparison of measured and predicted wet gas compositions.	104
5-2.	Comparison of measured and predicted combustion gas temperatures.	105
5-3.	Comparison of measured and predicted first stage solids.	107
5-4.	Mass distribution of solids from first and second stages.	109
5-5.	Comparison of measured and predicted interstage heat transfer rates.	110
5-6.	Progress of second stage combustion reactions.	113

Figure Number		Page
5-7.	The effect of mixing on combustion stability.	115
5-8.	Maximum interstage heat removal for $0.95 < \phi_2 < 1.05$.	118
5-9.	Temperature-conversion diagram at $\phi_1 = 1.28, \phi_2 = 1.00$.	121
5-10.	Temperature-conversion diagram at $\phi_1 = 2.00, \phi_2 = 1.00$.	122
6-1.	System operating region.	131
6-2.	Maximum interstage heat removal constraints.	134
7-1.	Calculated gas composition and temperature for steam atomisation.	141
7-2.	Calculated gas composition and temperature for air atomisation.	143
7-3.	Comparison of first stage solids for air and steam atomisation.	145
7-4.	Surface mean diameter of sprays from twin fluid atomisers.	147
7-5.	First stage solids as % of fuel at $F_2/F_1 = 0.5$.	149
7-6.	First stage solids as % of fuel at $F_2/F_1 = 0.7$.	150
7-7.	First stage solids as % of fuel at $F_2/F_1 = 0.9$.	151
7-8.	First stage solids as % of fuel at $F_2/F_1 = 1.3$.	152
7-9.	The effect of steam/fuel mass ratio on first stage solids.	154
7-10.	The effect of steam/fuel mass ratio on particle mass distribution.	155
7-11.	Heat transfer surface for maximum interstage heat removal at $F_2/F_1 = 0.9$.	157
7-12.	Heat transfer surface for maximum interstage heat removal at $F_2/F_1 = 1.3$.	158
7-13.	Particle mass distribution after first and second stages.	160
7-14.	Second stage solids as % of fuel at $F_2/F_1 = 0.5$.	161
7-15.	Second stage solids as % of fuel at $F_2/F_1 = 0.7$.	162
7-16.	Second stage solids as % of fuel at $F_2/F_1 = 0.9$.	163
7-17.	Second stage solids as % of fuel at $F_2/F_1 = 1.3$.	164

Figure Number		Page
7-18.	The effect of interstage heat removal on solid combustion in the second stage.	166
7-19.	Solid combustion efficiency at $F_2/F_1 = 0.9$	167.
7-20.	Comparison of solid combustion efficiency for air and steam atomisation.	169
7-21.	First stage combustion intensity at $F_2/F_1 = 0.7$.	170
7-22.	First stage combustion intensity at $F_2/F_1 = 0.9$.	171
7-23.	First stage combustion intensity at $F_2/F_1 = 1.3$.	172
7-24.	Second stage combustion intensity at $F_2/F_1 = 0.9$.	174
7-25.	Maximum interstage heat removal as % of first stage heat release.	175
7-26.	Comparison of export steam value and operating costs.	180
7-27.	The effect of first stage volume on first stage solids.	182
A5-1.	The effect of uncertainty in experimental settings on particle mass distribution from the first stage.	224
A5-2.	The effect of uncertainty in experimental settings on particle mass distribution from the second stage.	229
A5-3.	Error associated with arithmetic mean diameter assumption.	231
A5-4.	The effect of uncertainty in interstage heat removal on blowout conditions.	234
A7-1.	The effect of scrubber water flow rate on sampling error.	241

LIST OF TABLES.

Table Number		Page
2-1	Comparison of combustion models.	31
4-1	Capital cost functions.	60
4-2	Installation cost and operating power functions.	61
4-3	Utility, raw material and product values.	62
6-1	System constraints.	130
7-1	Summary of conditions for designed model tests.	139
7-2	Range of settings for model tests.	140
7-3	Test results for first stage solids minimisation.	178
A1-1	Residual fuel oil ash composition.	196
A2-1	Spray cooling requirements.	198
A2-2	Length of cooling section for spray cooling.	202
A4-1	Comparison of measured and predicted first stage solids.	210
A4-2	Comparison of measured and predicted second stage solids.	211
A4-3	Experimental blowout settings.	212
A4-4	Interstage heat removal at blowout.	213
A4-5	Calculated blowout conditions.	214
A5-1	Accuracy of experimental settings.	216
A5-2	Spray water flow rates.	217
A5-3	Error in second stage temperature due to uncertainty in experimental settings.	217
A5-4	Error in first stage solids due to uncertainty in experimental settings.	222
A5-5	Errors associated with first and second stage solids and solid combustion efficiency.	228
A5-6	Errors associated with interstage heat removal at blowout conditions.	233
A7-1	System water balance errors.	240

ACKNOWLEDGEMENTS.

The author wishes to thank Dr. Paul Eisenklam, D.Sc. for his direction of this work and for his encouragement throughout.

Facilities in the Department of Chemical Engineering and Chemical Technology were made available by permission of Professor A.R.Ubbelohde, F.R.S.

Financial support from the Board of Trade in the form of an Athlone Fellowship, and from the National Research Council of Canada made this work possible. The author wishes to thank the Athlone Fellowship Managing Committee for their help and advice during his studies in Great Britain.

The author also wishes to thank his colleagues for their help and constructive criticisms during the execution of this work.

In particular, the author wishes to thank his wife Karen, to whom this thesis is dedicated, for her love and encouragement at all stages of the work.

NOMENCLATURE.

All dimensions are in terms of S.I. Units
unless other wise specified.

Length	L	metre	m
Mass	M	kilogram	kg
Time	T	second	s
Thermodynamic Temperature	θ	degree Kelvin	K

Derived units used

Force	F	newton	$N = \text{kg m/s}^2$
Energy	E	joule	$J = N \text{ m}$
Power	P	watt	$W = J/s$

<u>Symbol</u>		<u>Dimensions</u>
a	particle radius	L
A_c	total projected particle cross sectional area.	L^2
A_i	cross sectional area of inlet ports.	L^2
A_p	projected particle cross sectional area per unit volume.	L^{-1}
A_x	heat exchange surface area.	L^2
A_y	cross sectional area of heat exchange tube.	L^2
b_s	stoichiometric coefficient.	-
B	Transfer number.	-
B_c	Transfer number for combustion.	-
c	concentration.	M/L^3
$c(\underline{x})$	constraint function.	-
c_1	constant.	L^{-1}
C_D	drag coefficient.	-
C_p	molal heat capacity at constant pressure	$E/M\theta$
C_v	molal heat capacity at constant volume	$E/M\theta$

Symbol.		Dimensions
D	diameter.	L
D_{med}	mean evaporative diameter.	L
D_{mm}	mass median diameter.	L
D_p	particle diameter.	L
D_{sm}	surface mean diameter.	L
D_μ	arithmetic mean diameter.	L
$\mathcal{D}$	molar diffusivity.	L^2/T
E_a	activation energy.	E/M
F	probability factor.	-
$F_1(F_1')$	mass(molar)flow rate of fuel.	M/T
$F_2(F_2')$	mass(molar)flow rate of atomising fluid.	M/T
$F_3(F_3')$	mass(molar)flow rate of forward air.	M/T
$F_4(F_4')$	mass(molar)flow rate of reverse air.	M/T
$F_5(F_5')$	total mass (molar) flow rate from stage 1.	M/T
$F_6(F_6')$	total mass (molar) flow rate from stage 2.	M/T
$F_7(F_7')$	mass (molar) flow rate of stage 2 inlet air.	M/T
$F_8(F_8')$	mass (molar) flow rate of stage 2 outlet air.	M/T
F_{pm}	perfectly mixed fraction of stage 2 well mixed section.	-
$\mathbf{g}$	gradient vector.	-
G_n	number size distribution function.	-
G_w	weight size distribution function.	-
h_c	convective heat transfer coefficient.	$E/L^2\theta T$
h_t	overall heat transfer coefficient.	$E/L^2\theta T$
$\underline{H}$	Hessian matrix.	
H_c	heat of combustion.	E/M
H_f	heat of formation.	E/M
H_r	heat of reaction.	E/M
H_s	sensible heat.	E/M
H	enthalpy.	E/M
k	thermal conductivity.	$P/L\theta$
k_1-k_{15}	constants.	
$k_{s1}-k_{s6}$	rate constants.	

Symbol.		Dimensions.
K	global are constant.	
K_d	diffusional resistance constant.	T/L
K_e	equilibrium constant.	
K_s	surface resistance constant.	T/L
l_m	microscale for turbulent mixing.	L
L	characteristic dimension.	L
L_a	air loading.	--
L_b	characteristic dimension of the radiating enclosure.	L
$\dot{m}$	specific mass reaction rate, or mass flux.	M/L^2T
m_c	mass of char per kg of fuel.	--
m_s	mass of soot per kg of fuel.	--
m_T	total mass of solid per kg of fuel.	--
m_r	mass ratio of spray water to combustion gas.	M/L
M	molecular weight.	
$\dot{M}$	mass flow rate.	M/T
$\dot{M}_a$	mass flow rate of air.	M/T
M_c	molecular weight of carbon.	
$\bar{N}$	fractional number of drops (equation 2.2)	
N_{Nu}	Nusselt number.	
N_{Pr}	Prandtl number.	
N_{Re}	Reynolds number.	
N_c	number of char particles.	
N_s	number of soot particles.	
N_t	total number of drops.	
p	partial pressure.	F/L^2
$\underline{p}$	displacement vector.	L
P	pressure.	F/L^2
$\dot{q}$	heat transfer rate.	P
$\underline{q}$	change of gradient vector.	
Q	heat generation or loss.	E
Q'	heat generation or loss per mole of oxygen.	E/M
$\dot{Q}_c$	convective heat transfer rate.	P
$\dot{Q}_r$	radiant heat transfer rate.	P

Symbol		Dimensions
$\dot{Q}_i$	interstage heat removal rate.	P
$\dot{Q}_{\text{max}}$	maximum interstage heat removal rate limited by 600 K gas temperature.	P
$\dot{Q}_R$	maximum interstage heat removal rate limited by blowout.	P
r	radial distance.	L
$\dot{r}$	molar reaction rate (forward).	M/L^3T
$\dot{r}'$	molar reaction rate (reverse).	M/L^3T
$\dot{r}^*$	molar reaction rate (net).	M/L^3T
R	Gas constant.	$L^2/T^2\theta$
s	Laplace operator.	
S	entropy.	E/θ
t	time.	T
T	temperature.	θ
T_{cm}	convective mean temperature (equation 2.44).	θ
T_{rm}	radiant mean temperature (equation 2.43).	θ
T_s	surface temperature.	θ
T_w	wall temperature.	θ
T_{IN}	inlet gas temperature to cooling tube.	θ
T_{OUT}	outlet gas temperature from cooling tube.	θ
T_2	steam temperature.	θ
v	relative velocity.	L/T
v_i	inlet velocity.	L/T
V	volume.	L^3
$\bar{w}$	weight fraction (equation 2.32).	
x	mole fraction.	
$\underline{x}$	state variable vector.	
$\underline{x}_s$	slack variable vector.	
X	molar ratio of CO_2 to CO at carbon particle surface.	
z_1-z_{14}	constants.	
Z	collision factor.	

Symbol		Dimensions
α	reaction order constant.	
β_T	turbulent mixing rate.	T^{-1}
β	reaction order constant.	
γ	surface tension.	E/L^2
δ	reaction order constant.	
ϵ_e	water spray evaporation efficiency.	
ϵ_v	fuel spray evaporation efficiency.	
ϵ_{O_2}	oxygen consumption efficiency.	
ϵ_T	total emissivity.	
ϵ_c	char emissivity.	
ϵ_s	soot emissivity.	
ϵ_n	non-luminous emissivity.	
λ	latent heat.	E/M
ϕ	equivalence ratio.	
η	absolute viscosity.	FT/L^2
ν	kinematic viscosity.	L^2/T
τ	mean residence time.	T
σ	standard deviation.	
ρ	density.	M/L^3
ρ_L	liquid density.	M/L^3
ρ_g	gas density.	M/L^3
ρ_p	particle density.	M/L^3
ρ_c	char density.	M/L^3
ρ_s	soot density.	M/L^3
Δ	finite change.	
Γ	Gamma function.	

CHAPTER 1.

INTRODUCTION.

Mathematical models for the high intensity combustion of residual fuel oils are developed in this work. These models are then used in conjunction with minimisation techniques to optimise the operation of a two stage combustion reactor with independent interstage heat removal. The optimisation of theoretical multistage gaseous combustion reactors has been achieved previously, however this is the first attempt to optimise an operating pilot-scale spray combustion reactor.

Increasing use is being made of low cost high intensity combustion systems as an alternative to the more expensive natural and forced draught systems. Recent restrictions on the level of solid emission from combustion equipment and concern over excessive noise levels has led to the further acceptance of the inherently quieter and cleaner high intensity combustion chambers.

They have been employed successfully for the combustion of distillate fuels, however use of low cost residual fuels containing sodium, vanadium and sulphur impurities has met with only limited success. In particular, combustion of such residual oils has led to severe corrosion of heat exchange surfaces as well as high pollutant emission and combustibles loss in the exhaust gases.

Combustion control is the most flexible yet economic method of approaching these problems. This method

involves control of both the design and operation of the system, in particular the accurate adjustment of fuel and air so as to exclude excess air and prevent high temperature corrosion. A recent advance in combustion control involves staged combustion with independent interstage heat removal, which allows second stage temperature control without altering fuel and air settings. This two stage combustion concept improves heat transfer characteristics by controlling the solid burden from the first stage and then reducing it to acceptable levels below the statutory limit of 0.0040 kg solids/ kg fuel by combustion in the second stage.

The object of this work is to assess the relative merits of high combustion efficiency, high heat transfer rates, low solid emission and the use of air or steam atomisation, in a two stage high intensity (of the order 10 MW/m³) system. Fuel is injected in the first stage only and the system is operated so as to exclude excess air completely. The optimal combustion conditions both from an economic and operational point of view are determined.

This system is analysed using chemical reactor theory, and mechanistic and empirical models are developed. In particular, an experimental investigation of the effects of interstage heat removal on combustion stability in the second stage is made and the stability limits at and near stoichiometric operation determined.

The optimisation study of the system is made using a constrained quadratic minimisation technique known

to be successful for non-linear systems. It was found that the solid emission limitations could easily be met using the two stage system. They could not, however, be met using a single stage system. The most economic mode of operation for any first stage conditions is achieved when interstage heat removal is maximised. Up to 92% of the heat released in the first stage can be removed interstage-wise at first and second stage equivalence ratios of 1.9 and 1.0 respectively.

The most economic operation of the system to generate high pressure steam on an annual basis, consistent with solid emission limitations, is achieved at two different conditions depending on the value of the steam produced. When the steam generated is worth £ .00092/kg, optimal operation is achieved using steam atomisation at a first stage equivalence ratio of 1.5, an atomising steam/fuel mass ratio of 0.9 and a fuel flow rate of 2.00 g/s. When the steam generated is worth £ .00150/kg, optimal operation is achieved using steam atomisation at a first stage equivalence ratio of 1.9, an atomising steam/fuel mass ratio of 0.9 and a fuel flow rate of 3.20 g/s.

The investigation of system design indicates that the second stage chamber should be redesigned, in particular, the plug flow section can be removed completely without affecting system performance.

CHAPTER 2.

LITERATURE REVIEW.

The possibility of burning heavy residual fuel oil containing sodium, vanadium and sulphur as impurities, whilst preventing high temperature corrosion and eliminating pollution hazards, has led to the study of two stage combustion. In order to improve existing furnace design and operation, a systematic study has been made of the available information on two stage combustion, including extensive experimental work on one existing pilot-scale two stage furnace. These results have indicated that reductions in both corrosion and pollution can be achieved by careful regulation of the conditions within the furnace i.e. combustion control. This experimental work has shown that the following study, which is a more detailed analysis of previous results, is warranted.

The study requires the determination of system criteria upon which the evaluation of the system is based; the formulation and testing of models to describe the system; and the analysis of the models and system to improve or optimise design and operation. These findings may then be used for the implementation and control of a working system.

2.1. System Criteria.

Small high intensity combustion chambers are used for process heating and in package boilers where space limitations occur. These combustors must produce clean exhaust gases without causing corrosion of heat exchange surfaces, as well as being economic to operate.

The determination of furnace design and operating criteria requires the definition of the critical operating and process dependent variables. The application of the combustion unit usually determines which variables must be controlled and which, in fact, can be controlled.

The system upon which this study is based has been described in detail previously (1,2). The system consisted of a high intensity gasifier separated from the second stage gas burner by a heat exchanger. The fuel used was a reference residual fuel oil as specified in Appendix 1.

2.1.1. High Temperature Corrosion.

The combustion of residual fuel oil containing sodium, vanadium and sulphur as impurities is known to produce combustion products which, under oxidising conditions, cause corrosion of heat exchange surfaces. HTUN (2) and ARCHER (3) have made extensive surveys of high intensity residual fuel oil combustion and have concluded that combustion control rather than the use of additives is one of the most promising methods of controlling fireside corrosion in furnaces. Experimental evidence obtained by these workers indicated that potentially corrosive metal oxides are formed only in the presence of excess air. The formation of these compounds may be prevented by stoichiometric or fuel rich combustion in a well mixed furnace.

Heat exchange surface temperatures have been shown to influence the severity of corrosion resulting from sodium-vanadium compounds and metal sulphides which deposit at the surface (4). Gas borne carbon is also known to

reduce corrosion under low excess air conditions (5). The control of furnace temperature and carbon formation can thus be used to influence the rate of corrosion at heat exchange surfaces.

In multistage systems, both the carbon loading and combustion gas temperature at the exit may be controlled by interstage heat removal as well as by equivalence ratio*.

2.1.2. Emission of Pollutants.

The hazards arising from solid and gaseous pollutant emission from industrial furnaces has led to legislation limiting the amount of pollutants that may be discharged into the atmosphere (6,7). Early regulations were based on smoke number and overall acidity or alkalinity, present controls however specify maximum levels for both solid and gaseous pollutants.

Solid pollutants are generally fly ash and carbonaceous solids, and are specified as a weight per unit volume of dry exhaust gases or as a weight per unit fuel. In 1967, the Ministry of Housing and Local Government (6) specified the maximum level of dust and grit emission from boilers and furnaces with a fuel firing rate less than 0.35 kg/s as 0.4% by weight of the fuel fired. This corresponds to a solid burden of 395 mg/m^3 in the dry exhaust gases at stoichiometric conditions for the residual oil used in this study. For installations with firing rates in excess of 0.35 kg/s, the maximum allowable emission decreased linearly with firing rate to 0.2% of the fuel at a firing rate of 3.58 kg/s. More recently, a recommendation

* Equivalence ratio ϕ is defined as the ratio of stoichiometric air required to actual air used for combustion.

has been made for an absolute maximum emission of 460 mg/m^3 solids in the dry exhaust gas (7).

Gaseous pollutants from furnaces are usually sulphur compounds such as hydrogen sulphide, sulphur dioxide, sulphur trioxide, and gaseous mercaptans. More recently, consideration has been given to the possible formation of oxides of nitrogen in high intensity furnaces (8).

Control of solid emission can be affected either by preventing solid formation or by subsequent removal. The formation of carbonaceous solids in spray combustors is controlled by chamber design and operation. Combustible solids may be removed either by burning or by mechanical separation using a scrubber, filter or precipitation system.

The formation of gaseous pollutants such as H_2S and SO_3 is prevented by fuel rich combustion (2,7). It appears that the formation of SO_2 is also reduced, as a larger fraction of the sulphur entering in the fuel is retained in the solids formed (1,2). It is known (8) that the formation of oxides of nitrogen is favoured by air rich high temperature combustion. Steam injection is known to inhibit the formation of these oxides by lowering the combustion temperature. Recent work by WASSER (9) has indicated a 50% reduction in the nitric oxide emission from domestic furnaces can be achieved by the reduction of excess air from 50% to 0%.

2.1.3. Heat Generation and Transfer.

The generation of useful heat requires the best possible utilisation of the fuel available with respect to both combustion efficiency and the production of combustion

gas suitable for the type of heating required. In most cases it is desirable to generate heat rapidly within a small volume while maintaining high combustion efficiency. This leads to high intensity combustion characterised by high specific heat release rates of the order 10 MW/m^3 . High combustion efficiency, however, does not ensure a combustion gas with good heat transfer characteristics.

Heat transfer in furnaces occurs predominantly by a combination of convection and radiation. Many studies have been made to determine the convective and radiative heat transfer properties of solid laden combustion gases. These have been reviewed by HOTTEL and SAROFIM (10). Convective heat transfer is dependent on gas temperature and velocity while radiant heat transfer which predominates, depends on gas temperature and emissivity. The emissivity, as well as gas temperature and velocity, is determined by combustion conditions and furnace geometry.

High combustion intensity and high heat transfer rates allow the use of smaller furnaces and heat exchangers.

2.1.4. Furnace Control.

A prerequisite for the modelling and optimisation of a system is that the system is controllable. The application of the furnace determines the degree of control required whilst the design governs whether it can be controlled i.e. controllability. Previous work on the experimental system studied (2,3,11) has indicated that the flowrates of all inlet streams are controllable. Inlet fuel and steam temperatures are controllable but inlet air and water temperatures are not. It is known (2,3) that air

and cooling water temperature as well as cooling water flowrate have negligible effect on system performance. All inlet flows and temperatures were controlled excluding air and water temperature.

Consideration must be given to the use of system models for future on-line control. While the implementation of on-line control is not the object of this study, the complexity of the models developed should be noted with regard to potential computer control.

2.1.5. Economy.

The simultaneous consideration of the preceding criteria requires knowledge of the relative merit of each. The best combination of design and operation in the context of the chosen criteria is determined by an economic analysis of the system.

The criteria can be related by considering design and process variables in terms of capital, operating and raw material costs as well as value of saleable products. LOCKWOOD (12) has studied the design of package boilers considering the heat transfer criterion only in the economic analysis. His work involves two simplifying assumptions, i.e. the combustor is perfectly mixed at the adiabatic flame temperature and the emissivity is invariant. The analysis is thus inapplicable to industrial boilers as neither condition is satisfied. This is, however, the only study detailing furnace design from an economic point of view.

The use of an economic criterion such as maximum profit or return on investment appears to be the most promising method of analysing the overall system.

2.2. System Models.

Combustion systems are characterised by the type of fuel burned and can be classified as gas, liquid or solid fuel reactors. Liquid and solid fuel furnaces are heterogeneous reactors with oxygen present in the gas phase. The only truly homogeneous combustion systems burn gaseous fuels containing no suspended particles.

Extensive use has been made of reaction models to describe combustion processes using pulverised coal (13), iso-octane (14) or kerosene (15) as fuels. The basic models have been extended to include the effects of mixing and heat transfer (13,15), however the models have not been applied to multi-gas fuel systems or systems where solid, liquid and gas fuels react simultaneously.

Fuel rich combustion of residual fuel oil involves simultaneous solid, liquid and gas reactions. In a liquid fuel gasifier, reaction rate and hence heat release rate are controlled by mass transfer from a cloud of fuel drops. In a gas burner, solids and gases react in parallel, the gases being kinetically controlled and the solids being mass transfer or kinetically controlled.

The progress of the reactions is as shown in Fig.2-1. The initial steps involve the formation of a fuel spray which is then evaporated to yield combustible gas and a tarry residue. Soot forms from the gas phase whilst char forms from the tarry residue.

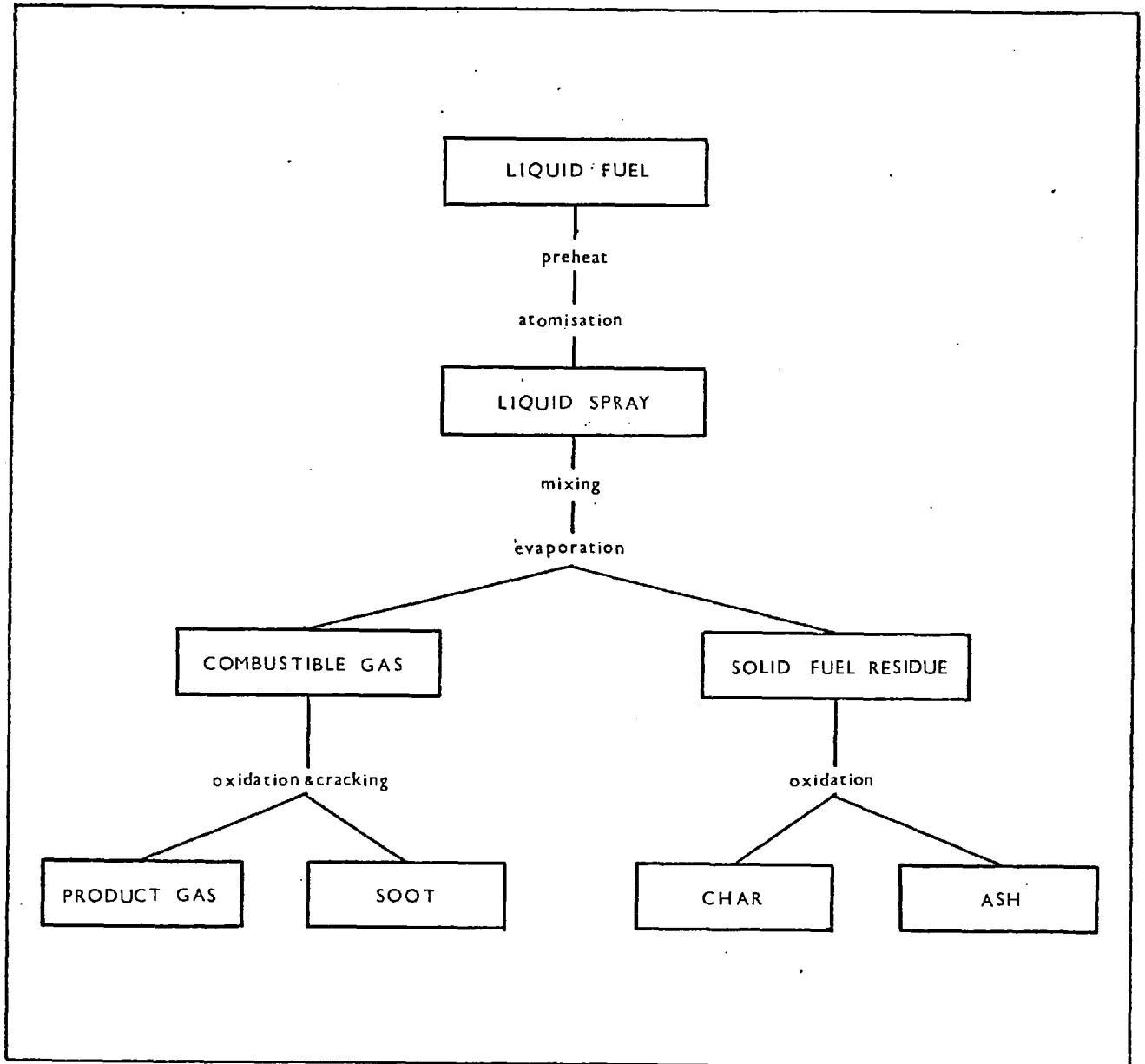


FIG.2-1 GASIFICATION AND COMBUSTION OF RESIDUAL FUEL OIL

2.2.1. Fuel Oil Atomisation.

The use of twin fluid atomisers for fuel oil atomisation is well documented (16,17,18). The characteristics of the resulting spray size distribution (18) have been described (19) by the use of a number size distribution function, G_n .

$$G_n = k_1 D^m \exp -k_2 D^n \quad 2.1$$

where

$$G_n = \frac{d\bar{N}}{dD} \quad 2.2$$

and $\bar{N}$ is the fractional number of drops less than diameter D . Both n and m are constants.

The spray is completely defined if a characteristic dimension such as the surface mean diameter D_{sm} or mass median diameter D_{mm} is known.

$$D_{sm} = \frac{\int_0^\infty G_n D^3 dD}{\int_0^\infty G_n D^2 dD} \quad 2.3$$

$$D_{mm} = \frac{\int_0^\infty G_n D^3 dD}{\int_0^\infty G_n D^3 dD} = 0.5 \quad 2.4$$

Both the spray size distribution and characteristic diameter are determined by atomiser design, fluid properties and operating conditions. The constants in equation 2.1 have been determined from experimental measurements on sprays (34). Expressions relating D_{sm} and D_{mm} to atomiser design, fluid properties and operating conditions have been developed by NUKIYAMA and TANASAWA(16) and by WIGG (17) respectively.

$$D_{sm} = \frac{5.85 \times 10^5}{v} \sqrt{\frac{\gamma}{\rho_L}} + 1684 \left(\frac{n}{\sqrt{\gamma \rho_L}} \right)^{0.45} \left(\frac{1000}{j} \right)^{1.5} \quad 2.5$$

$$D_{mm} = \frac{2 \times 10^5}{v} \sqrt{\gamma \left(1 + \frac{F_1}{F_2} \right)} (F_1 d)^{0.1} \frac{\gamma^{0.2}}{f_g^{0.3}} \quad 2.6$$

v	gas/liquid relative velocity at the nozzle exit	m/s
γ	surface energy	J/m ²
ρ_L	liquid density	kg/m ³
j	gas/liquid volume ratio at the nozzle exit	
η	absolute viscosity	Ns/m ²
ν	kinematic viscosity	m ² /s
F_1	fuel flow rate	kg/s
F_2	atomising fluid flow rate	kg/s
d	height of atomising fluid annulus	m
ρ_g	gas density	kg/m ³

2.2.2. Fuel Spray Gasification.

The rates of evaporation and combustion of single drops falling through a hot atmosphere have been studied (20). From these studies, the effect of mass transfer on heat transfer has been expressed using the transfer number B.

$$B = \frac{C_p \Delta T}{\lambda} \quad 2.7$$

where C_p is the heat capacity of the vapour, ΔT is the temperature difference between the drop surface and the surroundings and λ is the latent heat of vaporisation. For evaporating drops where combustion and radiant heat transfer occur, the transfer number is modified to

$$B_c = \frac{C_p \Delta T + H_c}{\lambda + q_r} \quad 2.8$$

where q_r is the radiant heat transfer rate per unit of fuel evaporated and H_c is the heat of combustion normalised with respect to oxygen concentration.

This work has also shown that as predicted from laminar boundary layer theory, the Nusselt number for moving drops undergoing mass transfer $\bar{N}_{Nu}$ is related to the Nusselt number where no mass transfer occurs, N_{Nu} , viz.

$$\bar{N}_{Nu} = \frac{N_{Nu}}{1 + B_c} \quad 2.9$$

In combustion systems where the Prandtl number N_{Pr} is approximately constant, the following equation in terms of Reynolds number N_{Re} has been shown to apply (20).

$$\bar{N}_{Nu} (1 + B_c) = 2 + k_3 N_{Re}^{0.5} \quad 2.10$$

Expressing $\bar{N}_{Nu}$ in terms of mass transfer,

$$\bar{N}_{Nu} = - \frac{k \lambda}{4k \Delta T} \frac{d D^2}{dt} \quad 2.11$$

where k is thermal conductivity.

Equations 2.10 and 2.11 may be combined and integrated to give

$$\int_0^D \frac{D dD}{2+k_3 N_{Re}^{0.5}} = \frac{2k_4 \Delta T}{R \lambda (1+B_c)} \int_0^t dt = k_4 D^n \quad 2.12$$

HOPKINS and EISENKLAM (19) have shown that the simplification of the integral expressions to $k_4 D^n$ is accurate to $\pm 5\%$, and that the rate of evaporation of a single drop travelling at constant relative velocity is

$$\frac{dD}{dt} = - \frac{1}{nk_5} D^{1-n} \quad 2.13$$

where

$$k_5 = \frac{R \lambda (1+B_c) k_4}{2k \Delta T} \quad 2.14$$

The dependence of drop relative velocity on drop diameter has been determined using the SCHILLER-NAUMANN steady state drag equation for non-evaporating spheres and the transfer number concept to account for mass transfer effects (20). For particles with $N_{Re} < 100$ the equation is

$$3\pi\mu Dv (1 + 0.15 N_{Re}^{0.687}) = \frac{\pi}{6} R D^3 (1+B) \frac{dv}{dt} \quad 2.15$$

HOPKINS (34) has solved equations 2.13 and 2.15 in terms of initial velocities v_0 and diameters D_0 assuming N_{Re} to be constant over small changes in v and D . viz

$$\log \frac{v}{v_0} = k_6 \left(\log \left(\frac{D}{D_0} \right) \right)^2 + k_7 \log \left(\frac{D}{D_0} \right) \quad 2.16$$

where

$$k_6 = \frac{3.75 \times 10^5 D_0}{T - 950} \quad 2.17$$

$$k_7 = \frac{446 \cdot v_0}{T^{0.33}} \quad 2.18$$

and T is the temperature of the surroundings.

It has been shown that single drop evaporation theory can be applied to unidimensional sprays (21) and that for a spray with an initial drop size distribution corresponding to G_n in equation 2.1, with m equal to $n - 1$, the fractional evaporation efficiency $\Delta \mathcal{E}_v$ of the spray in an incremental reactor is

$$\Delta \mathcal{E}_v = 1 - \exp \left(- \frac{k_2 \Delta t}{k_5} \right) \quad 2.19$$

where all drops have the same initial velocity relative to the gas over the time interval Δt . Equation 2.19 has been used in conjunction with equation 2.16 for sprays with varying velocities (34).

For plug flow reactors with mean residence time τ , equation 2.19 becomes

$$\mathcal{E}_v = 1 - \exp \left(- \frac{k_2 \tau}{k_5} \right) \quad 2.20$$

and for perfectly mixed reactors

$$\mathcal{E}_v = 1 - \frac{k_5}{k_2 \tau + k_5} \quad 2.21$$

Equation 2.19 may be applied to any furnace or reactor if the mixing history is known.

Studies of the mixing patterns in furnaces under cold (13,43) and hot (39) conditions using inert tracer(34) techniques have shown that high intensity furnaces cannot be described by perfectly mixed or plug flow models alone. The analysis of residence time distributions in hot furnaces has indicated that the mixing characteristics may be duplicated by the use of perfectly stirred reactors in series and recycle. In particular, it has been shown (34) that this type of model represents the mixing characteristics of the first stage combustion chamber used in this work.

2.2.3. Formation of Solids.

Pure component fuel drops entering a high temperature oxidising environment experience an ignition delay until the drop reaches its saturation vapour temperature. If the reactor temperature is above the ignition temperature of the fuel vapour, combustion occurs and the drop is consumed. The liquid temperature of wide cut residual fuel oils depends on the fractional evaporation and follows the boiling point curve of the fuel oil (22), hence fuel properties change as evaporation proceeds. In previous analyses (34) this increase in drop temperature and change in fuel properties has been ignored. Although the effect on gas composition and temperature is small, at high evaporation efficiencies, the change in the amount of char formed is up to 30%. The properties of this residue or char vary considerably depending on fuel properties and operating conditions.

It is known that at least two types of solids are formed during the gasification of residual fuel oil (1,2). Soot, consisting of sub-micron carbon particles, is known to form during hydrocarbon cracking reactions. Soot is denser than char, having a density between 1500 kg/m^3 and 2000 kg/m^3 whereas char density is about the same as that of the original fuel.

2.2.4. Combustion of Gases.

Previous models developed to describe liquid fuel fired reactors (14,15,23,24) have been based on the assumption that the fuel entered the combustion zone as a vapour and that reaction rate was kinetically controlled. The combustion of kerosene (15,23) and iso-octane (14,24) has

been analysed using collision theory of gases.

The volumetric reaction rate is

$$\dot{r} = FZ \exp \left(-\frac{E_a}{RT} \right) \quad \frac{\text{kmole}}{\text{m}^3 \text{ s}} \quad 2.22$$

where Z is the total number of molecular collisions, and F is a probability factor indicating the fraction of collisions that result in reaction. For a second order bimolecular reaction

$$Z \propto x_a x_b \left(\frac{P}{RT} \right)^2 T^{0.5} \quad 2.23$$

where x_a and x_b are the mole fractions of the reacting species. Equations 2.22 and 2.23 are combined to give

$$\dot{r} = K x_a x_b T^{0.5} \exp \left(-\frac{E_a}{RT} \right) \left(\frac{P}{RT} \right)^2 \quad 2.24$$

where K is the rate constant. For reactions other than second order, viz reaction order $\alpha + \beta$

$$\dot{r} = K x_a^\alpha x_b^\beta T^{0.5} \exp \left(-\frac{E_a}{RT} \right) \left(\frac{P}{RT} \right)^{\alpha+\beta} \quad 2.25$$

The constants α , β , E_a and K have been determined experimentally (14) for high intensity combustors burning single component gaseous fuels where the mixing characteristics were known.

Combustion of multi-gas fuels is more difficult to model using collision theory because the number of constants that must be evaluated is increased, and whereas E_a is approximately constant for a single component fuel, it varies with composition for multi-component fuels. In industrial furnaces where multi-gas fuels are common, it is not usually possible to determine the constants required for a model based on collision theory. A more promising approach is to consider the reaction rate of each gas individually.

Product gases from high intensity gasifiers are known to contain methane, carbon monoxide and hydrogen (2,3). Little information is available for the $\text{CH}_4 - \text{CO} - \text{H}_2 - \text{H}_2\text{O}$ reaction system although some flame speed measurements have been made for the $\text{CO} - \text{H}_2 - \text{H}_2\text{O}$ system (25). It is known (26) that carbon monoxide and water vapour are the primary products of methane combustion, carbon monoxide being subsequently oxidised to carbon dioxide. KOZLOV (25) found that the reaction rate of methane in the presence of water vapour is of the form

$$\dot{r}_{\text{CH}_4} = k_8 x_{\text{CH}_4}^{\alpha} x_{\text{O}_2}^{\beta} \exp\left(\frac{-E_a}{RT}\right) \left(\frac{P}{RT}\right)^{\alpha+\beta} T^{-1} \quad 2.26$$

where $k_8 = 7 \times 10^{11} \frac{\text{kmole}}{\text{m}^3 \text{ s}}$, $E_a = 2.51 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$,

$\alpha = -0.5$ and $\beta = 1.5$ for $1200 \text{ K} < T < 1400 \text{ K}$.

The reaction rate of carbon monoxide is

$$\dot{r}_{\text{CO}} = k_9 x_{\text{CO}}^{\alpha} x_{\text{O}_2}^{\beta} x_{\text{H}_2\text{O}}^{\delta} \exp\left(\frac{-E_a}{RT}\right) \left(\frac{P}{RT}\right)^{\alpha+\beta+\delta} T^{-2.5} \quad 2.27$$

where $k_9 = 1.04 \times 10^{15} \frac{\text{kmole}}{\text{m}^3 \text{ s}}$, $\alpha = 1.0$, $\beta = 0.25$ and $\delta = 0.5$. The activation energy E_a is temperature dependent, increasing from $1.34 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$ at 1300 K to $1.46 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$ at 900 K . FREEDMAN and CYPHERS (27) studied a similar system and found $E_a = 1.17 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$ in the range $1100 \text{ K} < T < 1600 \text{ K}$.

The combustion of hydrogen has been studied extensively (28) and is known to follow kinetics similar to carbon monoxide oxidation (29). No published reaction rate expressions for high temperature H_2 - air systems have been found that do not contain free radical terms unmeasurable by conventional gas chromatography. From flame speed

measurements (30), it is known that the rate constant for the oxidation of hydrogen is approximately three times as large as that for the oxidation of carbon monoxide, whilst the activation energies are comparable.

Experimental measurements on combustion chambers have shown that the conversion of gaseous combustibles may be limited by thermodynamic equilibrium conditions. Although BATCHELDER et al (31) predicted that the homogeneous water gas shift reaction should reach equilibrium in 5 ms during the gasification of pulverised coal, experimental measurements (14) on an iso-octane combustor indicated that equilibrium was not attained in 15 ms. NARASIMHAN and FOSTER (32) found from experimental evidence that this equilibrium was achieved within 40 ms during methane combustion, and more recently it has been found that equilibrium is attained in 60 ms with respect to temperature in both stages of the system studied in this work.

2.2.5. Combustion of Solids.

The combustion of pulverised fuels has been investigated fully (13,35,36,37,38). BATCHELDER et al (31) have reviewed the work completed before 1953, and have concluded that the high temperature of gasification of coal and char may be achieved by the use of air, steam or carbon dioxide. The reaction rate may be controlled by gaseous diffusion or surface kinetics depending on particle size, temperature and velocity relative to the gas stream.

MAYERS (39) studied spherical carbon particles with diameters in excess of 200 μm , and found that the burning rate $\dot{m}$ as expressed by equation 2.28 was dependent

on particle/gas relative velocity.

$$\dot{m} = \frac{20 M_c D p_{O_2}}{D_p RT} \quad \frac{\text{kg}}{\text{m}^2 \text{ s}} \quad 2.28$$

where M_c = molecular weight of carbon, D = molecular diffusion coefficient, p_{O_2} = partial pressure of oxygen in the bulk gas, D_p = particle diameter, and R = gas constant.

The specific surface reaction rate was diffusionally controlled.

BATCHELDER et al (31) derived a similar expression for large stationary carbon spheres, viz

$$\dot{m} = \frac{1.67 M_c D p_{O_2}}{D_p RT} \quad \frac{\text{kg}}{\text{m}^2 \text{ s}} \quad 2.29$$

Equations 2.28 and 2.29 are shown in Fig.2.2 as curves 1 and 2 respectively. They show linear dependence on $T^{3/2}$ which is a characteristic of diffusional control, but are in poor agreement with experimental measurements made on particles less than 100 μm diameter.

The burnout of carbon particles in the range 1 μm - 30 μm has been studied by BEER and LEE (14). Their results were interpreted in terms of a kinetically controlled model and expressed as a semi-empirical equation, 2.30, shown as curve 3 in Fig.2-2.

$$\dot{m} = k_{10} p_{O_2} \exp\left(\frac{-E_a}{RT}\right) T^{-0.5} \quad \frac{\text{kg}}{\text{m}^2 \text{ s}} \quad 2.30$$

where $k_{10} = 1190$ and $E_a = 0.79 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$.

LEE et al (38) derived an expression similar to equation 2.30 for a different set of experimental results, where $k_{10} = 1.085 \times 10^5$ and $E_a = 1.64 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$. This new equation appears as curve 4 in Fig.2-2.

MARSDEN (36) analysed soot burning results using

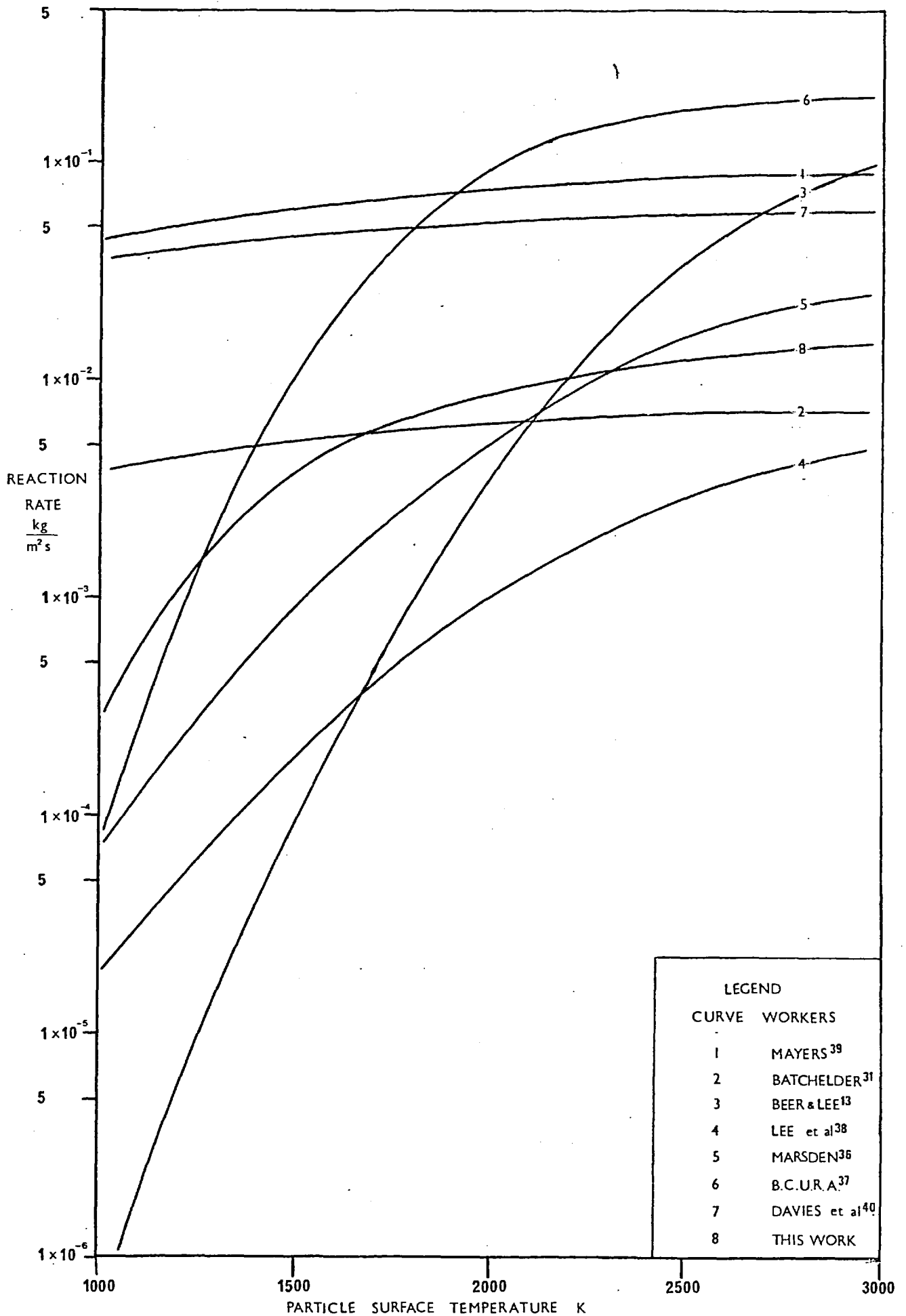


FIG.2-2 COMPARISON OF CARBON BURNING MODELS FOR
A 100 μm PARTICLE

a semi-empirical equation of the form

$$\dot{m} = k_{11} p_{O_2}^{1.23} \rho_p D_p \exp\left(\frac{-E_a}{RT}\right) T^{-0.5} \frac{\text{kg}}{\text{m}^2 \text{s}} \quad 2.31$$

as shown by curve 5 in Fig.2-2, where ρ_p is particle density, $k_{11} = 7.24 \times 10^6$ and $E_a = 0.82 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$. It can be seen that this rate expression approaches zero as D_p decreases, hence the burning rate of very small particles is underestimated.

B.C.U.R.A. (37) has interpreted the combustion of pulverised coal particles greater than 10 μm diameter by using a particulate burning theory combining gaseous diffusion through a laminar boundary layer with a semi-empirical surface reaction rate expression. This model shows good agreement with experimental results obtained by burning coal particles 100 μm to 10 μm diameter, but does not apply to sub-micron particles. The model predicts changes from diffusion to kinetic control and is shown as curve 6 in Fig. 2-2.

Recent work by DAVIES et al (40) for coal particle combustion using a combined diffusion-kinetic controlled model is shown in Fig.2-2 as curve 7 and is similar to previous diffusional controlled models. The authors' claim, that the diffusional term in the model may be ignored, leads to the conclusion that their choice of parameters is such that the kinetically controlled and diffusionally controlled models are indistinguishable.

The main assumptions made in the formulation of these models are as follows:

1. $N_{Nu} = 2$.
2. The particles are spherical and of constant density.
3. The particle diameter decreases with time.

4. The oxygen-carbon reactions are the only ones occurring.

The theory of single particle combustion has been applied to the combustion of clouds of particles. HEDLEY and GULDENPFENNIN (35) have considered a cloud of spherical particles of non-uniform size where the size distribution was defined by

$$G_w = - \frac{d\bar{w}}{dD} \quad 2.32$$

The weight fraction of particles larger than diameter D is $\bar{w}$ defined by the Rosin-Rammler equation

$$\bar{w} = \exp -k_{12} D^y \quad 2.33$$

where y is a constant, hence

$$G_w = y k_{12} D^{y-1} \exp -k_{12} D^y \quad 2.34$$

which is similar to equation 2.1. For the purpose of calculation, the cloud was considered to consist of a finite number of monosize fractions. It has been found (37) that for coal particles of the size range 1 to 300 μm , the combustion model is insensitive to the number of size increments considered if each contains the same weight fraction. DAVIES et al (40) have used constant size increments of 20 μm for the size range 1 to 200 μm . In both these analyses, the arithmetic mean diameter of each increment has been used in calculations of particle burnout. The use of this diameter can result in errors of up to 30% in the burning rate as shown in Appendix 5.

2.2.6. Thermal Stability.

One of the objectives in the design of high intensity combustion equipment is to minimise combustion

chamber volume. This minimum is limited by evaporation rate in spray combustors and by chemical kinetics or mixing in gas combustors.

The evaporation rate in spray combustors has been determined from spray characteristics, operating conditions and chamber aerodynamics (34), however the thermal stability characteristics have not been reported.

Several investigators have studied kinetically limited gas combustors burning pure component fuels (14,15, 41). The analyses have been based on the collision theory of gases using equation 2.22. In order to evaluate the constants in the expression, the systems have been designed to approach perfect micromixing, however no account has been taken of departures from this condition. If the fuel and air are premixed, the rate of oxygen consumption per unit volume for the fuel rich case (14) is

$$\dot{r}_{O_2}^* = \frac{0.21 \dot{M}_a \epsilon_{O_2}}{V} \quad 2.35$$

where ϵ_{O_2} is the oxygen consumption efficiency, defined as the oxygen consumed divided by the oxygen consumed if combustion were complete, $\dot{M}_a$ is the mass flow rate of air, and V is chamber volume.

For the air rich case

$$\dot{r}_{O_2}^* = \frac{0.21 \dot{M}_a \epsilon_{O_2} \phi}{V} \quad 2.36$$

By equating 2.25 and 2.35

$$\frac{\dot{M}_a}{V P^{\alpha+\beta}} = \frac{K}{0.21 \epsilon_{O_2}} x_{O_2}^{\alpha} x_{fuel}^{\beta} T^{0.5} \exp\left(\frac{-E_a}{RT}\right) \left(\frac{1}{RT}\right)^{\alpha+\beta} \quad 2.37$$

LONGWELL and WEISS (14) assumed that $E_a = 1.75 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$ and that the overall reaction order was 1.8 for the

combustion of iso-octane in air. Using these initial assumptions, K and β were adjusted to give the best fit to experimental results. The experimental data consisted of blowout measurements obtained by increasing the air loading for a fixed equivalence ratio until the flame in the chamber was extinguished. Once again it was assumed that the chamber was perfectly micromixed in all cases. The effects of any departure from this condition would be hidden by the choice of K and β .

HERBERT (15) studied the effects of inlet temperature and chamber pressure on the combustion stability in a baffle stabilised kerosene combustor, and concluded that stability was increased by increasing inlet temperature and chamber pressure. He also indicated that combustion stability would be decreased by the addition of water vapour. This conclusion was based on the assumption that water vapour did not take part in the combustion reaction. In a baffle stabilised chamber it has been found (15) that

$$\frac{\dot{M}_a}{V P^{\alpha+\beta}} \propto N_{Re}^{k_{13}} \quad 2.38$$

where $k_{13} > 0$, hence both inlet conditions and chamber geometry affect flame stability. In these analyses, mixing effects have been neglected and stability found to vary with inlet temperature, equivalence ratio and air loading as shown in Fig.2-3. The curves indicate the maximum air loading for a given inlet temperature and equivalence ratio. If this maximum loading is exceeded, the flame will blowout.

Little work has been done concerning combustion stability in imperfectly stirred reactors. EVANGELISTA et al (42) have indicated how departures from instantaneous

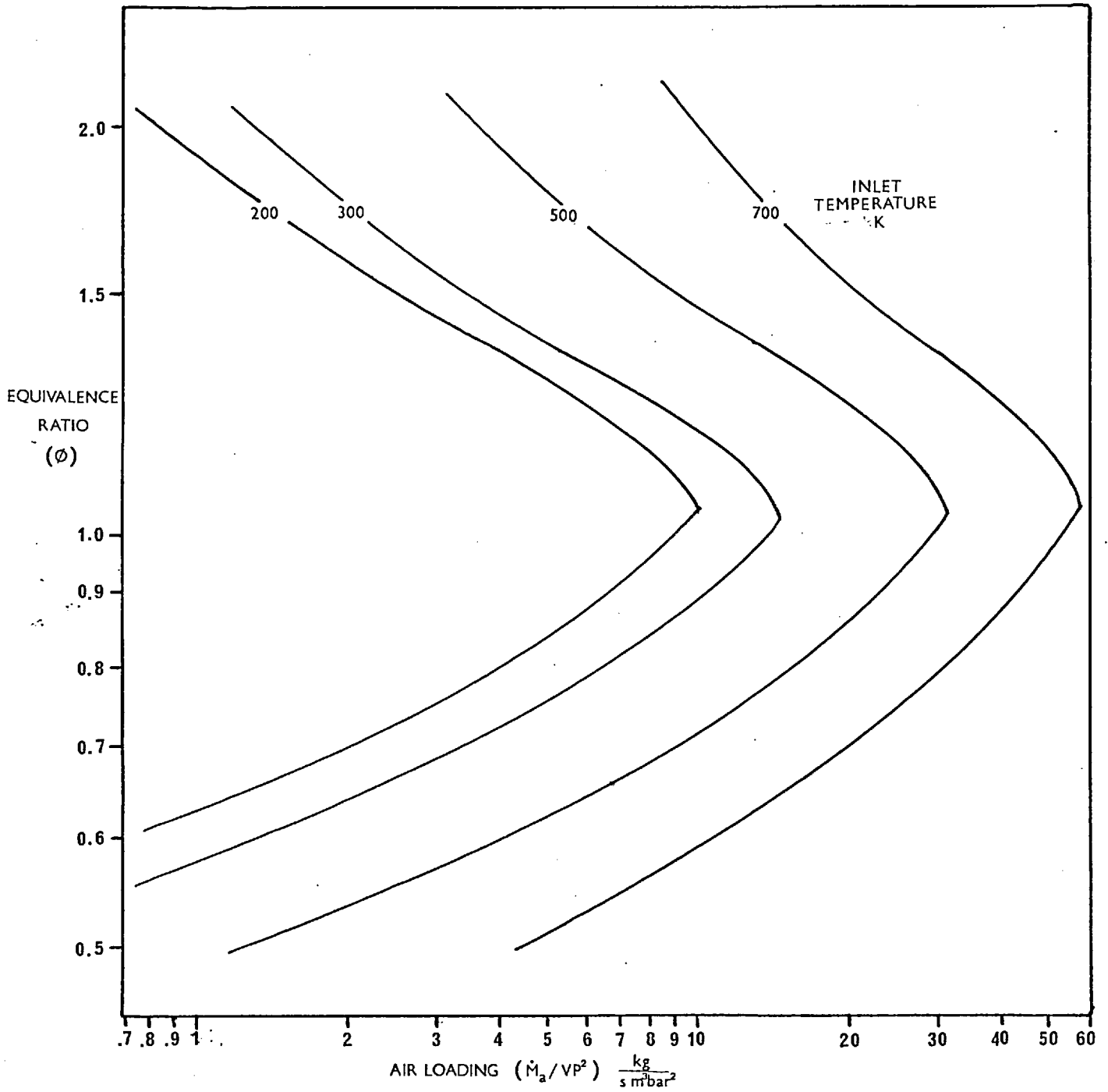


FIG.2-3 THE EFFECT OF INLET TEMPERATURE ON COMBUSTION STABILITY

micromixing can be measured, and what effects these deviations have on reactor stability. The effect of mixing intensity on reaction rate and hence on the blowout limit has been determined using a coalescence model based on Ficks law of diffusion. By defining a turbulent mixing rate,

$$\beta_T = \frac{l_m^2}{\tau} \quad 2.39$$

where l_m is the microscale for turbulent mixing, and τ the mean residence time of the reactor,

$$\tau = \frac{\pi L^3}{6 A_i v} \quad 2.40$$

where L is the equivalent spherical diameter of the chamber, and A_i is the cross sectional area of the inlet ports,

the mixing conditions have been related to the inlet jet design parameter $\frac{L^2}{A_i}$ viz

$$\beta_T \tau = \frac{1}{4} \left(\frac{L^2}{A_i} \right)^{0.67} \quad 2.41$$

Use of this analysis indicated that in high intensity combustors reported in the literature $\frac{L^2}{A_i}$ varied from 30 to 3000 and $\beta_T \tau$ varied from 2.5 to 50. As $\beta_T \tau$ approached infinity, the reactor approached perfect mixing. For first and second order kinetics, it has been shown that an increase in $\beta_T \tau$ increased the blowout limit, hence an increase in inlet velocity or a decrease in mean residence time will decrease combustion stability.

2.2.7. Heat Transfer.

Heat transfer in furnaces is achieved by a combination of convection and radiation with radiation predominating in the region near the flame zone. Standard expressions based on the Nusselt number are available for the

determination of convective heat transfer coefficients h_c in turbulent flow systems. Heat transfer measurements have been made on the system studied (3,11) using a heat exchange section between the first and second stages. Both workers interpreted the observed heat transfer rates in terms of combined radiation and convection. JOHNS (11)

characterised convective heat transfer by the coefficient h_c

$$h_c = \frac{0.02 k}{D_T} \left(\frac{v D_T}{\nu} \right)^{0.8} \quad 2.42$$

where D_T is the characteristic flow diameter in the heat transfer section. This equation corresponds with the Dittus-Boelter and Colburn equations assuming the Prandtl number N_{Pr} equal to 0.74. This assumption is unjustified as N_{Pr} for combustion gases may be as high as 3.0 at 1800 K.

Radiant heat transfer was determined by the difference between the measured heat transfer and the convective heat transfer calculated using equation 2.42. The calculated emissivity was considerably higher than expected from a small diameter furnace and this can be explained in part by the absence of the Prandtl number in the Colburn equation, which would lead to an underestimation of convective heat transfer. It was shown from theoretical arguments (11) that the mean gas temperatures used to characterise heat transfer are T_{rm} for radiation and T_{cm} for convection where

$$T_{rm} = \sqrt{T_{IN} T_{OUT}} \quad 2.43$$

and

$$T_{cm} = \frac{2T_{IN} T_{OUT}}{T_{IN} + T_{OUT}} \quad 2.44$$

The inlet and outlet gas temperatures are T_{IN} and T_{OUT} respectively.

Further measurements were made by ARCHER (3) who interpreted the results by calculating the radiant heat transfer from theoretical considerations and the convective term by difference. By assuming that all the gas borne solids were soot, the luminous emissivity ϵ_L was defined as

$$\epsilon_L = 1 - \exp\left(\frac{k_a L_b T}{2055}\right) \quad 2.45$$

where k_a is the absorption coefficient of the soot, and L_b is the radiating length. The temperature T corresponded to the mean radiant temperature over an incremental length in the heat exchange section.

The non-luminous emissivity ϵ_n was calculated as

$$\epsilon_n = 0.414 - 0.0000294 T_{rm} \quad 2.46$$

and the total emissivity was defined as

$$\epsilon_T = 1 - (1 - \epsilon_L)(1 - \epsilon_n) \quad 2.47$$

This method of analysis did not take into account either solid particle size distribution which affects ϵ_L or gas composition which affects ϵ_n , although the author noted that these factors might explain the discrepancies between measured and predicted heat transfer rates.

The effect of particle size on luminous emissivity has been reported by HOTTEL and SAROFIM (10). The emissivity of a cloud of spherical black particles greater than $0.1 \mu\text{m}$ diameter is

$$\epsilon_c = 1 - \exp(-A_p L_b) \quad 2.48$$

where A_p is the projected cross sectional area of the particles per unit volume of gas. The emissivity of a cloud of soot particles with diameters less than $0.1 \mu\text{m}$ is independent of particle size but proportional to the volume fraction of the furnace occupied by the particles f_v .

$$\epsilon_s = 1 - \frac{1}{1 + k_b \frac{f_v}{L_b T}} \quad 2.49$$

The constant k_b has been found to vary from 440 to 980 $m^{-1} K$ for residual fuel oil flames, and depends predominantly on fuel composition.

2.3. System Optimisation.

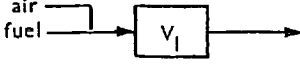
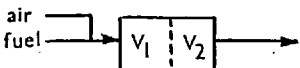
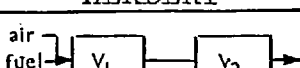
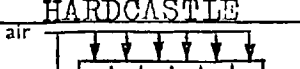
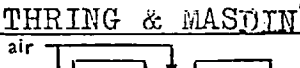
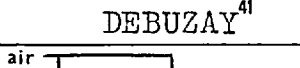
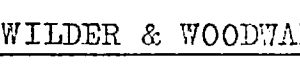
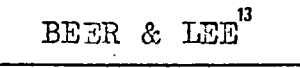
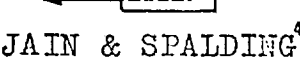
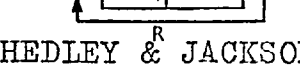
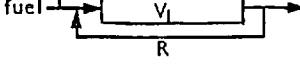
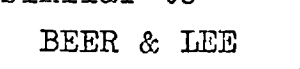
Both experimental and theoretical studies have been made of single stage combustion systems in an attempt to maximise or minimise some feature of the system (14,15). Two stage (47) and multistage systems (23) have been analysed theoretically, but they have not been tested experimentally. In general, the objective has been to minimise combustion chamber volume, or to maximise reaction rate within the limits of overall air rich or stoichiometric combustion. The types of systems that have been studied are summarised in Table 2-1.

2.3.1. Single Stage Systems.

Homogeneous adiabatic systems have been analysed using kinetically controlled models without mixing limitations (14,15). It is known that flame stability is increased by the use of preheat, and that the maximum air loading in a system is achieved by stoichiometric operation as shown in Fig.2-3. In both cases, the fuel and oxygen consumption efficiencies were assumed to be equal. Combustion inefficiency was represented by unreacted fuel in the outlet stream, although this assumption was not confirmed by gas analysis, which indicated the complete absence of unburnt fuel. Combustion efficiency decreased as the blowout condition was approached, and reached an

TABLE 2-1.

COMPARISON OF COMBUSTION MODELS

FLOW SYSTEM	THEORETICAL SYSTEM	FUEL	REACTION MODEL
 LONGWELL & WEISS ¹⁴	homogeneous perfectly stirred V micromixed	propane iso-octane	EQUATION 2-22
 BRAGG ²³	homogeneous perfectly stirred V micromixed internal recycle	kerosene	EQUATION 2-22
 HERBERT ¹⁵	homogeneous perfectly stirred V micromixed	kerosene	EQUATION 2-22
 HARDCASTLE ²⁴	homogeneous perfectly stirred V micromixed	iso-octane	EQUATION 2-22
 THRING & MASDIN ⁴⁸	homogeneous perfectly stirred V micromixed	kerosene	EQUATION 2-22
 DEBUZAY ⁴¹	homogeneous perfectly stirred V micromixed	propane	EQUATION 2-22
 WILDER & WOODWARD ⁴⁷	homogeneous perfectly stirred V micromixed	propane	EQUATION 2-22
 BEER & LEE ¹³	heterogeneous perfectly stirred V ₁ plug flow V ₂ segregated-micromixed	pulverised coal	EQUATION 2-30
 JAIN & SPALDING ⁴⁹	homogeneous imperfectly stirred internal recycle	NOT SPECIFIED	$\dot{r} = \frac{(\alpha + \beta)^{\alpha + \beta}}{\alpha^{\alpha} \beta^{\beta}} (1 - \tau)^{\alpha} \tau^{\beta}$
 HEDLEY & JACKSON ⁴⁴	homogeneous plug flow V micromixed external recycle	ethylene	EQUATION 2-22
 HEDLEY & JACKSON ⁴⁵	heterogeneous plug flow V micromixed external recycle	pulverised coal	similar to EQUATION 2-31
similar to BEER & LEE DRAKE & HUBBARD ⁴³	heterogeneous perfectly stirred plug flow	residual fuel oil	EXPERIMENTAL
 EVANGELISTA et al ⁴²	homogeneous imperfectly stirred segregated	NOT SPECIFIED	$\dot{r} = k x^{\alpha} (1 - x)^{\beta}$
 HOPKINS ³⁴	heterogeneous perfectly stirred V micromixed internal recycle	residual fuel oil	EQUATION 2-19

overall minimum near stoichiometric operation. Combustion efficiency at blowout decreased as the inlet temperature increased.

Heterogeneous systems using pulverised coal or oil fuels have been studied under adiabatic conditions. BEER and LEE (13) have compared perfectly stirred and plug flow combustors burning pulverised fuel and have shown that in a perfectly stirred combustor, the reaction rate as given by equation 2.30 exceeds that in a plug flow combustor until a conversion of about 55% is reached. After this the reaction rate in the plug flow section is greater. The model was considerably simplified by the assumption that all fuel particles had an initial diameter of $30\text{ }\mu\text{m}$.

DRAKE and HUBBARD (43) studied the aerodynamics in a combustion chamber burning heavy fuel oil. The object was to minimise unburnt solids or maximise combustion efficiency at low levels of excess air by varying the degree of swirl imparted to the combustion air at the atomiser. This work showed that solid emission could be controlled by the correct positioning of the swirl vanes in the atomiser, thereby changing the flow patterns i.e. internal recycle, in the chamber. Using a tracer technique, they found that when the furnace was cold, the fraction of the chamber that appeared to be perfectly mixed varied as the swirl vane angle, and reached a minimum of 0.64 when the solid emission was lowest. The lowest solid emission determined by BEER and LEE (13) was achieved with a perfectly stirred fraction of 0.15. The difference in these findings is due to the non-uniform particle size distribution used by DRAKE and HUBBARD, which contained considerable numbers of particles in excess of $150\text{ }\mu\text{m}$.

HEDLEY and JACKSON (44,45) analysed homogeneous and heterogeneous reaction systems assuming external recycle of fully reacted combustion products. These products were assumed to be removed from the exit stream and mixed instantaneously with the incoming reactants, which then reacted in a plug flow reactor. They concluded that the reaction rate could be increased by the use of increased external recirculation, however the improvement decreased as high levels of recirculation were approached. In industrial furnaces where recirculation is achieved internally by aerodynamic design of the combustion chamber, increasing recirculation of inert products is known to decrease combustion efficiency (19). The apparent ambiguity of these results is explained by the fact that as recycle increases, internally recirculated products occupy an increasing amount of the available combustion space, whereas externally recirculated products do not.

The effects of imperfect mixing on homogeneous adiabatic combustion systems have been investigated (42) and it is indicated that for practical systems, perfect mixing is never achieved. HOW (46) found that mixing limitations affected heterogeneous systems in a similar way. These results indicate that where reaction rate studies are concerned, some account must be taken of the degree of imperfect mixing occurring in the furnace.

The operation of a heterogeneous single stage system with internal recirculation has been studied extensively (2,11,19). HTUN (2) found that with air atomisation, combustion intensity was highest at stoichiometric conditions with solid emission decreasing as

ϕ decreased to 1.0. Steam atomisation reduced

solid loading. HOPKINS (19) applied mixing models and reaction rate theory to the evaporation and combustion of a fuel spray and has shown that evaporation efficiency is dependent on the level of internal recycle in the chamber. He did not, however, attempt to optimise the operation of the chamber. JOHNS (11) found that combustion gas emissivity increased with equivalence ratio, but that the maximum heat transfer coefficient in a 76 mm diameter cooling tube was achieved at $\phi = 1.15$. This result indicated the opposing effects of increasing solids and decreasing temperature on heat transfer.

2.3.2. Multi-stage Systems.

The theoretical aspects of multi-stage combustion systems with and without recycle have been extensively analysed. Most analyses have dealt with two stage systems operating with excess air overall. In all cases, the reaction models were kinetically controlled, and where liquid fuels were used, the fuel was vapourised before entry into the combustion zone.

DE ZUBAY (40) studied the combustion of propane in a two stage system where all the fuel entered the first chamber. The maximum heat release was limited by dissociation of the fuel and the dissociated fuel was assumed to be unburnt. Excess air was used throughout, and it was found that for an overall equivalence ratio of 0.5, the highest overall intensity was achieved with two chambers of equal volume and the first stage operating stoichiometrically. WILDER and WOODWARD (47) subsequently found that for a two

stage system of fixed volume, where all the fuel entered the first stage, but where dissociated fuel could be burned, the maximum intensity resulted at stoichiometric operation when the second stage volume was equal to zero. For the purposes of calculation, the inlet temperatures to each stage were assumed to be equal. This requirement would place a serious restriction on a practical system as excessive interstage cooling would be required.

Staged fuel and air injection was considered by HARDCASTLE (23) for air rich and stoichiometric conditions. This analysis, based on data obtained by LONGWELL and WEISS (14), also indicated that maximum intensity was achieved when all the fuel was injected into the first stage at an equivalence ratio of unity. As in previous work, the only combustible leaving each stage was assumed to be unreacted fuel. This assumption limits the use of this type of analysis to practical systems where the efficiency in each stage is virtually 100% or where the unburnt fuel remains unchanged as in soot combustion.

Several investigators have considered homogeneous multi-stage systems with recycle (22,48,49). BRAGG (22) considered the combustion of aviation kerosene at an overall equivalence ratio of 0.86, with recirculation of hot partially burned reactants to the inlet section achieved by aerodynamic design of the chamber. The reaction model was kinetically controlled and based on collision theory. It was found that the "optimum burning mixture" or maximum intensity was achieved at a temperature equal to 80% of the adiabatic flame temperature and at a fuel to air ratio of twice stoichiometric in the first stage. In the second stage, the maximum

intensity was achieved at 90% of the adiabatic flame temperature and at $\phi_2 = 0.86$ corresponding to 16% excess air. It was noted however, that to achieve combustion efficiencies approaching 100%, excessively large second stage chambers would be required.

JAIN and SPALDING (49) studied a reactor with the inlet and outlet at the same end using a model including heat and mass transfer between the forward and reverse streams. This work showed that for the analysis of reaction kinetics, the reactor should be operated near the blowout point.

A complex combustion system consisting of six sections of equal volume in series with recycle from last to first was analysed by THRING and MASDIN (48). All the fuel was added in stage 1, but air was added sequentially to maintain the maximum burning rate in each stage. With a recirculation ratio of 4:1 based on inlet fuel and air to the first stage, the minimum system volume to achieve complete conversion resulted when 10% of the stoichiometric air was added to stage 1, 20% in each of stages 2,3,4, and 25% in each of stages 5 and 6. The intermittent stage-wise addition of air to this system would make the system too complex for large scale applications.

The performance of a two stage residual fuel fired system with interstage heat removal, operating at $\phi=1$ overall, has been investigated (50). For this constant volume system, the maximum intensity was approached as the fuel flow rate was increased and was limited only by the unburned solids in the exhaust gases. The useful interstage heat removal was maximised as the first stage equivalence ratio decreased towards unity. A more detailed analysis of this

work is presented in Chapters 4 and 5.

2.3.3. Optimising Techniques.

The optimisation problem can be defined as the minimisation or maximisation of a suitable objective function. Problems may be classified as either dynamic or steady state. Dynamic optimisation problems occur in on-line process control where optimal policies must be determined for changes in the process from one state to another. This involves the continuous manipulation of input variables so as to maintain arbitrary state variables within system constraints. This type of optimisation will not be considered here. This study involves the optimisation of design and economic problems and requires a steady state approach to maximise profit or minimise cost.

The search for the optimum condition is made either directly using patterned searches, or indirectly using gradient techniques. The type of approach used depends upon the complexity of the objective function, as well as the number and type of constraints on the system. Direct methods for minimising a univariate function such as the Fibonacci and bisectional searches (51) make use of elimination techniques by which successive steps shrink the region in which the maximum or minimum must lie. Multi-variate problems have been solved using a similar approach by the use of the parallel tangent method (PARTAN) (51). Steepest ascent methods or modified searches are known to be successful where the objective function does not contain sharp ridges or valleys in the vicinity of the extremum (52).

Most direct methods based on patterned searches

in the co-ordinate directions, fail at constraints where discontinuities occur and on sharp ridges where progress along the ridge is very slow. ROSENBROCK (52) has suggested a method to deal with these problems by rotating the co-ordinate axes such that the primary axis corresponds to the direction of steepest ascent. At constraints, a penalty function is used to modify the objective function so that it changes continuously from the last allowable point inside the constraint to some arbitrarily large or small value at the constraint.

Direct search methods are generally the most reliable approach to the solution of extremum problems. These methods are slow, and involve large numbers of objective function evaluations, hence where the objective function is complex or where the number of variables is large, faster techniques are required even if reliability is reduced.

Indirect optimisation methods are highly problem dependent but may be much faster than direct search methods in terms of computational time. In general, these methods are based on the solution of a set of first or second order non-linear partial differential equations which have been constructed to approximate the objective function in the region of interest. These methods require the determination of the first or second derivatives of the objective function with respect to all variables. The method has been developed by FLETCHER and POWELL (53) for unconstrained minimisation problems without discontinuities using only the first derivatives or gradient of the object function.

Constraints generally complicate the problem,

however equality constraints can usually be handled conveniently by combining them with the objective function. Inequality constraints may be either linear or non-linear, however non-linear constraints can be linearised using slack variables and suitably redefining the objective function. MURTAGH and SARGENT (54) have developed a minimisation technique with quadratic convergence, based on the FLETCHER-POWELL (53) method, which is capable of solving multivariate problems with linear and non-linear constraints. This method is highly successful where function derivatives are evaluated analytically. Where numerical techniques are required to evaluate derivatives, the method becomes more problem dependent, and is unreliable where gradients are very small.

Less general methods such as linear and geometric programming have been applied to linear systems and some specialised problems where the system equations can be expressed conveniently in analytic form (55).

2.4. System Implementation and Control.

At present, the bulk of process heating is achieved by relatively low intensity high capital cost furnaces. Only recently have small, high intensity combustion chambers been employed for large scale process heating (56). Very little industrial use has been made of the two stage combustion concept although some development has been done (57) where precise control of the furnace is required.

Overall combustion system design and control on an economic basis has received little attention, however work in the Soviet Union (58) has indicated that the question

is being studied. LOOSEMORE and BRAY (59) have developed a steady state optimisation model for a lead-zinc blast furnace based on polynomial equations, with coefficients that are continually updated by regression analysis of plant data. This type of model is suitable for set point determination for on-line control.

Little work has been published on the on-line control of furnaces, however MAAS (60) has developed an on-line analogue model to control furnace temperature during the charging of a lead-zinc furnace. Some theoretical work has been done recently by ROOTES and MEEKER (61) on methods of analysis of control stability in furnaces, however these control systems have not yet been implemented.

To summarise; it has been indicated that corrosion and pollution control as well as high specific heat release rates are the objectives of successful residual fuel oil combustion. Two stage combustion systems which allow precise control of combustion conditions have shown promise in this respect. The purpose of this study is to correlate the work that has been carried out on a particular system to determine the best operating conditions which satisfy the above criteria and to indicate the economic viability of this system.

CHAPTER 3.

EXPERIMENTAL SYSTEM.

The experimental system studied is shown schematically in Fig. 3-1, and has been described in detail previously (2,3). The system consists of two refractory lined high intensity combustion chambers, separated by a single-pass water cooled heat exchanger. The fuel used in most of the work was a residual fuel oil containing sodium, sulphur and vanadium as impurities. A lighter distillate gas oil was used during start-up, shut-down, and for some mixing studies. The fuel specification is given in Appendix 1.

An Urquhart High Duty Combustion Chamber operating at an internal pressure of 5.4 kN/m^2 with 0.0116 m^3 internal volume was used as the gasifier and is shown in Fig. 3-2. The chamber consisted of two coaxially opposed refractory cones fitted with a medium pressure twin fluid atomiser and having an exit diameter of 76 mm. Either air or superheated steam at 70 kN/m^2 was used as the atomising medium. Combustion air was used to cool the refractory lining and entered the combustion zone through jets in both refractory cones. Air entering the upstream cone was denoted as forward air whilst that entering the downstream cone was reverse air. The flow patterns developed in the chamber were such that the flame was stabilised at the atomiser for a wide range of operating conditions.

Experimental measurements on the first stage were made through a water cooled refractory lined cross piece of

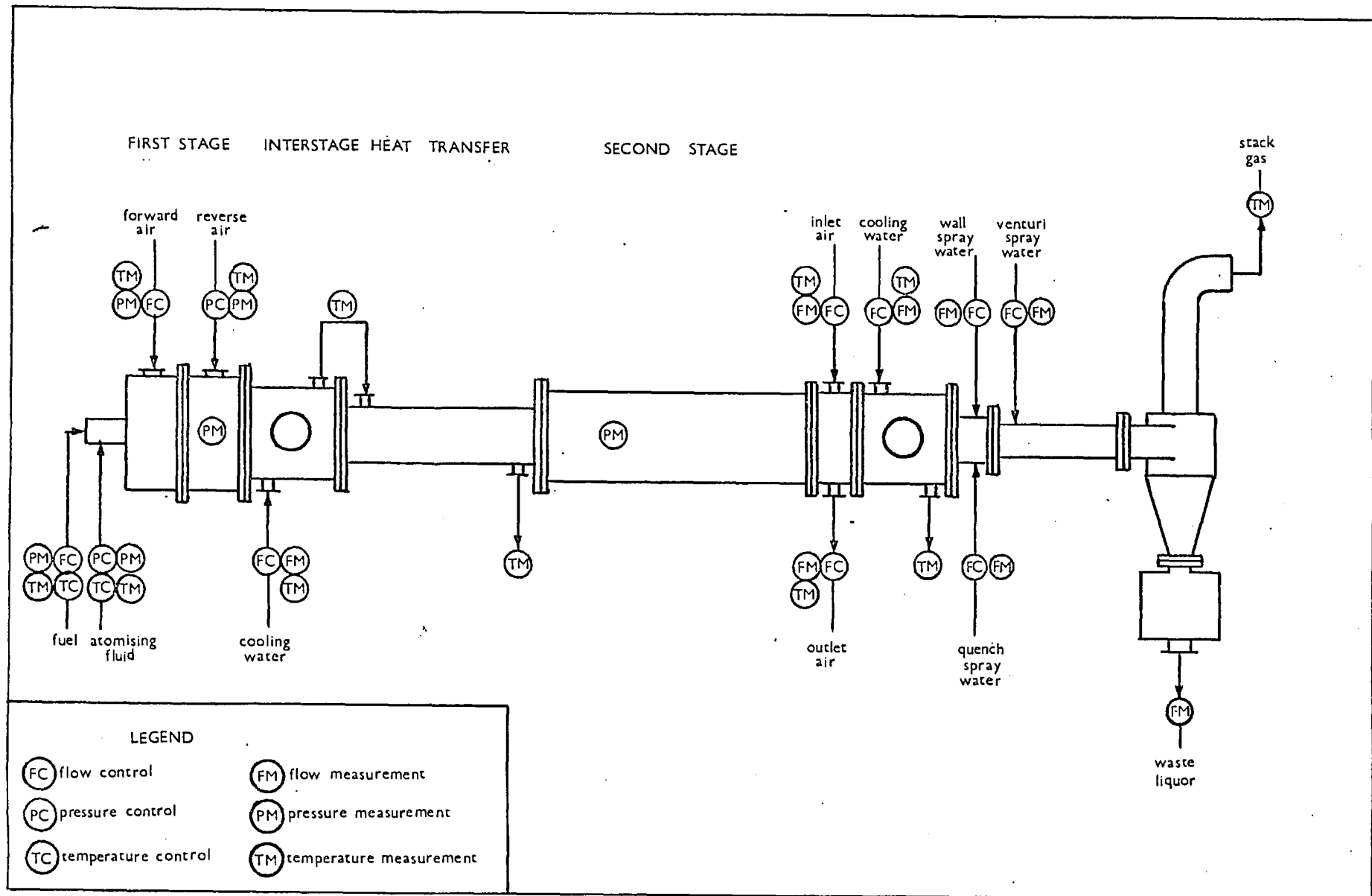
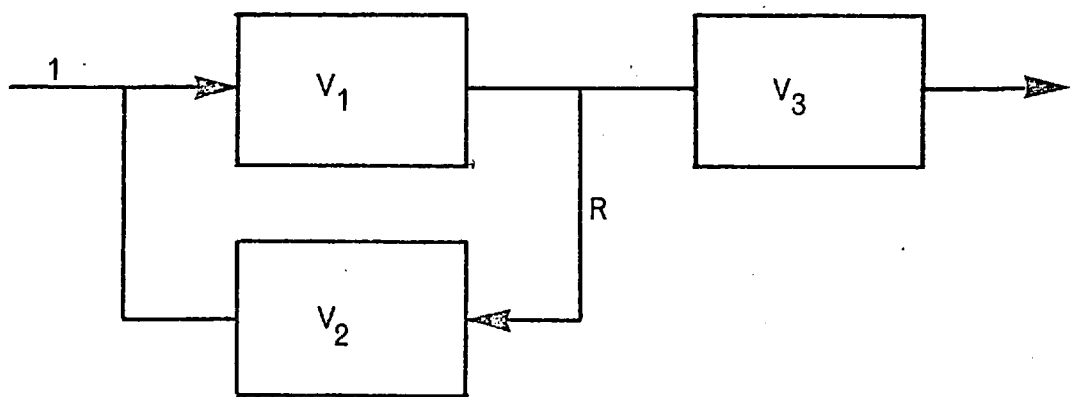
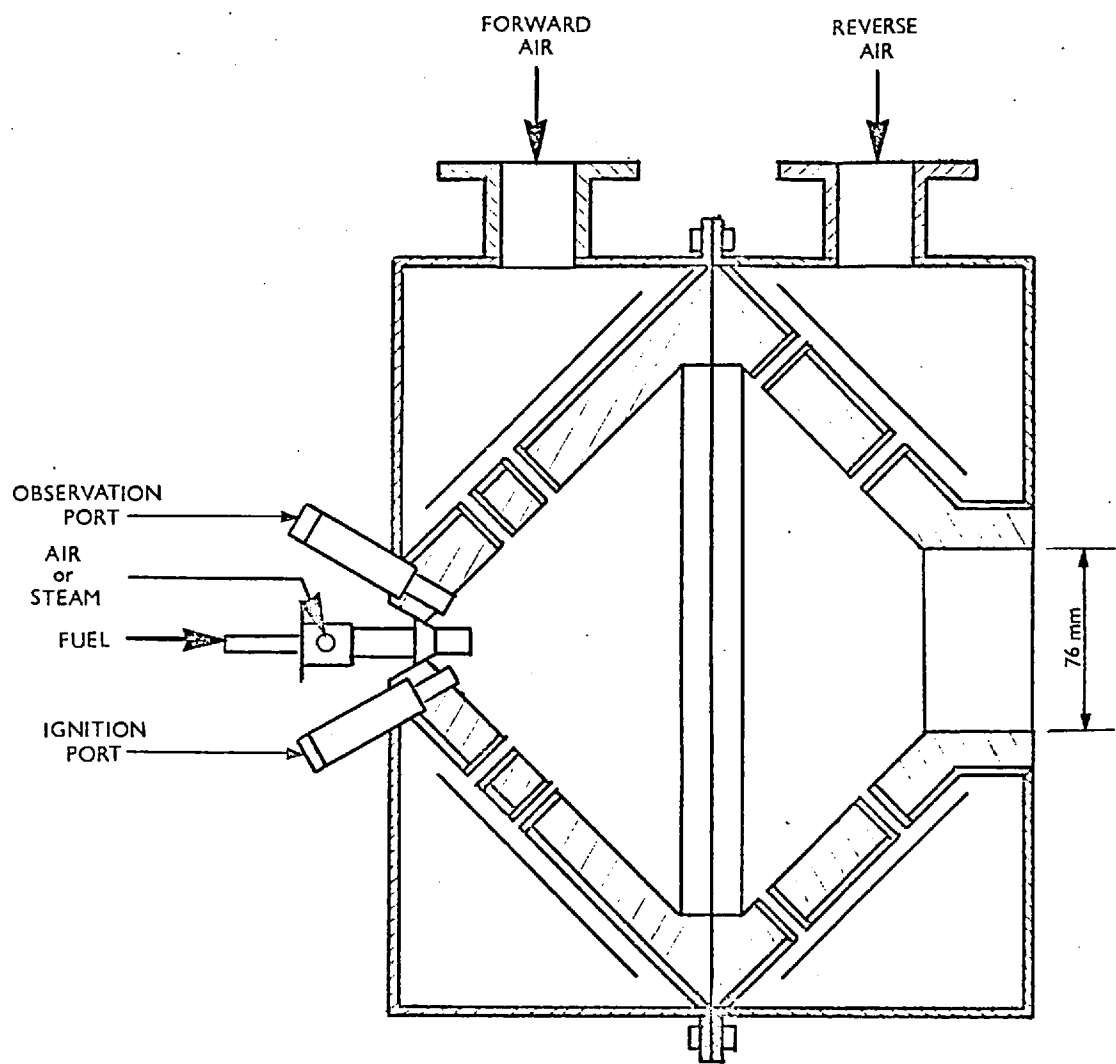


FIG. 3-1 SCHEMATIC OF EXPERIMENTAL SYSTEM



SIMULATED FLOW CONFIGURATION

FIG. 3-2 FIRST STAGE CHAMBER

76 mm^{bore}/situated immediately downstream from the combustion zone. A gate valve was fitted on one arm of the cross piece to allow the insertion of sampling and measuring probes. A collar and face plate were provided on the opposite arm to support the spray cooler probe.

A stainless steel cooling tube, 76 mm diameter and 1.20 m long, was connected in series with the cross piece. Cooling water flowed co-currently with the gas stream from the cross piece.

The second stage chamber, shown in Fig. 3-3, was 1.35 m long with an internal volume of 0.0131 m^3 . The chamber was designed to provide a well mixed section at the inlet, followed by a plug flow section. The refractory walls in the plug flow section were air cooled. Combustion air entered the well mixed section only. The angle of injection was opposed to the inlet flow of combustibles to ensure good mixing.

Experimental measurements on the second stage chamber were made by observation through a sight port in the well mixed section, as well as through a water cooled tee piece fitted to the downstream end of the chamber.

Outlet gases were cooled in a water spray quench section and solids were collected and removed in a venturi-scrubber-cyclone system before the gases were vented to atmosphere.

The flow rates of all process streams were controllable. Inlet fuel and steam temperatures were controllable but inlet water and air temperatures were not.

The experimental measurements made on the system were limited by the configuration of the system and the

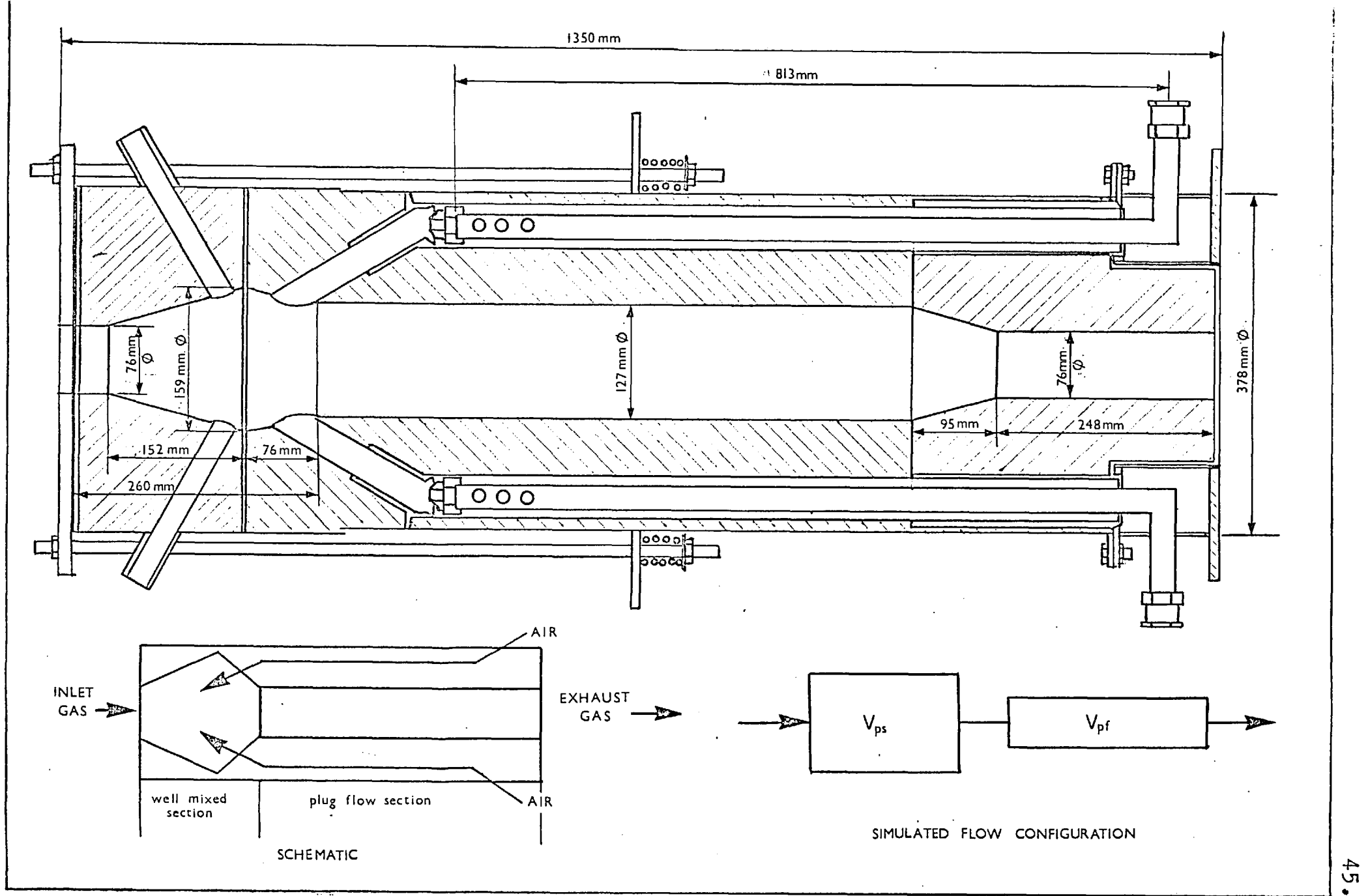


FIG. 3-3 SECOND STAGE CHAMBER

sophistication of the measuring equipment. The location and type of measurements made are shown in Fig. 3-1 and Fig. 3-4.

3.1. Data Available.

Considerable experimental work has been carried out on the system under consideration. The first stage has been studied by HTUN (2) and the effects of equivalence ratio, and air or steam atomisation on the combustion products have been determined. The quantity and composition of the carbonaceous solids formed has been determined in detail with the conclusion that the use of steam atomisation reduces the amount of solid formed compared with the use of air. Component mass balances were not checked, resulting in errors in the reported gas compositions (2) determined by gas chromatography. In particular, the reported values for carbon monoxide and carbon dioxide were too high. Sulphur dioxide and trioxide were measured by a different method and are correct, indicating the absence of sulphur trioxide under reducing conditions.

A similar study has been made with steam atomisation to determine first stage gas compositions and temperature (1). The conclusions drawn from that study were that the outlet gas composition from the first stage chamber did not correspond exactly with the composition expected at water gas shift equilibrium, but that the gas temperature did correspond.

Measured gas compositions from the second stage chamber corresponded with water gas shift equilibrium. Hydrogen mass balance errors were present in the reported

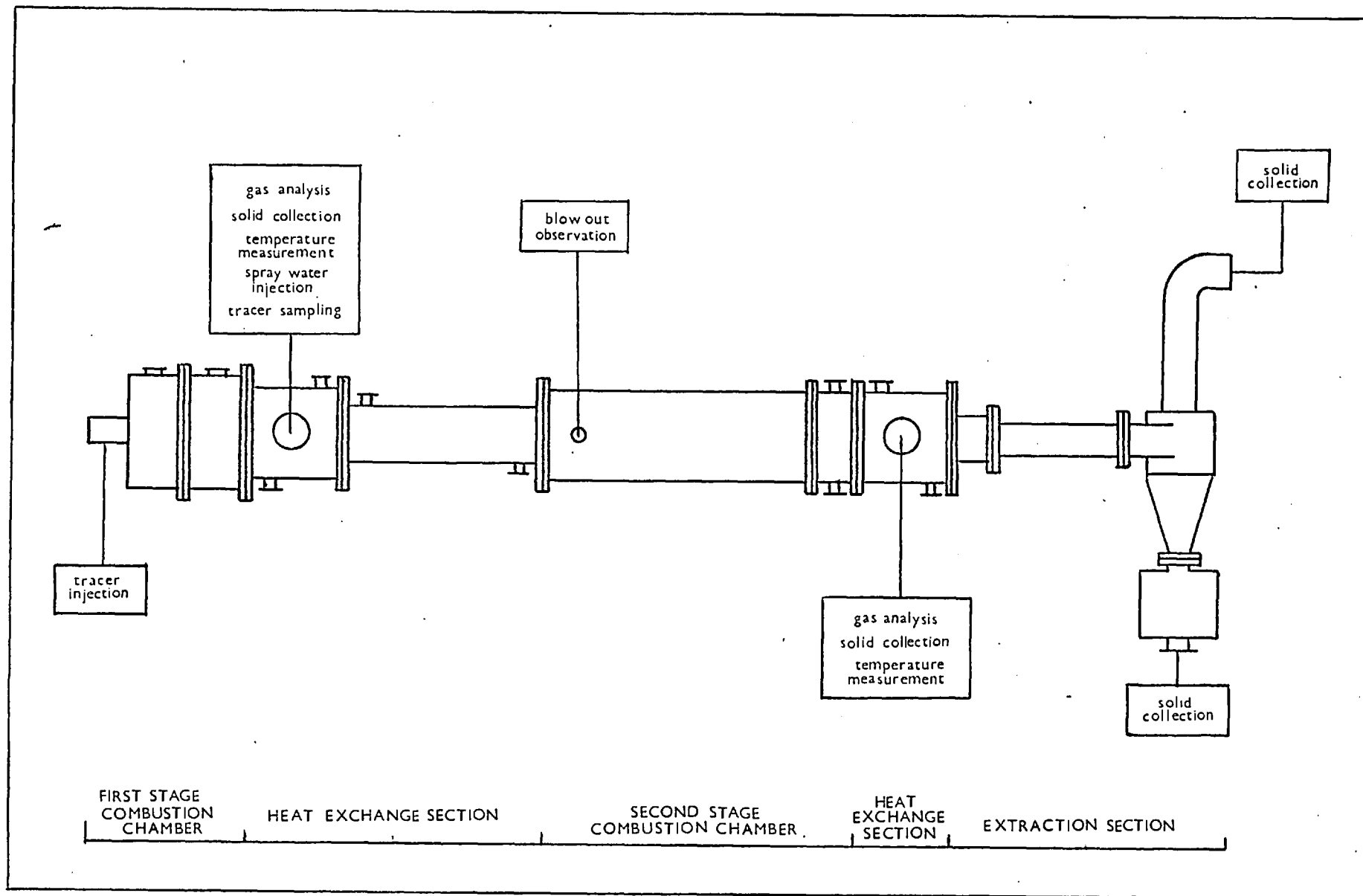


FIG. 3-4 SCHEMATIC OF SYSTEM SUBSECTIONS SHOWING SAMPLING POINTS

results from the first stage, but were absent from the second stage results. This error may explain the apparent departure from water gas shift equilibrium in the first stage.

Solids measurements for steam atomised conditions have been correlated with operating variables (1) to allow the prediction of solids formed in the gasifier, viz.

$$m_{T_1} \pm 0.0022 = 21.6 F_1 + 0.0165 F_2/F_1 + 0.006 (F_2/F_1)^2 - 0.027 \phi_1 F_2/F_1 - 0.0399 \frac{\text{kg solids}}{\text{kg fuel}} \quad 3.1$$

The standard deviation is ± 0.0010 , the error in m_{T_1} being expressed at the 95% confidence interval with 12 degrees of freedom.

When air was used in place of steam as the atomising fluid, the following correlation was obtained

$$m_{T_1} \pm 0.0051 = 0.041 \phi_1^2 - 9.6 F_1 - 0.0074 \frac{\text{kg solids}}{\text{kg fuel}} \quad 3.2$$

In this case, the standard error was ± 0.0023 with 10 degrees of freedom.

Residence time studies on the first stage chamber have been made using both residual and gas oils (34) to determine the relative effects of equivalence ratio, steam/fuel ratio, and forward/reverse air ratio on the mixing and conversion in the chamber. Using a Laplace Transform analysis (19) of the input and output tracer decay curves, the transfer function has been defined as

$$f(s) = \frac{c_1(t) \exp(-st) dt}{c_0(t) \exp(-st) dt} \quad 3.3$$

where $c_1(t)$ and $c_0(t)$ are the normalised outlet and inlet tracer concentrations and s is the Laplace operator. The transfer function has been related to chamber operating conditions, viz.

$$f(s) \pm 0.036 = 0.158 + (13.3 F_4 - 2.88 F_3) \phi_1 + 36.4 F_2 \quad 3.4$$

where s is chosen such that

$$\gamma_s = 1 \quad 3.5$$

The forward and reverse air flow rates, F_3 and F_4 respectively, are in kg/s. The standard deviation of $f(s)$ is ± 0.0175 , the error being expressed at the 95% confidence interval with 26 degrees of freedom.

These results have indicated that a flow system as shown in Fig. 3-2 would have to be considered with recycle R and perfectly stirred volumes V_1 , V_2 and V_3 as parameters. HOPKINS (34) found that in order to equate the experimental values of $f(s)$ with those predicted from theory, the parameters took values in the ranges

$$-0.4 \leq R \leq 1 \quad 3.6$$

$$0 \leq \frac{V_2}{V_1} \leq 2 \quad 3.7$$

$$1 \leq \frac{V_1+V_2}{V_3} \leq 6 \quad 3.8$$

Where R was negative, the system was treated as perfectly mixed reactors in series.

Heat transfer measurements have been made (3,11) using the heat exchange section, however previous interpretations of these results based on combined convection and radiation are unsatisfactory as noted in Section 2.2.7.

The performance of the second stage chamber has been reported previously (50). The conclusions drawn from the analysis of composition and temperature measurements were that water gas shift equilibrium was attained and that appreciable quantities of carbonaceous solids formed in the first stage could be burned in the second stage chamber.

A correlation between operating conditions and second stage solids has been made. viz.

$$m_{T_2} \pm 0.0048 = 17.8F_2 - 0.0145 \phi_1 - 8.60F_1 - 0.0697 F_2/F_1 + 0.0688 \frac{\text{kg solids}}{\text{kg fuel}} \quad 3.9$$

The standard deviation is ± 0.0023 , the error in m_{T_2} being expressed at the 95% confidence interval with 19 degrees of freedom.

3.2. Data Required.

The experimental measurements required to build a comprehensive model of the combustion system can be more easily determined by considering the overall system as a group of simpler sub-sections as indicated in Fig. 3-4. This simplification allows the system to be treated as five essentially independent sections without recycle or by-pass streams.

The first section includes the first stage chamber with twin fluid atomiser, but excludes the sampling cross piece. The cross piece and cooling tube are considered together as the heat transfer section. The second stage chamber and the tee piece are considered individually as sections three and four, whilst the venturi-scrubber and cyclone are grouped together in the fifth section.

System data include plant dimensions and specifications required to calculate the internal volumes and flow velocities. Physical properties as well as thermodynamic data are required for all products and reactants.

Operating data include all flow rates and temperatures including an indication of the error associated with each reading. These errors have been reported (2,3).

In order to obtain a complete description of the system, flow patterns and operating limits are required. In particular, the atomisation characteristics of the twin fluid atomiser, the efficiency of combustion in both chambers as a function of operating conditions, and the thermal stability or blowout limits for each chamber are required. Combustion efficiency can be determined by analysis of the outlet streams with respect to temperature and composition, whilst stability in either stage must be related to chamber aerodynamics. The flow patterns in the first stage have been analysed using tracer techniques (34). Second stage stability is related to interstage heat removal, as well as first stage operating conditions, hence an operating limit on the second stage will be imposed by the amount of interstage heat removal required to cause blowout.

Most of the experimental data required for the present optimisation study have been obtained previously (2,3,34), with the exception of stability measurements on the second stage chamber and some measurements on stack gas solids not collected in the venturi-scrubber-cyclone system. Residence time studies for the heat transfer section and second stage chamber were not attempted as the detection equipment was not suitable for recording the considerably shorter mean residence times associated with these sections.

3.3. Equipment Required.

The determination of the operating limits of the second stage chamber with respect to air loading and interstage heat removal requires the cooling of the inlet combustible gas stream until the flame in the chamber is

extinguished. It was found that the indirect heat removal achieved in the cooling tube was gas side controlled (see Sections 4.7 and 5.3) and was insufficient to cause flame extinction in the second stage. Greater heat removal could be achieved by redesign of the heat transfer section or by the installation of a direct cooling device. The latter alternative was chosen as the simpler of the two.

Direct cooling could be achieved by the injection of either inert gas or water spray. Preliminary calculations indicated that excessive amounts of inert gas would be required to achieve sufficient cooling, hence a water spray probe was designed for insertion into the cross piece. Design calculations and operating procedure are given in Appendix 2. The spray probe and schematic diagram of the experimental configuration are shown in Fig. 3-5 and Fig. 3-6 respectively.

The probe was constructed of stainless steel with brass couplings, and had constant water cooling through the body and head. The nozzle used was a hollow cone swirl spray type with a nominal included angle of 40° . It was found that high combustion gas flow past the probe head reduced the included angle to $20^\circ - 25^\circ$ thus preventing spray impingement on the cooling tube walls. The drop size distribution in a spray from a nozzle similar to this one indicated that D_{sm} was about $250 \mu m$. The pressure drop across the nozzle was about 170 kN/m^2 at a flow number of 4.5.

Measurements of the solid content of the stack gases after the venturi-scrubber-cyclone section were made gravimetrically. A modification to the existing stack was required in order to obtain samples of stack gas solids for

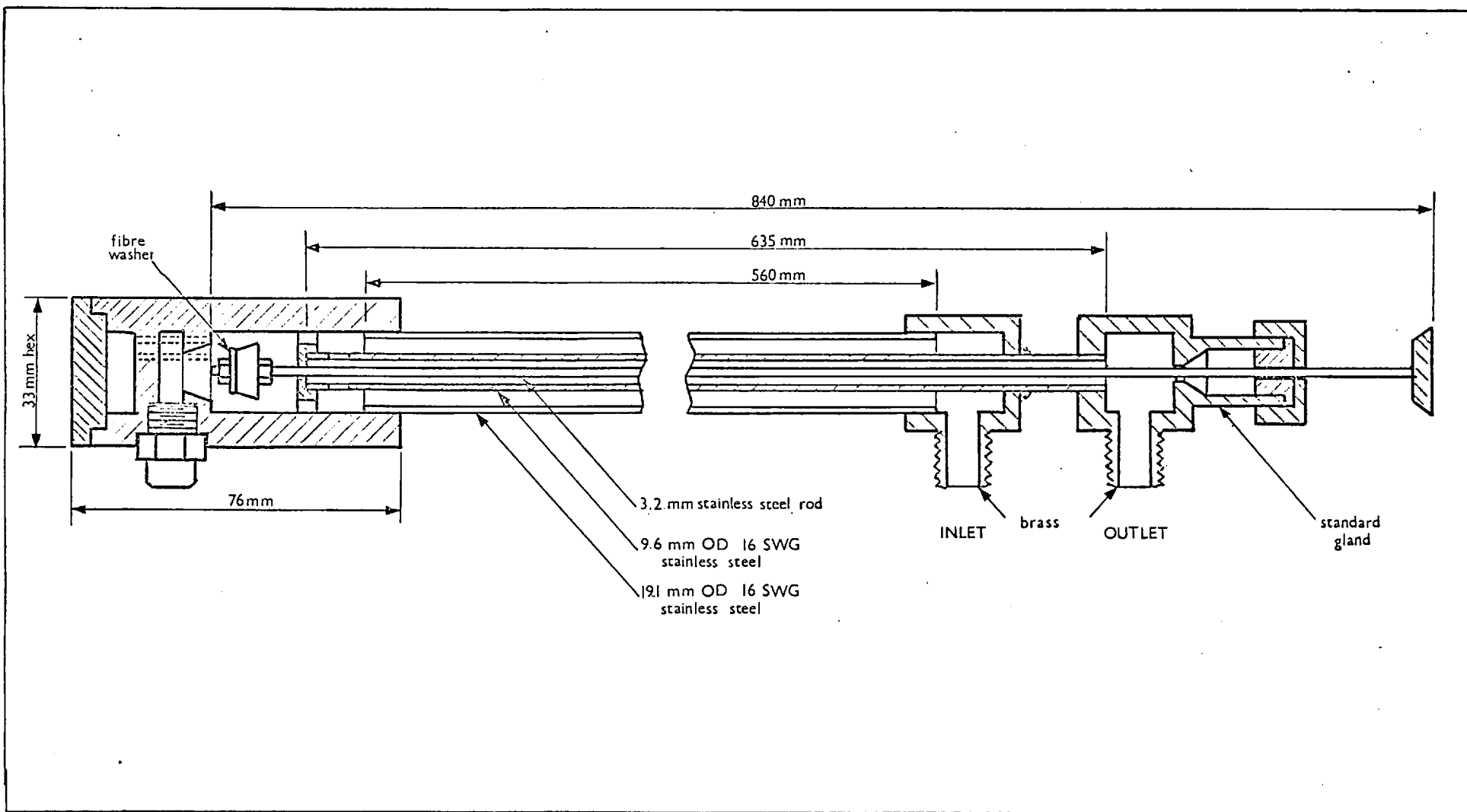


FIG. 3-5 WATER COOLED SPRAY PROBE

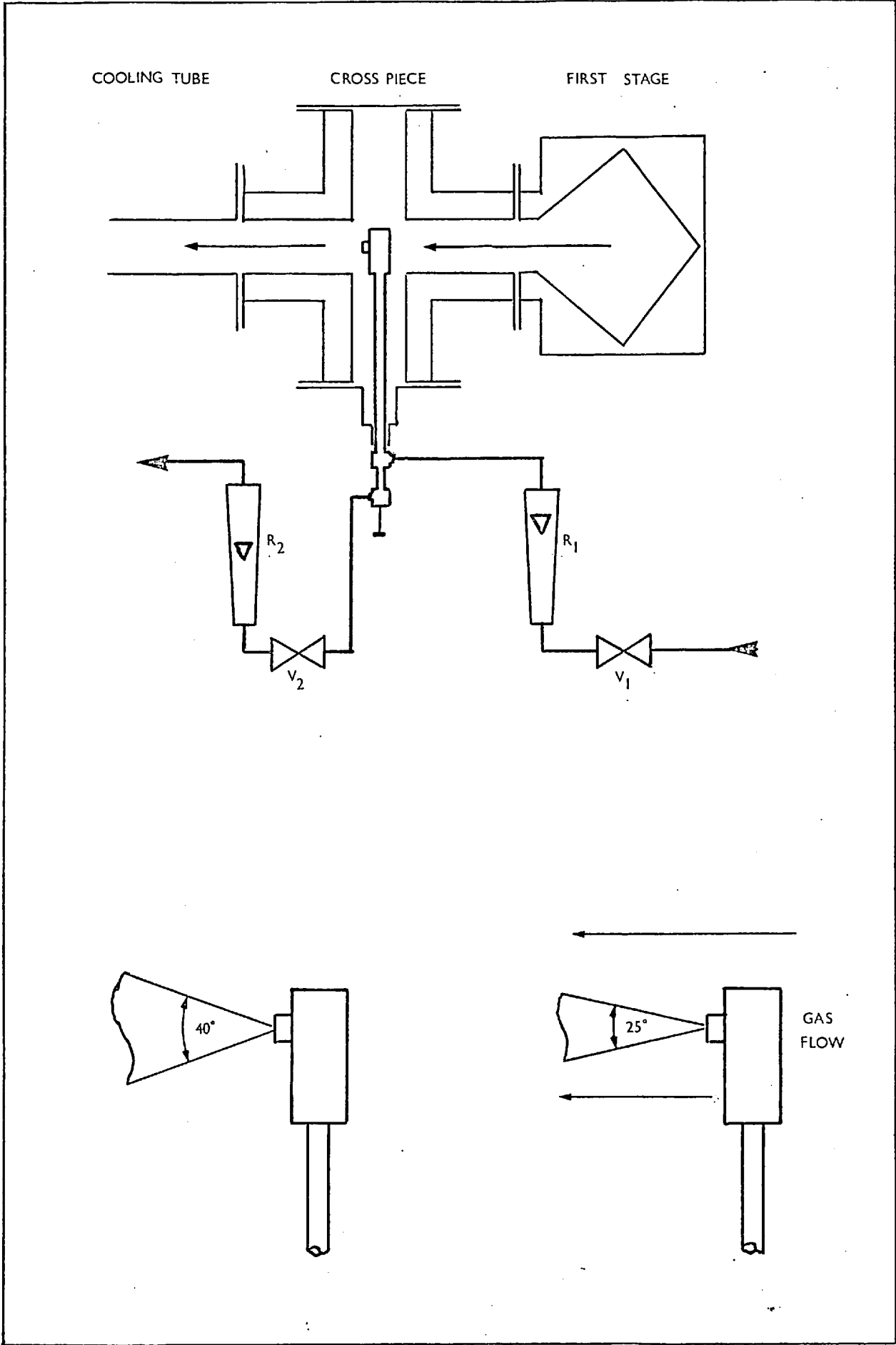


FIG. 3-6 SPRAY PROBE CONFIGURATION

quantitative analysis. This involved the addition of a downcomer section 2.00 m long and 76 mm diameter to the stack outlet. Two 90° bends were included in this section to aid impingement of entrained water at the outlet. The outlet was screened with stainless steel gauze and was directed vertically downwards into a steep sided collection vessel. The stack outlet temperature was recorded, and condensate from the stack was collected for a measured time interval. Checks were made to ensure that there was no accumulation of condensate in the piping. Calculation of the stack gas solid content is shown in Appendix 7.

CHAPTER 4.

DEVELOPMENT OF MATHEMATICAL MODELS.

System models are required to describe individual process sections and overall system performance. Model building is an iterative procedure involving the construction of a tentative model based on theoretical or experimental information. Experiments are then made on the system to verify the model. If the model is inadequate, it is modified and the testing procedure is repeated. In some situations, the only experimental data available for a sub-section or specific operational aspect have been obtained during the early stages of the development of the system, so that they cannot be repeated. In these cases, only one iteration is possible.

Ultimately, the model may be used as a basis for an optimisation study of the system. The optimisation of system operation can sometimes be carried out using a set of designed experiments to investigate system behaviour. Where a set of designed experiments is not practical or where optimisation of design is the objective, use must be made of the system model.

It must be remembered that where an overall optimisation is attempted, some sub-sections of the system will have a greater or lesser effect on performance than others. Where the system is complex and experimental data are not easily obtained, it is difficult to foresee which parameters will be significant.

The accuracy of any such model is thus limited by the quantity and quality of experimental information as well as system criteria and the designer's insight.

The method of approach requires the initial determination of the objectives and the construction of suitable objective functions. Screening experiments are then performed to determine which input variables are important, and which process variables can be controlled. Where possible, a sensitivity analysis of the objective function is performed to determine the required degree of sophistication of the various models. The model building strategy which is then followed may be either empirical or mechanistic.

4.1. Empirical Models.

Empirical models are based on experimental information that has been interpreted from a purely mathematical aspect. The simplest form of model building is the fitting of a line or curve to experimental data by eye. This procedure is improved by the use of the least squares method to minimise the error sum of squares. This basic technique is the foundation for most of the more complex linear and non-linear methods.

The empirical methods of model formulation have inherent drawbacks. While they are useful where carefully designed experiments have been made, or where there is a large volume of experimental data, they are misleading where data are limited or unplanned. In particular, unless there is some theoretical justification for extrapolation, the regression equations apply only within the limits of experimental evidence. It must be remembered as well, that regression techniques give only a mathematical interpretation of experimental evidence, and do not, therefore, give reliable insight into the mechanism of the process.

4.2. Mechanistic Models.

Mechanistic models are usually more realistic than empirical models and are derived from theoretical arguments. They may be considerably more complex than empirical models and generally describe a mechanism rather than a directly observed result. In practice, the mechanistic model has better extrapolation properties than its empirical counterpart. Mechanistic models are generally more time consuming to use and this disadvantage must be balanced with the degree of sophistication that is required. In some cases, the experimental data are not accurate enough to allow distinction between mechanisms, and only show an overall trend.

Mechanistic and empirical techniques can be successfully combined to eliminate some of their inherent disadvantages by constructing a simplified mechanistic model including a number of parameters. These parameters can be determined using empirical methods so as to give the best correlation between the model and experimental data.

4.3. Economic Factors.

Overall system performance is best indicated by economic analysis. This involves comparison of the operating costs, capital costs and value associated with saleable products. In order to make such a comparison, relative costs must be associated with each raw material, piece of capital equipment and final product. The cost comparison of each sub-section must be made on a common basis such as per unit of raw material, product or time. The result however, may be highly dependent on the choice of relative costs, hence

qualitative advantages and disadvantages may be underestimated.

System economy is expressed in terms of initial costs, operating costs and product value. The cost of each piece of capital equipment has been determined from literature or from supplier's quotation. Capital cost has been related to unit capacity using a power law expression

$$\text{Capital Cost} = \text{constant 1} \cdot (\text{Capacity})^{\text{constant 2}}$$

4.1

The capital cost functions in terms of pounds sterling and the installation cost as a percent of the capital cost are shown in Table 4-1 and 4-2. The "Marshall-Stevens" all-industry-cost-index has been used to bring all costs to equivalent values for the first quarter of 1970. Piping, electricals and extras have been calculated as 50% of the capital cost. The initial costs have been depreciated over seven years using a straight line method. Taxes and other associated costs were included with depreciation to give a plant burden per year of 20% of the capital cost.

Operating costs have been divided into raw material and labour costs. Labour cost has been assigned a fixed value independent of plant capacity, hence it has no effect on the relative economy of operating the system at different conditions. Raw materials include electric power, fuel, steam and air. Power consumption is related to the capacity and operating conditions for each capital item in Table 4-2. The raw material values used in the preliminary study are given in Table 4-3.

TABLE 4-1. CAPITAL COST FUNCTIONS.

CAPITAL EQUIPMENT	QUANTITY	UNITS OF CAPACITY	CAPITAL COST FUNCTION 1 UNIT=£1.00	REF.
FIRST STAGE CHAMBER	1	$V = m^3$	$1445 * V^{**0.272}$	77
SECOND STAGE CHAMBER	1	$V = m^3$	$2220 * V^{**0.272}$	77
HEAT EXCHANGER	2	$A_x = m^2$	$461 * A_x^{**0.652}$	78
EXTRACTOR	1	$C = m^3/s$	$187 * C^{**0.290}$	79
COMPRESSOR	1	$F = kg/s$	$1610 * F^{**0.625}$	79
FAN	1	$F = kg/s$	$4790 * F^{**0.895}$	78
FUEL PUMP	1	$F = kg/s$	$91.8 * F^{**0.048}$	63
WATER PUMP	3	$F = kg/s$	$76.5 * F^{**0.048}$	63
DUST COLLECTOR	1	$C = m^3/s$	$189 * C^{**0.800}$	80

TABLE 4-2. INSTALLATION COST AND OPERATING POWER FUNCTIONS.

CAPITAL EQUIPMENT	INSTALLATION COST % of CAPITAL COST	OPERATING POWER REQUIREMENT kW	EFFICIENCY %	REF.
FIRST STAGE CHAMBER	50	-		78
SECOND STAGE CHAMBER	50	-		78
HEAT EXCHANGER	10 - 30			78
EXTRACTOR	20 - 60	$0.385 \dot{m} T \Delta P / P_s$	.40	78, 63
COMPRESSOR	20 - 60	$0.246 \dot{m} T \Delta P / P_s$	.50	78, 63
FAN	20 - 60	$0.385 \dot{m} T \Delta P / P_s$	.40	78, 63
FUEL PUMP	10 - 50	$1.34 \dot{m} \Delta P / \rho_L$	.50	78, 63
WATER PUMP	10 - 50	$1.34 \dot{m} \Delta P / \rho_L$	.50	78, 63
DUST COLLECTOR	25 - 45	-		78
$\dot{m}$ mass flow rate kg/s T temperature K ΔP pressure drop kN/m ² P absolute pressure kN/m ² ρ_L liquid density kg/m ³				

TABLE 4-3. UTILITY, RAW MATERIAL AND PRODUCT COSTS.

	REPORTED VALUE	VALUE USED	REF.
UTILITY :			
Electrical Power	£.004 - .010/unit	£.004/unit	78
Process Water	£.11 x10 ⁻⁴ - .38 x 10 ⁻⁴ /kg	£.11x10 ⁻⁴ /kg	78
Atomising Steam (100 kN/m ²)	-	£.15x10 ⁻³ /kg	
Atomising Air	-	Compressor operating cost £.22x10 ⁻³ /(kg/s)	
RAW MATERIAL			
Residual Fuel Oil	£.005-.015/kg (fuel oil)	£.004/kg	63, 78
Combustion Air	-	Fan operating cost £.16x10 ⁻⁴ /(kg/s)	
Feed Water	£.15x10 ⁻⁴ -.43x10 ⁻⁴ /kg	£.22x10 ⁻⁴ /kg	78
PRODUCT			
Export Steam (172 kN/m ² , 400K)	£.24x10 ⁻³ -.37x10 ⁻³ /kg	£.37x10 ⁻³ /kg	78
Export Steam (690 kN/m ² , 450K)	£.46x10 ⁻³ -.92x10 ⁻³ /kg	£.92x10 ⁻³ /kg	78

Heat for steam generation is assumed to be the saleable product of the system for the purpose of economic evaluation. The qualitative advantage of independent second stage temperature control has been indicated by assuming the generation of 690 kN/m^2 (100 psig) steam after the first stage and 172 kN/m^2 (25 psig) steam after the second stage. The respective values of the steam are given in Table 4-3.

The bases chosen for economic comparison were one unit of fuel and one year of operation (7200 hr).

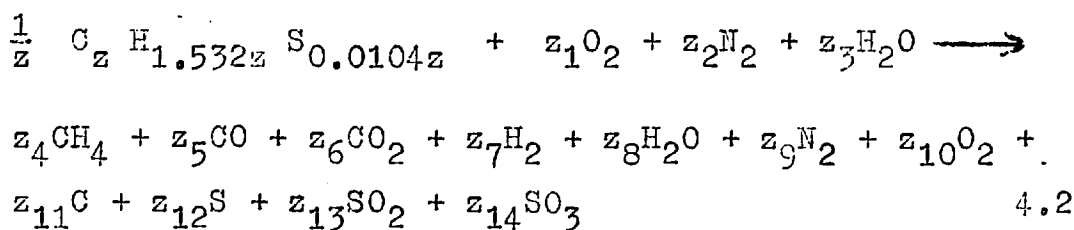
4.4. Gas Composition and Temperature.

The rate of gasification of residual fuel oil in well-stirred high intensity combustors is mass transfer controlled. Where the composition of the fuel is known, the overall gasification reaction can be written as a chemical equation. The empirical formula for the residual fuel oil used is derived from the knowledge of the weight percentage of the constituent elements, given in Appendix 1

$$\text{viz. } C_z H_{1.532z} S_{0.0104z}$$

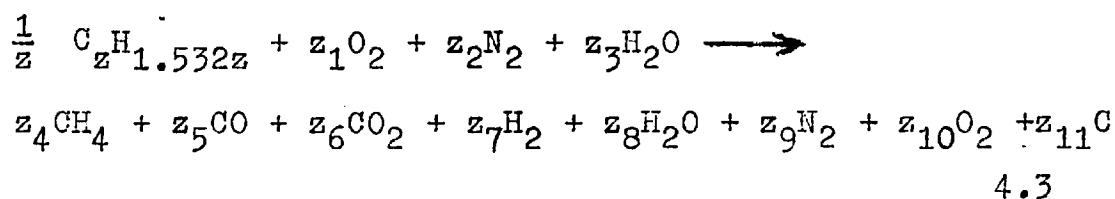
where z is an unknown constant.

The overall reaction can be expressed per mole of carbon in the fuel as



Equation 4.2 includes only those products that have been identified in quantities greater than trace amounts. It is known that SO_3 does not occur under reducing

conditions and that SO_2 never exceeds 0.2% in the exhaust gases (2). Sulphur retained in the solid combustion products does not exceed 5% by weight which represents only 3% of the sulphur entering in the fuel (1,2). Therefore, the effects of sulphur on system performance have been ignored in the formulation of system models. The simplified gasification reaction used is



4.4.1. Equilibrium Model.

Temperature measurements made on the first stage combustion products have indicated that they correspond closely to those in water gas shift equilibrium (1). Dry gas compositions are similar to those expected from the attainment of homogeneous water gas shift equilibrium (1,2).

The constants on the right hand side of equation 4.3 are unknown. Previous work (1,2) has indicated for steam atomisation and $\phi_1 > 1.1$, that methane concentration is linearly related to first stage equivalence ratio ϕ_1 . The following equation has been fitted to the data in Ref.(3)

$$z_4 \pm 0.0040 = (0.0428 \phi_1 - 0.0481) \left(\sum_{i=4}^{10} z_i - z_8 \right) \quad 4.4$$

The constant z_{11} is related to equation 3.1 viz

$$z_{11} = 0.861 m_{T1} \quad 4.5$$

since the weight fraction of carbon in the fuel is 0.861.

Component mass balances on the system are written as follows

Carbon :

$$1 = z_4 + z_5 + z_6 + z_{11} \quad 4.6$$

$$\text{Hydrogen as } H_2 : \quad 0.766 = 2z_4 + z_7 + z_8 - z_3 \quad 4.7$$

$$\text{Oxygen as } O_2 : \quad z_1 = 0.5(z_5 + z_8 - z_3) + z_6 + z_{10} \quad 4.8$$

$$\text{Nitrogen :} \quad z_2 = z_9 \quad 4.9$$

Under reducing conditions, the oxygen concentration in the outlet gases is usually equal to zero, hence

$$z_{10} = 0 \quad 4.10$$

If the system is in water gas shift equilibrium, the equation

$$K_{e1} = \frac{z_5 z_8}{z_6 z_7} \quad 4.11$$

is satisfied at a given temperature T . The calculation of the equilibrium constant K_{e1} from thermodynamic considerations is given in Appendix 6. Other reversible reactions such as the heterogeneous water gas reaction and methane reforming reaction do not attain equilibrium under present conditions. If this simplified model is modified to include the fuel evaporation efficiency ϵ_v , defined as the fraction of the fuel entering the chamber that is evaporated, equations 4.6 to 4.8 become

$$\epsilon_v = z_4 + z_5 + z_6 + z_{11} \quad 4.12$$

$$0.776 \epsilon_v = 2z_4 + z_7 + z_8 - z_3 \quad 4.13$$

$$\epsilon_{O_2} (z_1) + 0.5 z_2 = 0.5(z_5 + z_8) + z_6 + z_{10} \quad 4.14$$

where z_{11} now represents the solid products excluding the char left from incomplete evaporation hence equation 4.5 does not apply.

The oxygen consumption efficiency ϵ_{O_2} ($\epsilon_{O_2} = 1$ in the previous model) is defined as the fraction of the oxygen entering that is consumed by combustion. The

dependence of ϵ_{O_2} on ϵ_v is not known, however it has been suggested (34) that ϵ_{O_2} may be taken to be equal to ϵ_v without loss of accuracy. Under gasifying conditions, however $\epsilon_v < 1$ and no oxygen has been detected, therefore in this analysis $\epsilon_{O_2} = 1$. The general method for determining ϵ_v has been described previously. For the particular flow system considered here (see Fig.3-2), ϵ_v has been determined as described by HOPKINS (34).

An enthalpy balance is used to determine the combustion temperature. In general, the increase in enthalpy results from heat generated by reaction

$$H_r = \sum H_s(\text{products}) - \sum H_s(\text{reactants}) + \sum Q_L \quad 4.15$$

where H_r is the heat generated by reaction; H_s is sensible heat, and Q_L is heat loss from the chamber. The standard heat of combustion of the fuel is H_c as given in Appendix 1, and corresponds to the heat generated by reaction if the fuel is completely oxidised to CO_2 and H_2O . Since, in the present system, the fuel is not completely oxidised, the heat of reaction must be calculated from the heats of formation of the products and reactants.

$$H_r = \sum H_f(\text{products}) - \sum H_f(\text{reactants}) \quad 4.16$$

The heat of formation of the reactants per mole of carbon in the fuel is

$$H_{f(\text{fuel})} = H_f(CO_2) + 0.766 H_f(H_2O) - H_c \quad 4.17$$

as the heat of formation of dry air is zero. When steam is used as the atomising medium, its heat of formation must be included with that of the other reactants.

The enthalpies of the reactants and products are calculated using known properties of the fuel and standard

heat capacity expressions (62)

$$\sum H_s = \sum x_i \int_{298}^T C_{p_i} dT + \sum x_i \lambda_i \quad 4.18$$

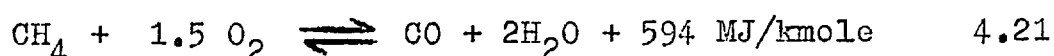
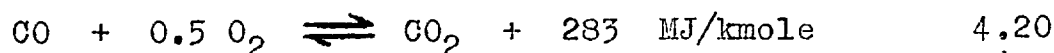
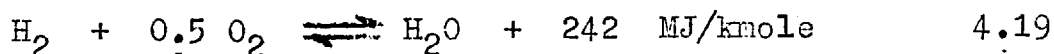
The method of solution of the system involves the assumption of a combustion temperature T from which Ke_1 is evaluated and the product gas composition determined using the foregoing equations. The heat losses Q_L from the first stage chamber are small and have been ignored. The dissociation of CO_2 and H_2O has been considered, but it is negligible at temperatures below 2000 K.

The heat of reaction is calculated using equations 4.15 and 4.16 and the results compared. If they do not correspond to within 1%, successive approximations are made to the combustion temperature until the heat balance is satisfied. The search technique can be shortened by the assumption that the heat capacity of the product gas does not vary appreciably with composition and temperature, resulting in rapid convergence with a slight loss of accuracy. A pattern search technique based on a bisectional search gave convergence in less than 8 function evaluations for all cases and without the limitation of invariant heat capacities. This latter method was used throughout the calculations.

4.4.2. Reaction Rate Model.

The gaseous products from the second stage are adequately described by the equilibrium model discussed in the previous section, using equation 3.9 to determine z_{11} in equation 4.5. In order to interpret flame stability and carbon burn-out, a dynamic model involving the reaction rate expressions for combustible gases and solids is required.

The combustibles entering the second stage chamber are hydrogen, carbon monoxide, methane and carbonaceous solids. The stoichiometry of each gas phase reaction is



The combustion of solids occurs more slowly and is therefore omitted from this model. Combustion of solids is discussed in Section 4.6.

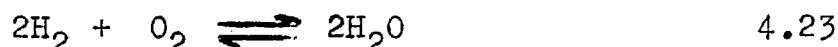
Equations 2.26 and 2.27 were used to predict the burning rates of CH_4 and CO respectively. The reaction rate of H_2 was predicted using an equation similar to equation 2.27

$$\dot{r}_{\text{H}_2} = k_9 x_{\text{H}_2}^{\alpha} x_{\text{O}_2}^{\beta} x_{\text{H}_2\text{O}}^{\gamma} \exp\left(-\frac{E_a}{RT}\right) \left(\frac{P}{RT}\right)^{\delta+\beta+\gamma} \quad 4.22$$

where α , β and γ have the same values as in equation 2.27.

$$k_9 = 3 \times 10^{15} \frac{\text{kmole}}{\text{m}^3 \text{s}} \quad \text{and} \quad E_a = 1.17 \times 10^5 \frac{\text{kJ}}{\text{kmole}}$$

Equations 2.26, 2.27 and 4.22 describe the forward reaction rate or oxidation of combustibles. These reactions are essentially irreversible. At high temperatures, dissociation occurs and an equilibrium is achieved between the forward and reverse rates. Using hydrogen as an example,



the forward and reverse rates can be written as

$$\dot{r}_{\text{H}_2} = k_f x_{\text{H}_2} x_{\text{O}_2}^{0.5} \quad 4.24$$

$$\text{and} \quad \dot{r}_{\text{H}_2}' = k_r x_{\text{H}_2\text{O}} \quad 4.25$$

respectively, where k_f and k_r are the forward and reverse reaction rate constants. The equilibrium constant for this

reaction Ke_{H_2} is evaluated from thermodynamic considerations.

$$Ke_{H_2} = \frac{x_{H_2O}}{x_{H_2} x_{O_2}^{0.5}} = \frac{k_f}{k_r} \quad 4.26$$

Hence, using the above equations, the reverse rate can be written in terms of known quantities

$$\dot{r}'_{H_2} = \frac{1}{Ke_{H_2}} \frac{x_{H_2O}}{x_{H_2} x_{O_2}^{0.5}} \dot{r}_{H_2} \quad 4.27$$

A similar approach is used for CO and CH_4 , thus avoiding computational problems arising from the large differences in reaction rates of the reacting species. The net rate of reaction is defined as

$$\dot{r}^* = \dot{r} - \dot{r}' \quad 4.28$$

The reaction rate model yields the same result as the equilibrium model if the system is allowed sufficient time to reach equilibrium (see Fig. 5.6).

4.5. Solids Formation.

Photographic studies have shown that at least two types of carbonaceous solids are formed in the gasifier while as many as three types may be present in the exhaust from the second stage. The particles formed in the gasifier are classified by origin and size. Particles larger than $0.1 \mu m$ diameter are chars resulting from incomplete fuel evaporation. Particles smaller than $0.1 \mu m$ are soot formed by thermal decomposition of methane and other hydrocarbons in the gas phase.

The operation of the system is constrained by the requirement of acceptably clean exhaust gases, hence the control of solid formation in the gasifier and combustion of these solids in the second stage are prime objectives.

The rate of diffusionally controlled carbon particle combustion is dependent on particle diameter, hence the distribution of particle sizes as well as the total mass of solids from the gasifier is required. The simple empirical model based on equation 3.1 is inadequate. The method of solid collection does not allow accurate particle size analysis because of agglomeration, hence some type of mechanistic model is required to determine the particle size groups formed.

4.5.1. Char Formation.

The evaporation model for spray combustion can be used to predict the amount of char formed. For pure component fuel oils, fuel properties remain approximately constant until the drop is completely evaporated. For wide cut residual fuel oils, fuel properties are not constant, hence the following modifications have been made to the evaporation model.

1). The fuel particle surface temperature T_s is related to the fractional evaporation efficiency ϵ_v . Equation 4.29 is based on the boiling point curve for a residual fuel oil similar to "Vanilla" (22)

$$T_s = 750 \epsilon_v + 173 \quad K \quad 4.29$$

where $\epsilon_v \geq 0.4$.

2). The char particle density ρ_c has been related to the fractional evaporation efficiency

$$\rho_c = \rho_L + 100 \epsilon_v \quad \text{kg/m}^3 \quad 4.30$$

based on the initial fuel oil density ρ_L of 932 kg/m³ and a final char density between 10% and 20% higher than the original fuel (63, p. 9-5).

3). The heat of vaporisation λ of the fuel is a function of particle surface temperature where

$$\lambda = 529 - 0.349 T_s \quad \frac{\text{kJ}}{\text{kg}} \quad 4.31$$

The data used to determine this relation were taken from reference (64, p.363).

The determination of the char particle number size distribution requires the solution of equation 2.1.

The value of n for an evaporating spray has been determined using evaporation theory (19). It is known that n increases slightly with ε_v but is generally in the range 1.5 - 1.6 for steam atomised residual fuel oil sprays. In the case of non-evaporating drops, the D^2 law applies, i.e. $n = 2$.

HOPKINS (34) has shown that equation 2.1 corresponds to the drop size distribution from a twin fluid atomiser measured by FRASER and EISENKLAM (18, Curve V), if $m = n - 1$ and the constants in the equation are

$$\begin{aligned} n &= 1.55 \\ k_1 &= 0.002 \\ k_2 &= 0.00129 \end{aligned}$$

if D is expressed in μm . These constants are applicable for one set of operating conditions only.

The dependence of G_n on operating conditions can be specified by the use of equation 2.5 or 2.6 as follows. Equation 2.3 is integrated as shown in Appendix 3 to give the relation between the constants in equation 2.1 and D_{sm}

$$D_{sm} = \frac{3 \Gamma\left(\frac{3}{n}\right)}{2k_2^{1/n} \Gamma\left(\frac{2}{n}\right)} \quad 4.32$$

Using the value of n determined from evaporation theory (34)

and D_{sm} from equation 2.5, k_2 is found from 4.32. To solve for k_1 , the normalising equation 4.33

$$\int_0^{\infty} G_n dD = 1 \quad 4.33$$

is integrated to give

$$k_1 = nk_2 \quad 4.34$$

as shown in Appendix 3.

Equation 2.4 may be used in conjunction with 2.6 to yield a similar result, but requires a trial and error procedure involving numerical integration.

The total number of char particles formed, N_c , is assumed to be equal to the initial number of drops in the spray N_t , which is assumed to remain constant throughout the furnace. Using 1 kg of fuel as a basis, the number of drops in the spray is calculated as

$$N_t = \frac{6}{\pi \rho_L \int_0^{\infty} G_n D^3 dD} \quad 4.35$$

Combining 2.1 and 4.35 and integrating,

$$N_t = \frac{2n^2 k_2 (1+3/n)}{\pi \rho_L k_1 \Gamma(3/n)} \quad 4.36$$

Since the right hand side of equation 4.36 contains expressions dependent on $\mathcal{E}_v$,

$$N_t = f(\mathcal{E}_v) \quad 4.37$$

The number of drops in the spray corresponds to $\mathcal{E}_v = 0$, however at this point, the evaporation model becomes unstable computationally. To evaluate N_t or N_c , values of $\mathcal{E}_v > 0$ have been considered and Fig. 4-1 shows the hypothetical relation between N_t and $\mathcal{E}_v$ for one set of operating conditions. It is seen that N_t is a linear function of $\mathcal{E}_v$ for $\mathcal{E}_v < 0.8$, hence it is possible to extrapolate to $\mathcal{E}_v = 0$

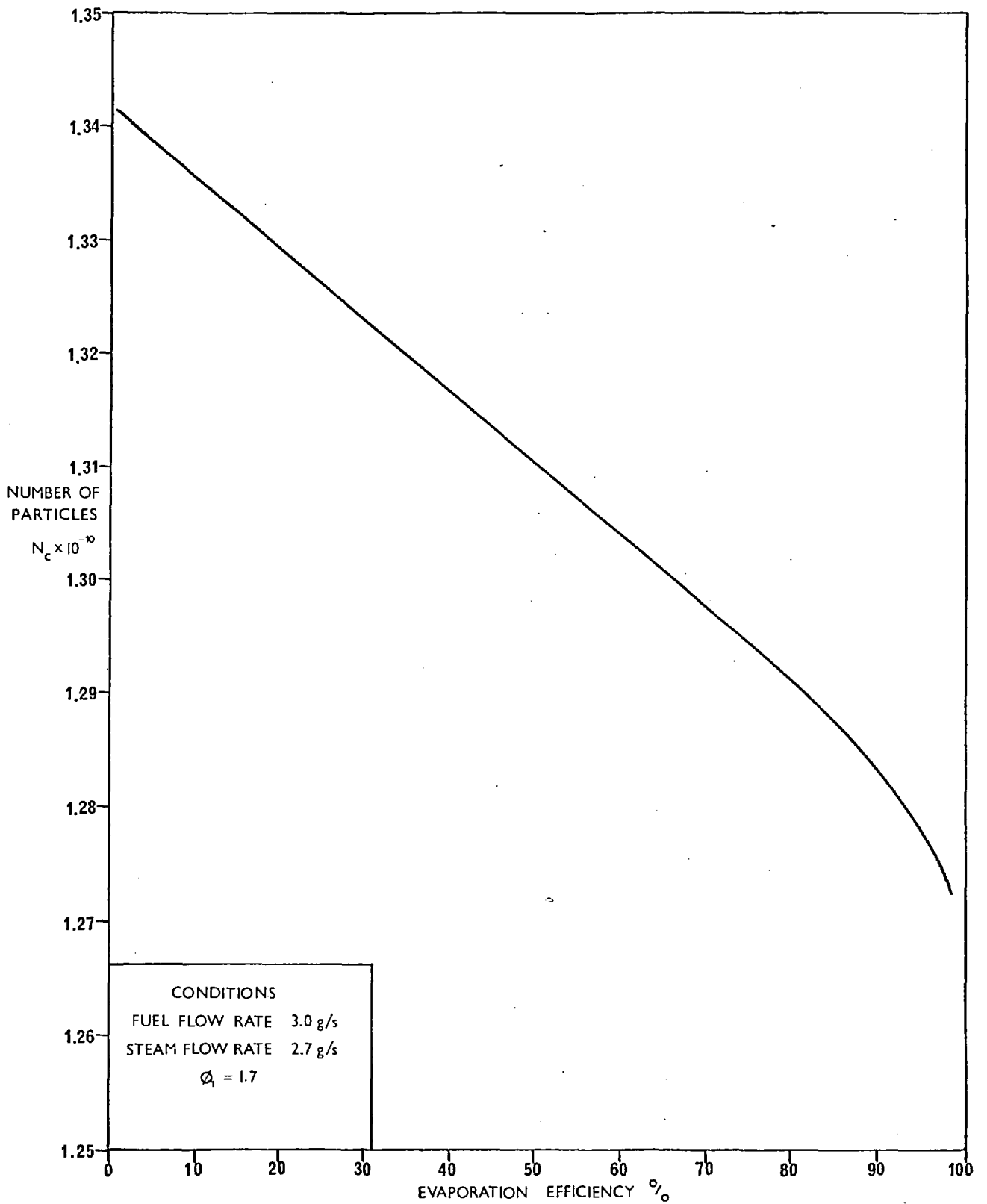


FIG.4-1 THE EFFECT OF EVAPORATION EFFICIENCY ON TOTAL NUMBER OF CHAR PARTICLES FORMED

to determine the true number of drops, and the number of char particles N_c . The method of calculation is outlined in Appendix 3.

The char particles do not change in size in the first stage once they are formed. In the first stage N_c is always equal to N_t , however it may decrease in the second stage where char particle combustion occurs. The char number size distribution G_{nc} is given by equation 2.1 with $n = 2$ in analogy with the rate of combustion of evaporating drops.

$$G_{nc} = k_{14} D^{n-1} \exp(-k_{15} D^n) \quad 4.38$$

The values of the constants k_{14} and k_{15} are determined using the previously calculated value of N_c since

$$N_c = \frac{6 m_c}{\pi \rho_c \int_0^\infty G_{nc} D^3 dD} \quad 4.39$$

The mass of char m_c per kg of fuel is

$$m_c = (1 - \epsilon_v) \frac{\rho_c}{\rho_L} \quad 4.40$$

where ρ_c is char density. The relation between k_{14} and k_{15} is the same as that given between k_1 and k_2 in equation 4.34.

4.5.2. Soot Formation.

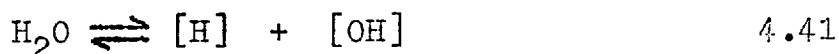
The mass of soot m_s formed per kg of fuel can be determined by subtracting the mass of char m_c (equation 4.40) from the total mass of solids m_{T1} .

At present, there is no established mechanistic model for the prediction of the rate of soot formation in turbulent flames. Various investigators (32,65,66,67) have found that soot formation is influenced by

1. fuel flow rate
2. equivalence ratio
3. mixing characteristics
4. [OH] concentration
5. temperature
6. fuel composition.

In general, it has been found that soot increases with increasing fuel flow rate, equivalence ratio, and C/H ratio in the fuel. Increased mixing reduces soot formation whilst temperature and [OH] concentration appear to give unpredictable results.

In order to reduce the error associated with equation 3.1 and improve the extrapolation properties, the evaporation model has been used in conjunction with operating data (2) to generate values of m_c using equation 4.40. These values have been subtracted from experimentally determined values of m_{T1} and the resulting values of m_s interpreted empirically. Chamber mixing was related to the forward and reverse air flow rates and [OH] concentration was determined assuming equilibrium dissociation



No method was available to determine the soot particle size distribution experimentally, and there is no theoretical basis upon which to base a treatment similar to the char analysis. Published results (65,66) indicate that the arithmetic mean particle diameter for soots lies between 0.02 μm and 0.08 μm . Photographic evidence from the system studied (1) indicated a value between 0.04 μm and 0.05 μm . The soot particle number size distribution in hydrocarbon

flames is known to be approximately normal(65), with a mean particle diameter D_μ of 0.04 μm and a standard deviation σ_d of $\pm 0.008 \mu\text{m}$, assuming no agglomeration.

The number size distribution for soot used in the following analysis is normal.

$$G_{ns} = \frac{1}{\sqrt{2\pi} \sigma_d} \exp \left(-\frac{(D - D_\mu)^2}{2 \sigma_d^2} \right) \quad 4.42$$

where it was assumed that $D_\mu = 0.04 \mu\text{m}$ and $\sigma_d = \pm 0.008 \mu\text{m}$ for all conditions.

The surface mean diameter D_{sm} is found using equations 2.3 and 4.42.

$$D_{sm} = \frac{1 + 3c_1 D_\mu \Gamma\left(\frac{3}{2}\right) + 3c_1^2 D_\mu^2 + c_1^3 D_\mu^3 \Gamma\left(\frac{1}{2}\right)}{c_1 \Gamma\left(\frac{3}{2}\right) + 2c_1^2 D_\mu + c_1^3 D_\mu^2 \Gamma\left(\frac{1}{2}\right)} \quad 4.43$$

$$\text{where} \quad c_1 = \sqrt{\frac{1}{2 \sigma_d^2}} \quad 4.44$$

The number of soot particles N_s per kg of fuel is determined by using equation 4.39 with subscript 'c' replaced by 's'. The equation is then integrated as before to give

$$N_s = \frac{6m_s c_1^3}{\pi \rho_s (1 + 3c_1 D_\mu \Gamma\left(\frac{3}{2}\right) + 3c_1^2 D_\mu^2 + c_1^3 D_\mu^3 \Gamma\left(\frac{1}{2}\right))} \quad 4.45$$

The soot density ρ_s was taken from reference (63, p.3-9) as 2000 kg/m^3 .

4.6. Solids Burning.

Previous particulate burning models are limited by the narrow size ranges for which they have been derived. The mechanistic model developed in this work incorporates the best aspects of the previous models to predict the rate of burn-out of carbon particles from 100 μm to 0.001 μm diameter.

Combustion of solid particles involves the

diffusion of reactant gas to the particle surface, absorption and reaction at the surface, followed by desorption and diffusion of products away from the surface. The rate may be controlled by absorption, desorption or chemical kinetics at the surface. Absorption or desorption control is indicated by a surface reaction rate coefficient K_s given in general form for reactants and products by equation 4.46

$$K_s' = \frac{k_{s1}}{1 + k_{s2} x_r + k_{s3} x_p} \quad 4.46$$

where k_s are constants and x_r and x_p are the mole fractions of the gaseous reactants and products respectively.

Kinetic control is characterised by an Arrhenius type expression

$$K_s'' = k_{s\infty} \exp \left(-\frac{E_a}{RT} \right) \quad 4.47$$

If the reaction rate is controlled by one of the diffusional steps, the rate is characterised by a diffusional rate coefficient K_d .

Two mechanistic models are available to describe carbon particle burn-out; the decreasing radius model, and the shrinking core model.

Photographic studies (1,2) have indicated the presence of small numbers of conospheres in the second stage exhaust gases only. Ash covered particles have not been identified although it is known that sodium, vanadium and sulphur compounds exist in the solids. It has been reported (68) that impurities in the solids may affect the burning rate by $\pm 50\%$, which indicates some type of catalytic effect. Since the catalytic effects cannot be estimated in this system, and surface ash layers do not form, the effects of ash have been ignored in this analysis.

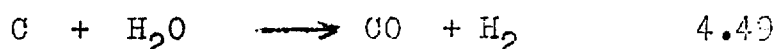
Although there is evidence to support particle swelling during combustion (69), the decreasing radius model has been applied in this analysis. Cenospheres do not appear to form in the first stage and are ignored.

A model combining surface and diffusion control based on the following assumptions is proposed.

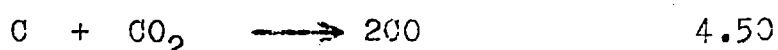
1. $N_{Nu} = 2$.
2. All particles are spherical and non-swelling.
3. The decreasing radius model applies.
4. The only three reactions occurring at the surface are :



where X is the molar ratio of CO_2 to CO formed at the surface,



and



The reaction indicated by equation 4.48 is exothermic, those indicated by 4.49 and 4.50 are endothermic.

Arthur (70) has related X to particle surface temperature T_s for burning carbon spheres by the equation

$$\frac{1}{X} = 2500 \exp (0.518 \times 10^5 / RT_s) \quad 4.51$$

The surface reaction rate coefficients for each reaction have been evaluated using semi-empirical equations based on experimental results available in the literature. For equation 4.48, the reaction is kinetically controlled and K_{s1} is calculated using a relation determined by LEE et al (38)

$$K_{s1} = 1.65 \exp (-0.711 \times 10^5 / RT_s) \quad 4.52$$

Reactions 4.49 and 4.50 are absorption controlled, hence

$$K_{s2} = \frac{k_{s1}}{1 + k_{s2} x_{H_2O} + k_{s3} x_{H_2}} \quad 4.53$$

$$K_{s3} = \frac{k_{s4}}{1 + k_{s5} x_{CO_2} + k_{s6} x_{CO}} \quad 4.54$$

The values of the constants are those determined by GADSBY and LONG (71) for equation 4.53 and GADSBY and HINSHELWOOD (33) for equation 4.54.

It can be shown by applying Fick's Law to a spherical particle in stagnant surroundings that

$$a_t^2 = a_o^2 - \frac{2b_s c_b D t}{\rho_p \xi} \quad 4.55$$

where a_t is the particle radius at time t ,

a_o is the initial particle radius,

ρ_p is particle density,

c_b is the concentration of the reactant in the bulk gas,

b_s is the molar ratio of solid reactant to gaseous reactant (stoichiometric coefficient)

and ξ is defined as

$$\xi = \frac{p_i}{p_{im}} \quad 4.56$$

where p_{im} is the logarithmic mean partial pressure in the boundary layer.

The mass flux of reactant $\dot{m}(r)$ toward the particle at a distance r from the centre of the particle is defined in terms of the mass flow rate $\dot{M}$

$$\dot{m}(r) = - \frac{1}{4\pi r^2} \dot{M} \quad 4.57$$

or in terms of partial pressure as a function of radius $p(r)$

$$\dot{m}(r) = - \frac{M_0 D}{RT} \frac{dp(r)}{dr} \quad 4.58$$

Logarithmic mean values of T and D in the diffusion layer

are used as an approximation of the actual temperature distribution which is hyperbolic with r .

Using equation 4.57 to equate the mass flux at distances r and a from the particle centre,

$$\dot{m}(r) = \frac{a^2}{r^2} \dot{m}(a) \quad 4.59$$

and combining 4.58 and 4.59

$$\frac{a^2}{r^2} \dot{m}(a) = \frac{M_c \mathcal{D}}{RT} \frac{dp(r)}{dr} \quad 4.60$$

Integration between a and r yields

$$a^2 \dot{m}(a) \left(\frac{1}{a} - \frac{1}{r} \right) = \frac{\mathcal{D} M}{RT} (p(r) - p(a)) \quad 4.61$$

As $r \rightarrow \infty$, this equation reduces to

$$a \dot{m}(a) = \frac{\mathcal{D} M}{RT} (p(\infty) - p(a)) \quad 4.62$$

where $p(\infty)$ is the partial pressure of the reactant in the bulk gas.

The specific mass rate of consumption of carbon is related to the mass flux of reactants at the particle surface, viz.

$$\dot{m} = \dot{m}(a) \frac{M_c}{M} b_s \quad 4.63$$

where M_c is the molecular weight of carbon.

Combining 4.62 and 4.63

$$\dot{m} = K_d (p(\infty) - p(a)) \quad 4.64$$

where

$$K_d = \frac{12 b_s \mathcal{D}}{a RT} \quad 4.65$$

and characterises gas film diffusional resistance.

Where the reaction is surface controlled, the rate is written as

$$\dot{m} = K_s p(a) \quad 4.66$$

As diffusion and surface reaction take place in series, the reaction rate is controlled by the slower step. The two steps are combined by equating equations 4.64 and 4.66 thus

eliminating $p(a)$.

$$\dot{m} = \frac{p(\infty)}{\frac{1}{K_d} + \frac{1}{K_s}} \quad 4.67$$

Equation 4.67 applies for each of the three reactions.

The molecular diffusion coefficient for the i^{th} diffusing component $\mathcal{D}_{iN_2}$ is evaluated using equation 4.63 assuming a nitrogen atmosphere (73)

$$\mathcal{D}_{iN_2} = 0.001858 T^{3/2} \sqrt{\frac{M_i + M_{N_2}}{M_i M_{N_2}}} / (P \sigma_{iN_2}^2 \Omega_D) \quad 4.68$$

M_i is the molecular weight of the diffusing species, P is the total pressure. The collision integral Ω_D and the collision diameter σ_{iN_2} are determined using the method described in ref. (76). The equation has been derived for diffusion of dissimilar binary gases. There is no simple expression for determining multi-gas diffusion coefficients. In this case, the net diffusion coefficient of the i^{th} species is

$$\mathcal{D}_i = \frac{x_i \mathcal{D}_{iN_2}}{\sum_i x_i} \quad 4.69$$

where x_i is the mole fraction of the diffusing reactant, and the summation includes reactant mole fractions only.

The evaluation of both K_d and K_s requires knowledge of the particle surface temperature T_s . The temperature is calculated using an iterative heat balance about the particle as follows.

The heat generated per unit surface area is

$$\dot{q}_G = \sum_{i=1}^3 \dot{m}_i H_{r,i} \quad 4.70$$

For the purpose of heat loss calculations, the particle was assumed to be stationary with respect to the surrounding gas hence $N_{Nu} = 2$. For small particles $N_{Nu} \approx 2$ even at a

particle/gas relative velocity of 100 m/s.

Heat loss from the stationary particle occurs by conduction and radiation only. Heat loss by conduction is

$$\dot{q}_D = - \frac{k (T_s - T)}{\Delta r} \quad 4.71$$

The mean thermal conductivity of the surrounding gas is k and the diffusional temperature boundary layer thickness is Δr . For calculation purposes, Δr was set equal to particle diameter, however little change in the heat balance occurred when Δr was varied between 0.5 and 2 particle diameters.

Heat loss by radiation is

$$\dot{q}_r = \sigma \epsilon_p (T_s^4 - T^4) \quad 4.72$$

The value of particle emissivity is known to be of the order 0.5 to 1.0 for small black particles (37). Figure 4-2 shows the effect on reaction rate of changing ϵ_p . In this analysis ϵ_p has been set at unity.

The iterative heat and mass balance was similar to that described in Section 4.4.

A comparison of the results from various carbon burning models including the one derived in this work (curve 8) is shown in Figures 2-2, 4-3 and 4-4. In Fig.2-2, the transition from surface to diffusion control is predicted using this model as particle surface temperature increases. Similar comparisons are made for particles 1 μm and 0.1 μm diameter in Fig.4-3 and Fig.4-4 respectively. The diffusion controlled models shown by curves 1 and 2 in Fig.2-2 indicate an increase in specific reaction rate as diameter decreases, and yield exceptionally high results for small particles. MARSDEN's (36) model (curve 5) shown in Figures 4-3 and 4-4

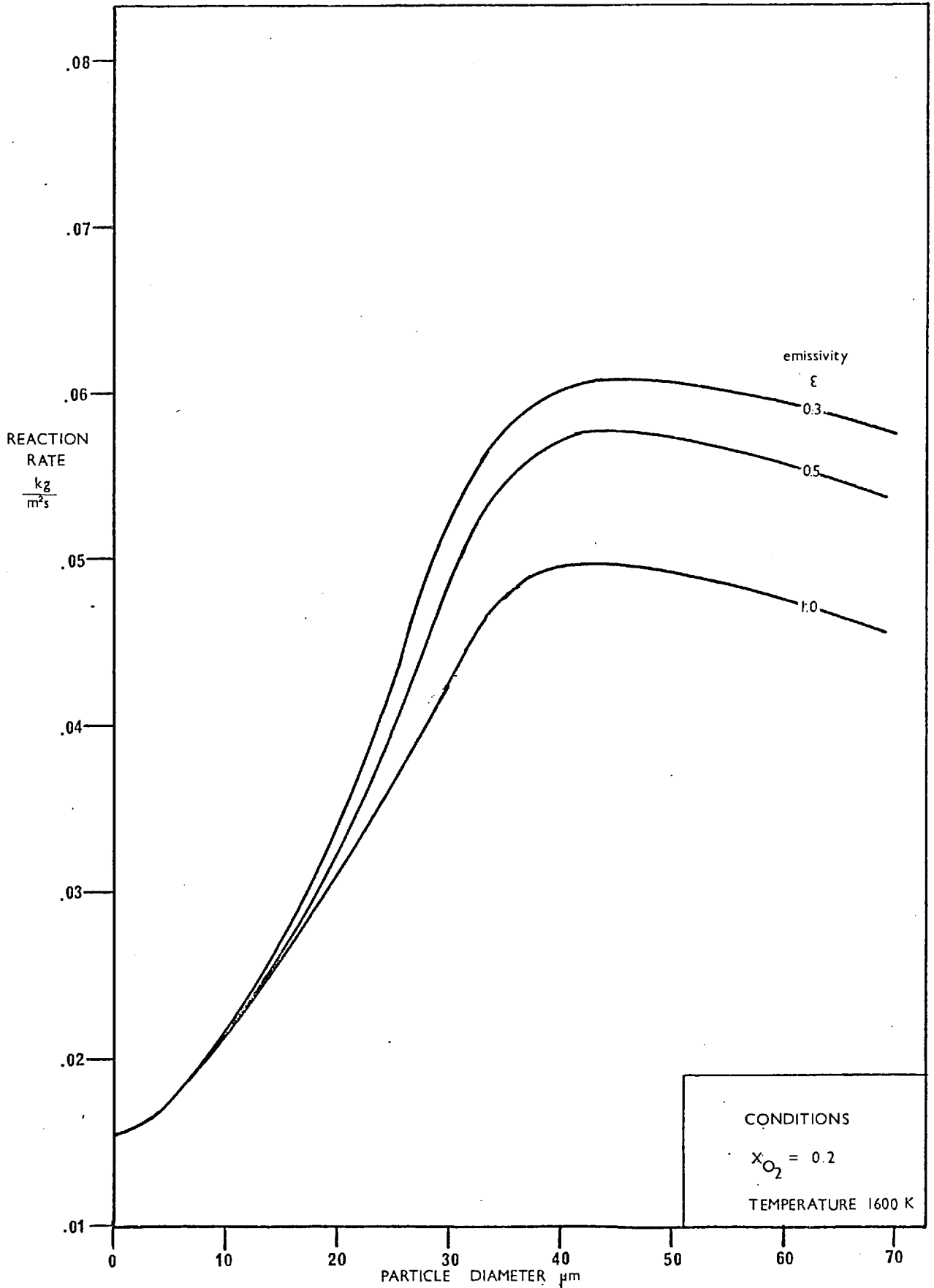


FIG.4-2 THE EFFECT OF PARTICLE EMISSIVITY ON REACTION RATE

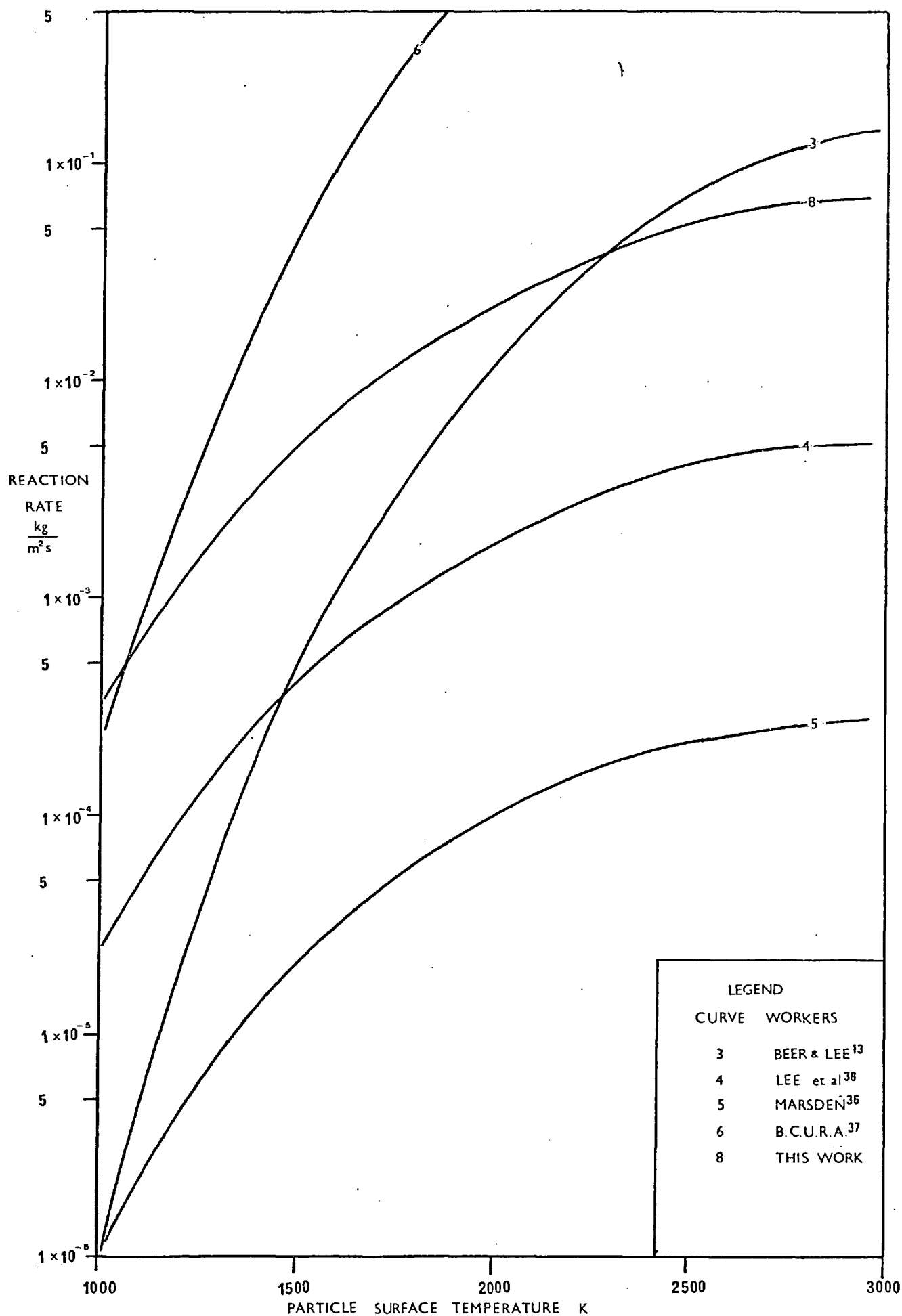


FIG. 4-3 COMPARISON OF CARBON BURNING MODELS FOR
A 1.0 μm PARTICLE

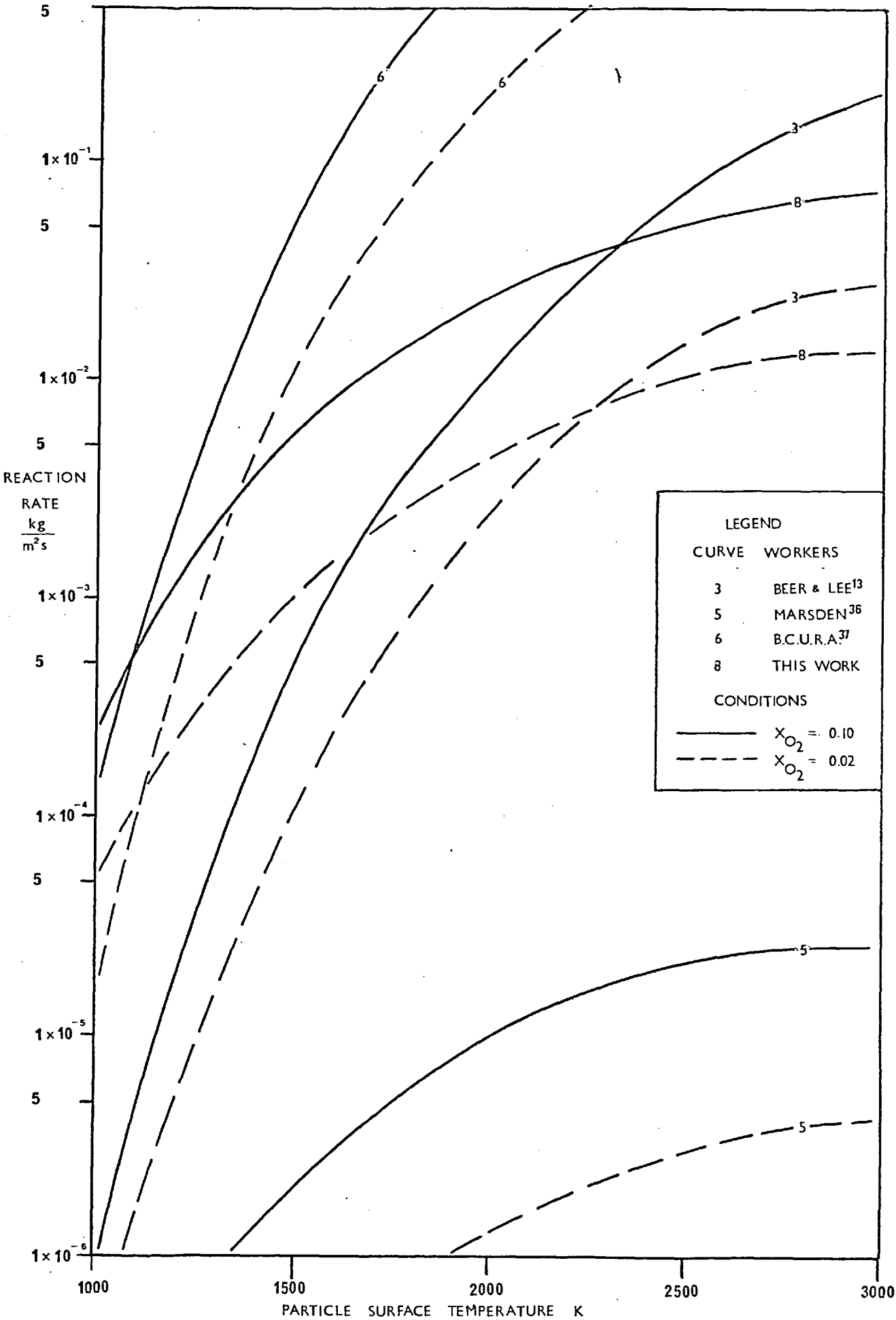


FIG.4-4 COMPARISON OF CARBON BURNING MODELS FOR
A 0.1 μm PARTICLE

indicates a rapid decrease in rate as particle size decreases. It is seen in Fig.4-4 that the predictions from this model are very different from other work. The corresponding broken and full lines in Fig.4-4 indicate the dependence of reaction rate on oxygen concentration.

In Fig.4-5, the effects of particle diameter and oxygen concentration on reaction rate for the model derived in this work are shown. The combustion rate of a cloud of particles is determined as indicated in Appendix 5.

4.7. Heat Transfer.

The use of high intensity combustion chambers for process heating often requires the generation of hot high velocity gases with good radiant properties. Two stage combustion can be used to advantage with heat transfer after each stage if the heat transfer characteristics of the gas can be controlled. Heat transfer in furnaces and air heaters is achieved by a combination of convection and radiation, with radiation predominating in the region of the flame zone. In order to assess the heating quality of the gas and determine the heat transfer surface required, a heat transfer coefficient must be associated with gas properties. Since heat transfer is gas-side controlled, the overall heat transfer coefficient can be determined by considering convection and radiation separately.

4.7.1. Convection.

There are numerous models based on the Nusselt equation for the prediction of convective heat transfer coefficients in turbulent high temperature systems. Those

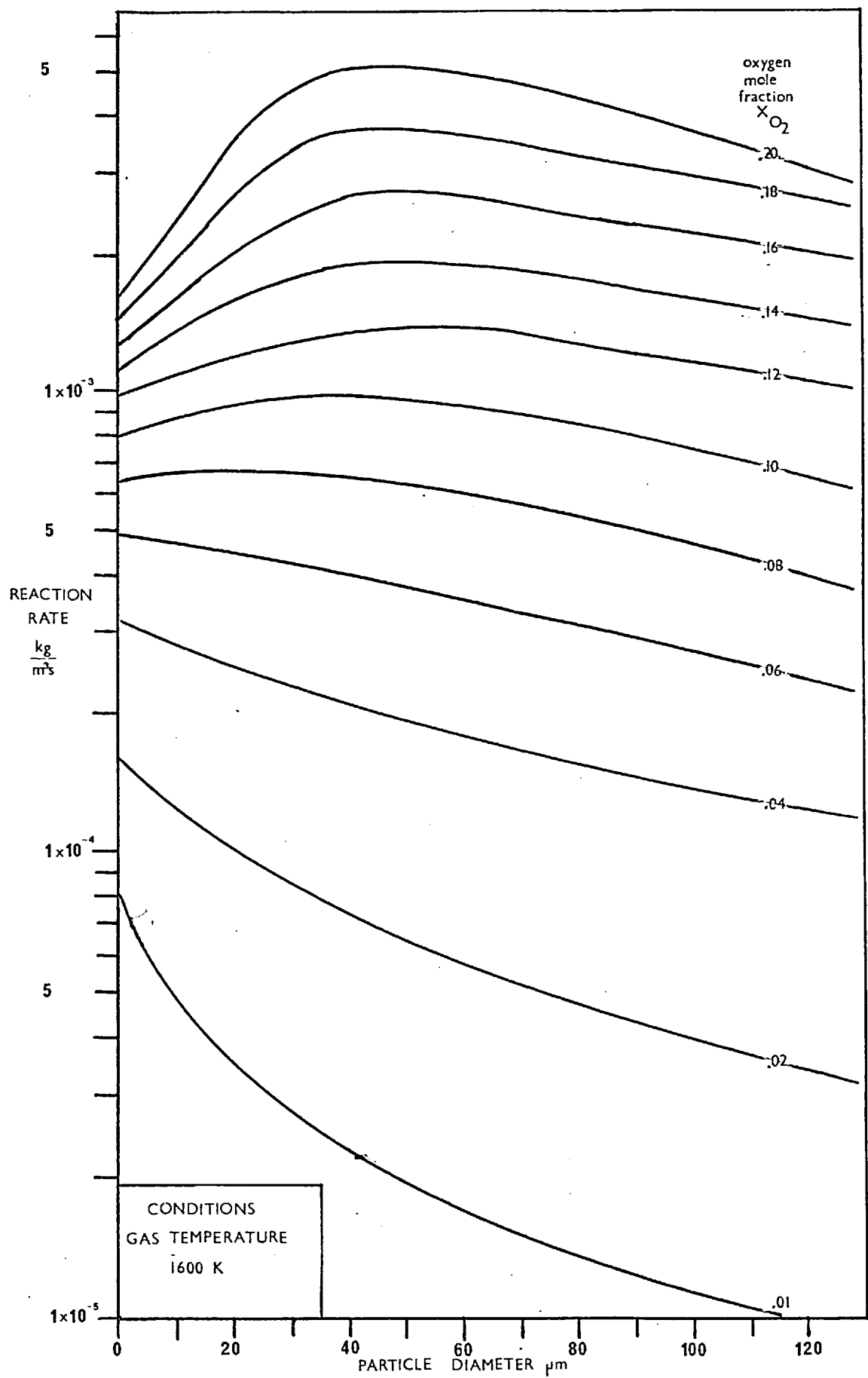


FIG.4-5 THE EFFECT OF OXYGEN CONCENTRATION ON REACTION RATE

considered in this work were the Colburn, Dittus-Boelter and Seider-Tate expressions. The accuracy of these equations is limited by the quality of the high temperature thermodynamic data available. The required thermodynamic properties of gases have been evaluated using the following expressions (64, p.63)

$$\text{Absolute viscosity} \quad \eta = \frac{\alpha_1 T^{1.5}}{T + \beta_1} \quad 4.73$$

where α_1 and β_1 are parameters dependent on gas composition

$$\text{Heat capacity} \quad C_v = \frac{C_p}{1.4 - 5 \times 10^{-5} T} \quad 4.74$$

$$\text{Thermal conductivity} \quad k = 1.6 \eta C_v \quad 4.75$$

In each case, the thermal properties were evaluated at the appropriate convective mean temperatures. The convective heat transfer rate has been predicted from equation 4.76

$$\dot{Q}_c = h_c A_x (T_{cm} - T_w) \quad 4.76$$

where T_w is the wall temperature and using equation 2.44 to determine T_{cm} .

4.7.2. Radiation.

Radiant heat transfer has been predicted using equation 4.77 with the mean radiant temperature defined by equation 2.43

$$\dot{Q}_r = \sigma \epsilon_T A_x (T_{rm}^4 - T_w^4) \quad 4.77$$

The total emissivity ϵ_T was evaluated by combining the individual emissivities viz.

$$\epsilon_T = 1 - (1 - \epsilon_n)(1 - \epsilon_s)(1 - \epsilon_c) \quad 4.78$$

The values of the luminous emissivities ϵ_c and ϵ_s were

calculated using equations 2.48 and 2.49 respectively.

The characteristic radiating dimension L_p is the cooling tube diameter (76 mm). The total projected cross sectional area of the char particles A_p per unit volume of combustion gas is evaluated using expressions derived in Section 4.5.1. For a single particle of diameter D , the cross sectional area is

$$A'_c = \frac{\pi D^2}{4} \quad 4.79$$

The total cross sectional area of all particles per kilogram of fuel is

$$A_c = N_c \frac{\pi}{4} \int_0^\infty D^2 G_{nc} dD \quad 4.80$$

on integrating using G_{nc} given by equation 4.38

$$A_c = \frac{\pi k_1 N_c \Gamma\left(\frac{2}{n}\right)}{n^2 k_2^{1+2/n}} \quad 4.81$$

A_p is found by dividing A_c by the volumetric flow per kg fuel.

The non-luminous emissivity ϵ_n was determined from data available in the literature (10). The non-luminous emissivity attributable to water vapour and carbon dioxide is

$$\epsilon_n = \frac{924 L_b (x_{H_2O} + x_{CO_2}) + 15.2}{T_{rm}} \quad 4.82$$

for $1000 \text{ K} < T_{rm} < 2000 \text{ K}$.

The best value of k_p in equation 2.49 was determined by comparison of the measured heat removal rates and those predicted using equations 4.76 and 4.77. The range of k_p was limited to the literature values for similar systems. By comparing the predicted and measured heat removal rates for each of the convective heat transfer expressions and for a range of k_p , the combination that best described the system could be chosen (see Section 5.3).

4.3. Combustion Stability.

Flame stability in the gasifier is dependent upon atomising conditions and the level of mixing in the chamber. Previous investigators (1,34) have found that the flame is stable over a wide range of conditions, and instability is encountered only at very high or low atomising fluid to fuel mass ratios. Practical limits are discussed in Section 6.3.

Flame stability in the second stage depends on the amount of interstage heat removal, as well as composition and flow rate of the inlet gas streams, and chamber geometry.

The conditions at which the flame is extinguished are the blowout conditions. The determination of these blowout conditions allows the prediction of the maximum interstage heat removal consistent with stable combustion in the second stage.

The analysis of reaction stability involves the initial assumption of adiabatic perfect micromixing. Previously, single component gaseous fuels have been considered using a semi-empirical model based on collision theory. This approach is not suited to this problem since the inlet gases contain H_2 , CO, CF_4 and carbonaceous solids, which burn in parallel. The carbonaceous solids are ignored in the stability analysis as these solids burn much more slowly and have little effect on overall heat generation. The reaction rates for H_2 , CO and CH_4 are calculated using equations 4.22, 2.27 and 2.26 respectively, including the corresponding reverse rate due to dissociation as indicated in Section 4.4.2.

It is seen from Fig.3-3 that the second stage chamber may be considered as a well mixed section followed by a plug flow section. This interpretation is suggested by the countercurrent method of air injection. The combustion reactions occurring in this well mixed section control flame stability, and the following discussion refers to this section only.

For the purpose of analysis, the well mixed section is considered as a perfectly micromixed reactor of volume V_r . Mass and enthalpy balances are determined as indicated in Figure 4.6. Referring to this figure it can be seen that at steady state, the component mass balances per unit time are

$$x_i (F'_5 + F'_7 - F'_8) = \dot{n}_i^* V_r + x_i F'_6 \quad 4.83$$

where x_i' refers to outlet mole fraction.

The conversion of the i^{th} species is $\mathcal{E}_i$ where

$$\mathcal{E}_i = 1 - \frac{x_i F'_6}{x_i (F'_5 + F'_7 - F'_8)} \quad 4.84$$

and combining 4.83 and 4.84

$$\mathcal{E}_i = \frac{\dot{n}_i^* V_r}{x_i (F'_5 + F'_7 - F'_8)} \quad 4.85$$

The mean residence time τ is defined as

$$\tau = \frac{PV_r}{F'_6 RT} \quad 4.86$$

If the conversion in a perfectly stirred reactor as expressed by equation 4.85 is plotted as a function of reactor temperature for reactions 4.19, 4.20 and 4.21, the S-shaped curve ABCD, shown in Figure 4-7, typical of an exothermic reversible reaction, results. As these reactions

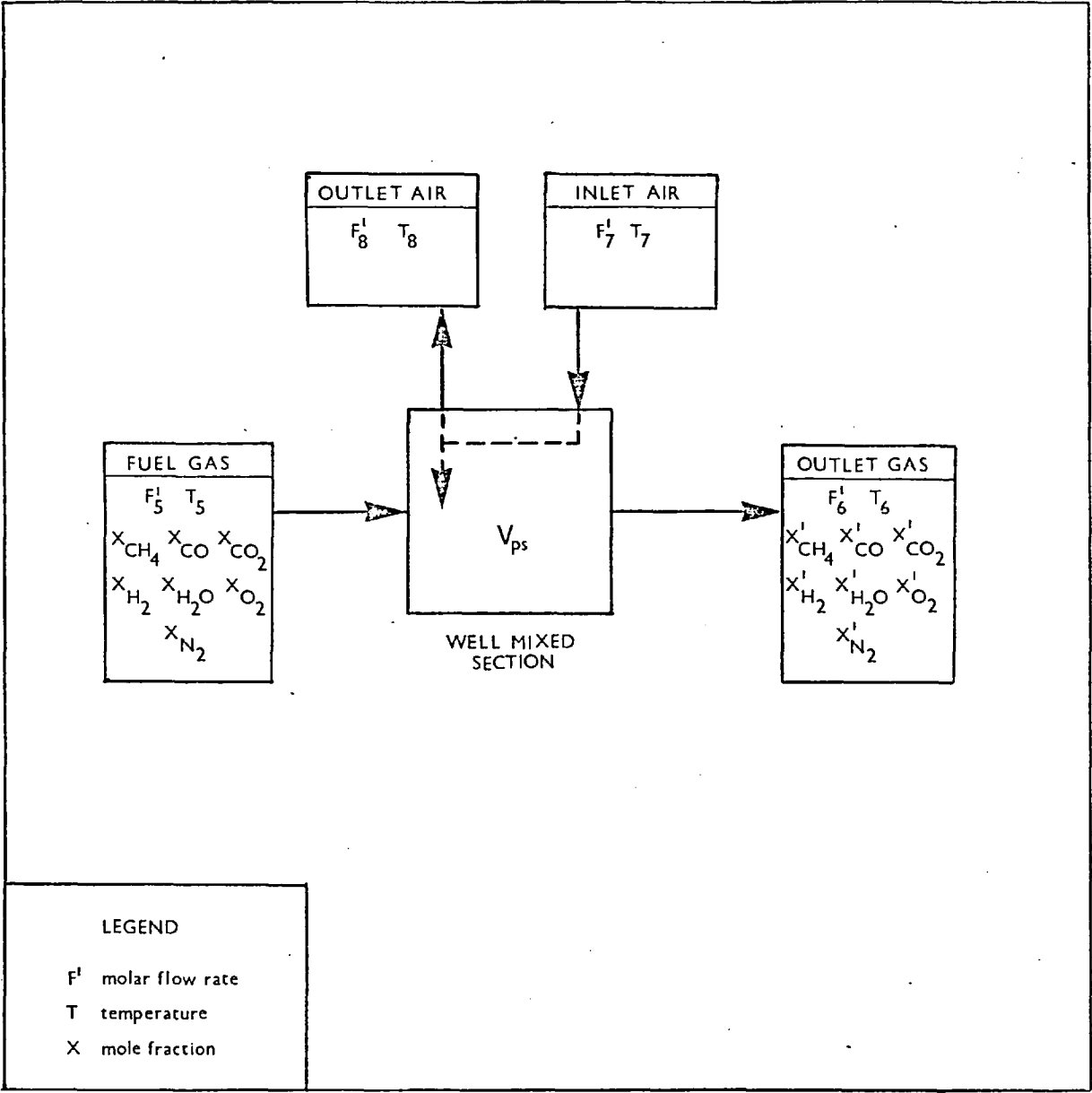


FIG. 4-6 MASS AND ENTHALPY BALANCES ON THE WELL MIXED SECTION OF THE SECOND STAGE

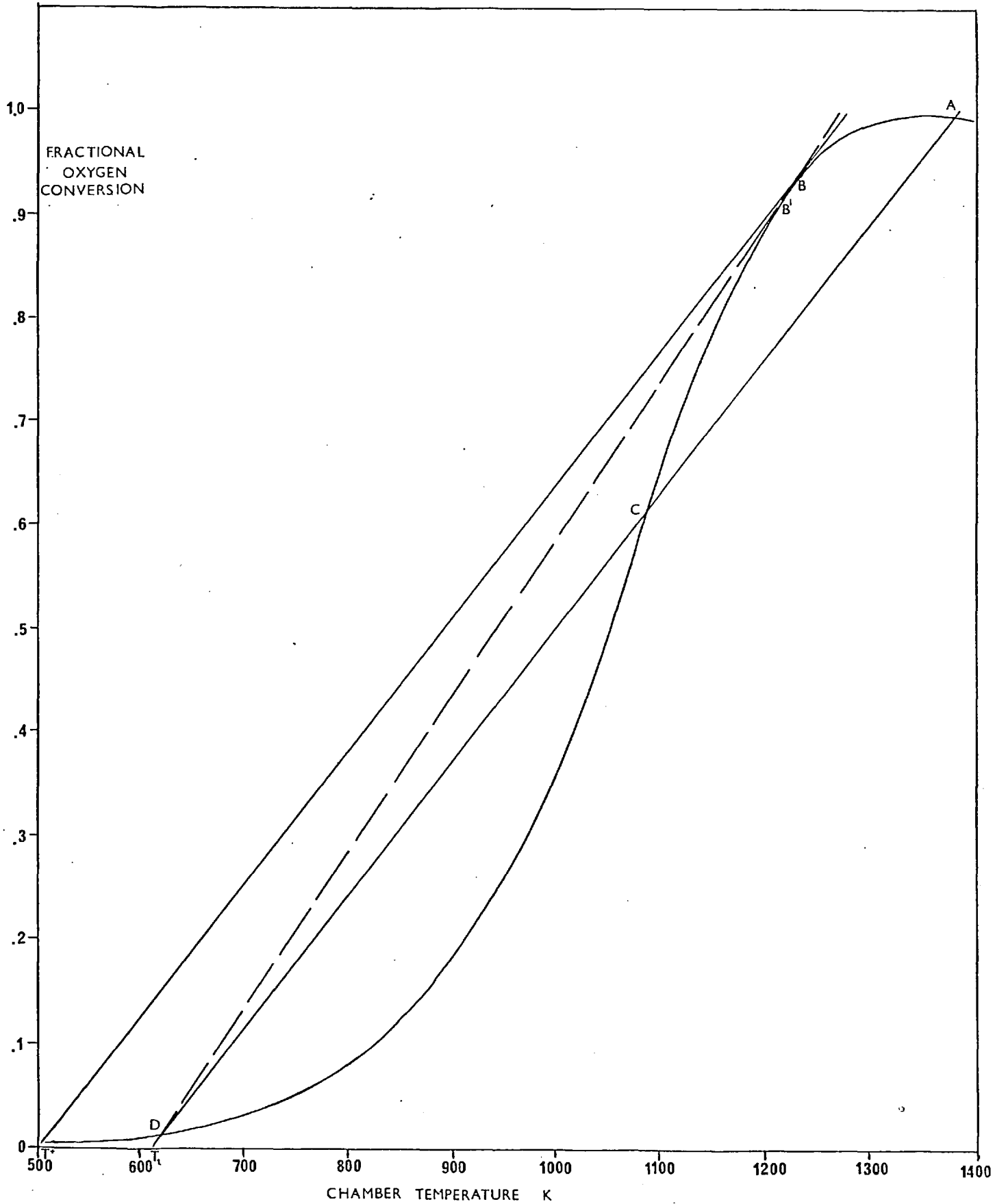


FIG. 4-7 TYPICAL TEMPERATURE-CONVERSION DIAGRAM

are essentially irreversible, the conversion only begins to decrease at very high reactor temperatures. The common reactant in all these combustion reactions is oxygen, hence the oxygen conversion efficiency ϵ_{O_2} is used to indicate the combined extent of all reactions.

Oxygen conversion is defined in two ways depending on operating conditions. For air rich operation ($\phi_2 < 1.0$), ϵ_{O_2} is the oxygen consumed by reactions 4.19, 4.20 and 4.21 as a fraction of that consumed if all the gaseous combustibles were burned to CO_2 and H_2O . For fuel rich conditions ($\phi_2 \geq 1.0$), ϵ_{O_2} is the oxygen consumed as a fraction of the oxygen present.

If the heat losses Q'_L per mole of oxygen from the reaction zone are limited to the heat removal by the return air of flow rate F'_8 ,

$$Q'_L = \frac{F'_8}{x_{O_2} F'_5} \int_{298}^{T_8} C_{p(\text{air})} dT \quad 4.87$$

the enthalpy balance per mole of O_2 is

$$\epsilon_{O_2} \Delta H'_r = \sum H'_{s, \text{products}} - \sum H'_{s, \text{reactants}} + Q'_L \quad 4.88$$

The conversion ϵ_{O_2} as calculated from equation 4.88 is shown as curve T_1A in Figure 4-7. The curve is slightly concave upwards as a result of the increase in gas heat capacity with temperature. The positive slope is characteristic of an exothermic reaction, and is increased by increasing inerts in the feed streams, as indicated by the broken line T_1B . Non-adiabatic conditions are reflected by the fact that the S-curve crosses the $\epsilon_{O_2} = 0$ abscissa at a temperature lower than the combined inlet feed temperature, indicating that some reaction must occur in

order to maintain the reactants at their inlet temperature.

Values of ϵ_{O_2} are calculated using equations 4.85 and 4.88 as follows.

The thermodynamic reference temperature is 298 K and all calculations for the case $\phi_2 \geq 1$ are made per mole of oxygen entering*. The inlet conditions including flow rates, temperatures and compositions are known, and the volume V_r is assumed equal to the geometric volume of the well mixed section. For the purpose of calculation, the net inlet air and combustion gases are combined to form a single inlet stream with temperature T_i .

The first step in the calculation is to assume a reactor temperature equal to T_i . Using this temperature and assuming $\epsilon_{O_2} = 0$, the mean residence time τ is calculated using equation 4.86. Equations 2.26, 2.27 and 4.22 and the corresponding reverse equations are used to calculate $\dot{r}^*$ for each combustible. The oxygen conversion ϵ_{O_2} is calculated using equations 4.19, 4.20 and 4.21 in conjunction with 4.85. The outlet molar flow rate and gas composition are calculated using component mass balances.

It can be seen from the stoichiometry of the combustion reactions that the outlet molar flow rate will not correspond to the inlet flow rate when $\epsilon_{O_2} > 0$. In a perfectly stirred reactor, the prevailing conditions in the reactor correspond to the outlet conditions, hence the outlet conditions are used to recalculate ϵ'_{O_2} which is less than ϵ_{O_2} because of the reduced reactant concentrations. Using the outlet conditions corresponding to ϵ_{O_2} , the

* When $\phi_2 < 1$, calculations are made per mole of oxygen required for complete combustion.

chamber temperature is increased in small increments until the calculated oxygen conversion efficiency ϵ'_{O_2} corresponds to the initial calculated value ϵ_{O_2} . This iteration procedure is satisfactory in all cases where the temperature-conversion curve is concave upwards. At high conversions ($\epsilon'_{O_2} > 0.8$) where the curve becomes convex, the iteration is unstable. At this point, the first approximation ϵ_{O_2} is made using the outlet conditions calculated at the previous point instead of using the inlet conditions.

The oxygen conversion expressed by equation 4.88 is characterised by the temperature and composition of the inlet gas, and can be determined directly as the conversion based on equation 4.85 is calculated above. If the heat generated by reaction to conversion ϵ_{O_2} at chamber temperature T is Q_1 , and the heat required to heat the inlet gas stream from T_1 to T is Q_2 , the corresponding conversion ϵ'_{O_2} can be found using a similar successive approximation procedure. A more direct method of finding ϵ'_{O_2} is to consider the heat generation per unit conversion. If this ratio Q/ϵ_{O_2} is approximately constant, then

$$\epsilon'_{O_2} = \epsilon_{O_2} \frac{Q_2}{Q_1} \quad 4.89$$

It has been found from calculations based on the rigorous method that Q/ϵ_{O_2} is constant for a given set of inlet conditions for $0.2 < \epsilon_{O_2} < 0.9$, and therefore this direct method has been used.

It can be seen both from equations 2.27, 4.22. and curve T_1B' in Fig.4-7 that the addition of spray water as a cooling agent will affect the thermal stability of the

chamber. As the experimental method requires spray water cooling but heat removal for steam generation does not, thermal stability of the latter case is inferred from the experimental results obtained using direct cooling.

4.8.1. Direct Cooling.

Equations 4.85 and 4.88 are satisfied at the points of intersection of curves ABCD and T_1A in Figure 4.7. Points A and D characterise stable conditions while point C indicates an unstable one. The blowout condition occurs at the temperature defined by point B where the two curves are tangent but do not cross. It can be seen that this condition can be achieved by reducing the inlet temperature from T_1 to T^* or by increasing the mole fraction of inert gases, hence increasing the slope of T_1A to become T_1B' indicated by the broken line.

The effect of water spray cooling of the combustion gases is both to shift curve T_1A to the left and to increase its slope. Indirect cooling would only shift the curve to the left. It can be seen from reaction expressions 2.27 and 4.22 that water vapour is not a truly inert species, and a change in its mole fraction will shift curve ABCD vertically. In Figure 4.8 the effect of increasing water vapour mole fraction can be seen to first increase and then decrease the reaction rates of CO and H_2 , while consistently decreasing the reaction rate of CH_4 . The effect of chamber temperature on reaction rate is shown in Figure 4.9 and it can be seen in comparison that temperature effects are much larger than composition effects. The effect of imperfect mixing was not considered in the preliminary analyses (see Section 5-5).

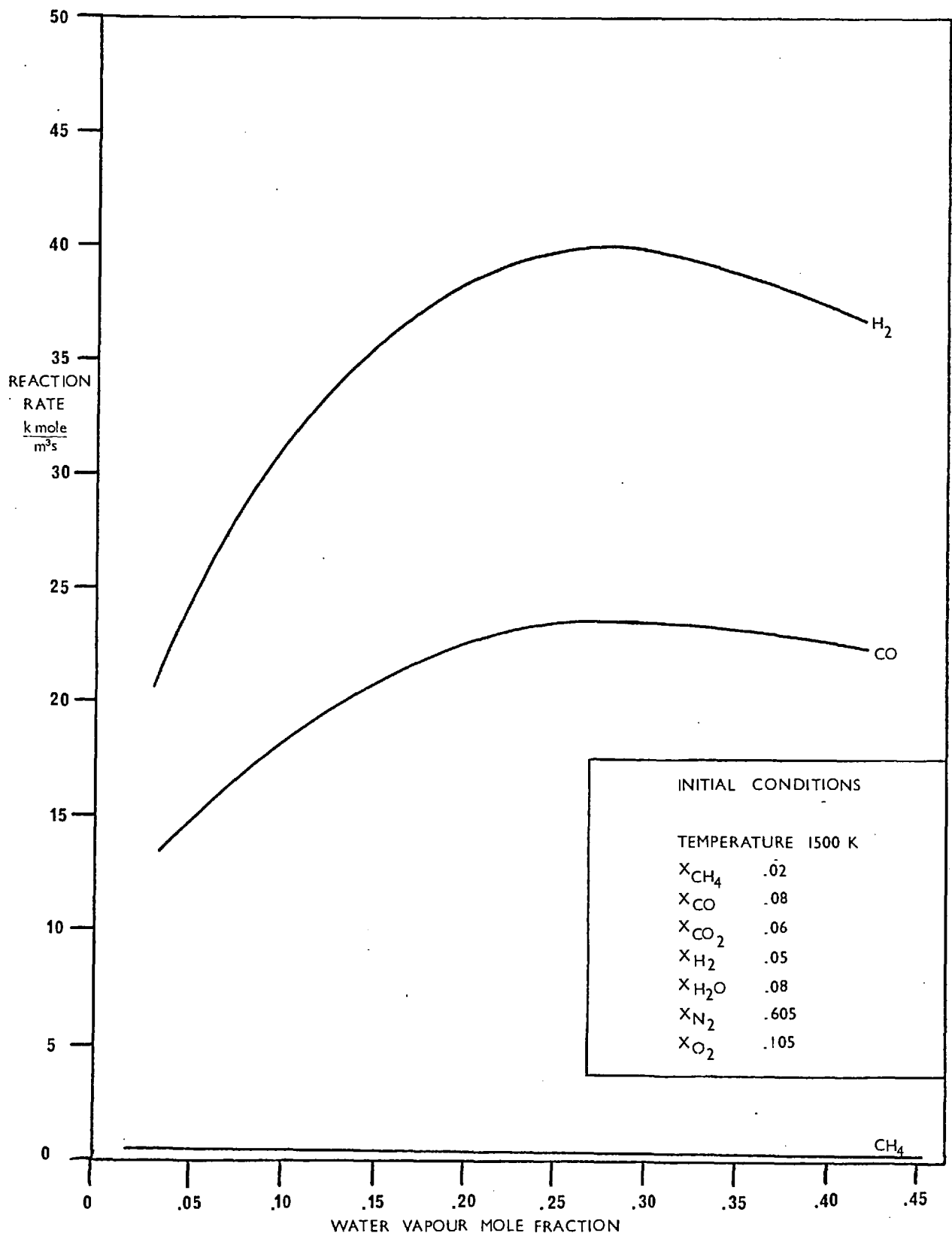


FIG.4-6 THE EFFECT OF WATER VAPOUR MOLE FRACTION ON GASEOUS REACTION RATES

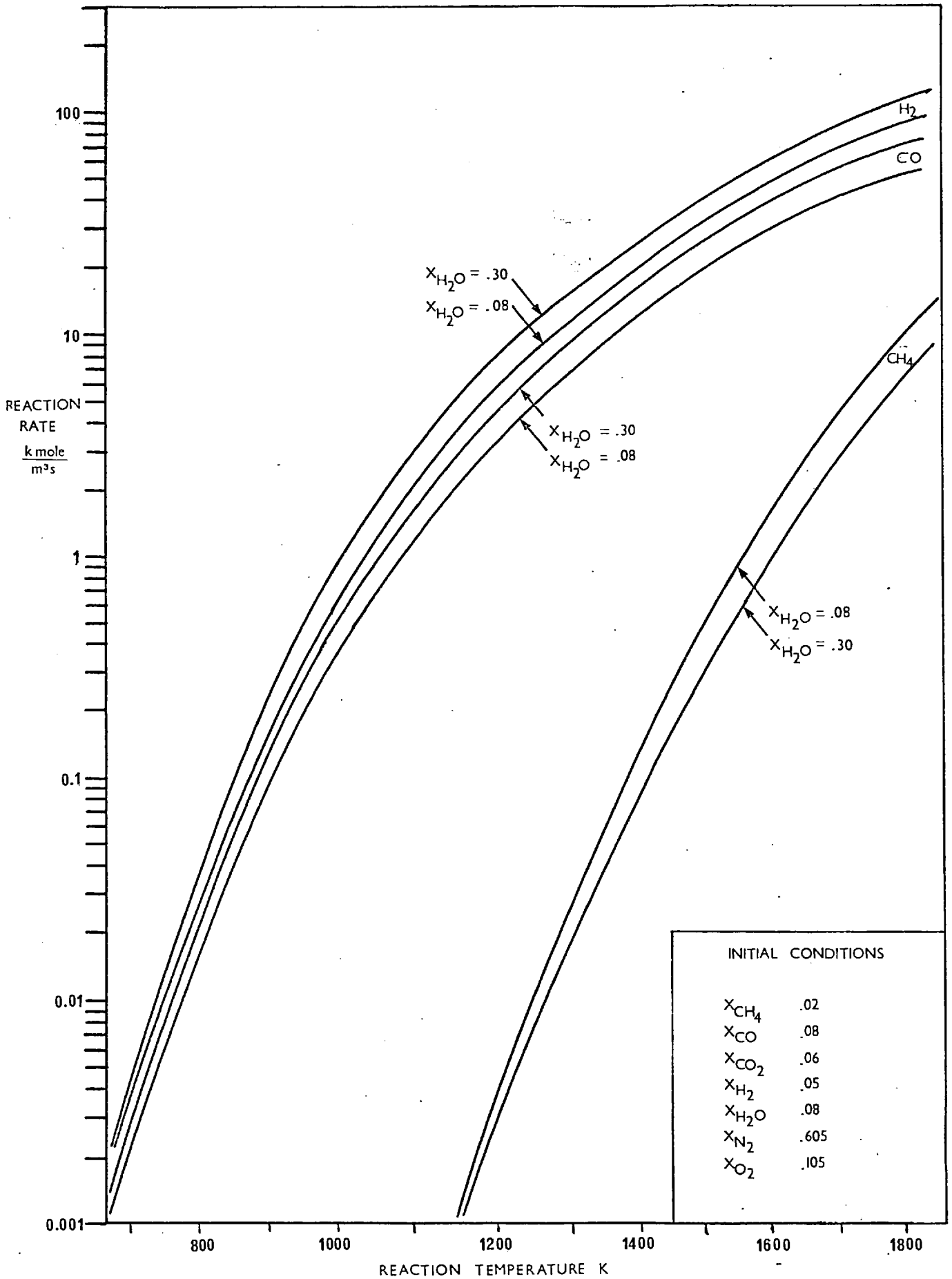


FIG.4-9 THE REACTION RATES OF GASEOUS COMBUSTIBLES

4.8.2. Indirect Cooling.

Although spray water cooling is useful for experimental analyses, heat removal on an industrial scale would be accomplished by indirect heat transfer in the heat exchange zone. Since it is known that water spray cooling will affect the stability characteristics of the chamber, the experimentally measured blowout conditions are interpreted using equations 2.26, 2.27 and 4.22. Once the use of these equations has been justified, maximum heat removal by indirect means can be predicted.

The procedure used is as follows. Blowout was determined experimentally over a wide range of conditions and the maximum interstage heat removal was calculated. These results are tabulated in Appendix 4. The inlet conditions to the second stage were calculated including the spray water in vapour form. Using these conditions and the technique described in Section 4.8, equations 4.85 and 4.88 were solved as a function of chamber temperature. The system model could then be adjusted to correspond within experimental error with the experimental results. The system equations were then applied to a system where heat removal was by indirect means, to predict the maximum heat removal in each case.

4.9. Solids Removal.

Solids extraction from the exhaust gases is required when the solid burden exceeds the statutory maximum of 0.4% by weight of the fuel. In this system, solids removal and exhaust gas cooling were achieved by the use of a venturi-scrubber-cyclone separation system.

For the purpose of economic modelling, the operational characteristics of this removal system were specified. In general, the method of collection of solids is by impingement of the solid particles onto the walls and spray drops in the collection system. The walls are washed and the drops and wash water separated from the cleaned gas stream in a cyclone. Efficient collection requires cooling of the gas stream to such an extent so that the evaporation of all the drops is prevented. Measurements on the stack gases indicated that efficient collection could be achieved if the stack gas temperature were reduced to 350 K. By comparing the amount of spray water used at these conditions with the amount theoretically required to reduce the gas temperature to 350 K, it was found that the scrubber water requirements F_{vw} were 150% of the water necessary to reduce the exhaust gas temperature to 350 K.

$$F_{vw} = \frac{1.5 F_6}{\lambda_w} \int_{350}^T C_{pm} dT \quad 4.90$$

CHAPTER 5.COMPARISON OF EXPERIMENTAL AND PREDICTED RESULTS.

In general, comparisons between experimental measurements and results predicted using models should be made for all available experimental data. Where parts of the model have been derived empirically, comparisons are made using statistical tests showing either the standard deviation or the error at the 95% confidence level. It should be noted that where results have been compared graphically, absolute differences between experiment and model as opposed to percentage differences have been plotted.

The program used for all empirical correlations in this work was of the multi-variate non-linear type obtained from the SHARE (72) program library and reprogrammed in Fortran IV for use on IBM-7094 and CDC-6600 computers. The program was capable of handling weighted observations as well as exponential and logarithmic variables. The program was successfully tested on known non-linear equations.

Some care was required in the use of the program as tests for significance of variables were made by comparing the error variance of the dependent variable before and after inclusion of each independent variable in the regression equation using the F-test.

It was necessary to specify the value of F above which a variable would be included in, and below which removed from the regression equation. To make the program applicable for all conditions, this value of F was set equal to 1.0 corresponding to infinite degrees of freedom. The regression variables were then re-analysed making use of F tables corresponding to a 0.5% confidence level.

5.1. Gas Composition and Temperature.

The prediction of gas composition and temperature using the mechanistic model described in Section 4.4.1 is based on the attainment of water gas shift equilibrium. Methane has been predicted using equation 4.4. Measured (3) and predicted wet gas compositions are compared in Fig.5-1. The scatter of some of the results is explained by total mass balance errors in the reported gas compositions from the first stage chamber (2,3). It has been found that these errors are as large as 25% for H_2 and H_2O (3) and slightly less for CO and CO_2 (2). Where there are no mass balance errors in the measured compositions, i.e. from the second stage, the measured and predicted values correspond.

Measured and calculated combustion gas temperatures are compared in Fig.5-2 for equivalence ratios greater than 0.9. Where $\phi < 0.9$, the heat balance iteration used in the model based on water gas shift equilibrium fails to converge. The error associated with the experimental measurements (1) is ± 24 K whereas the errors associated with the predicted results may be as large as ± 25 K as indicated in Appendix 5. Within these error limits, all but three of the predicted temperatures correspond with experimentally measured ones. The sources of error in the predicted results as reported by ARCHER (3) were eliminated by the use of temperature dependent heat capacity expressions and by improved methods of estimating first stage solids.

5.2. Solid Formation in the First Stage.

The regression equation for soot m_s formed in the first stage is

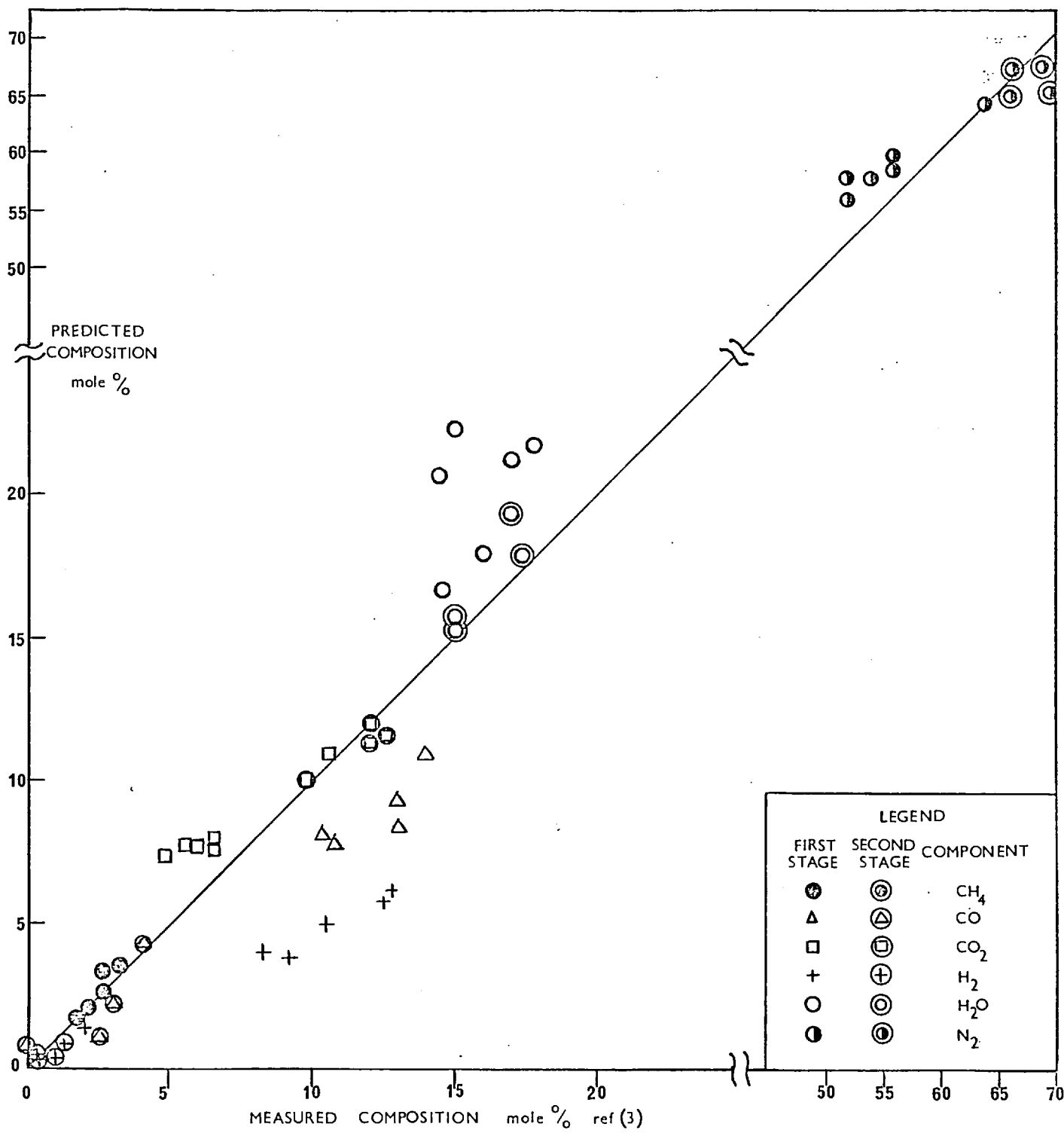


FIG.5-1 COMPARISON OF MEASURED AND PREDICTED WET GAS COMPOSITIONS

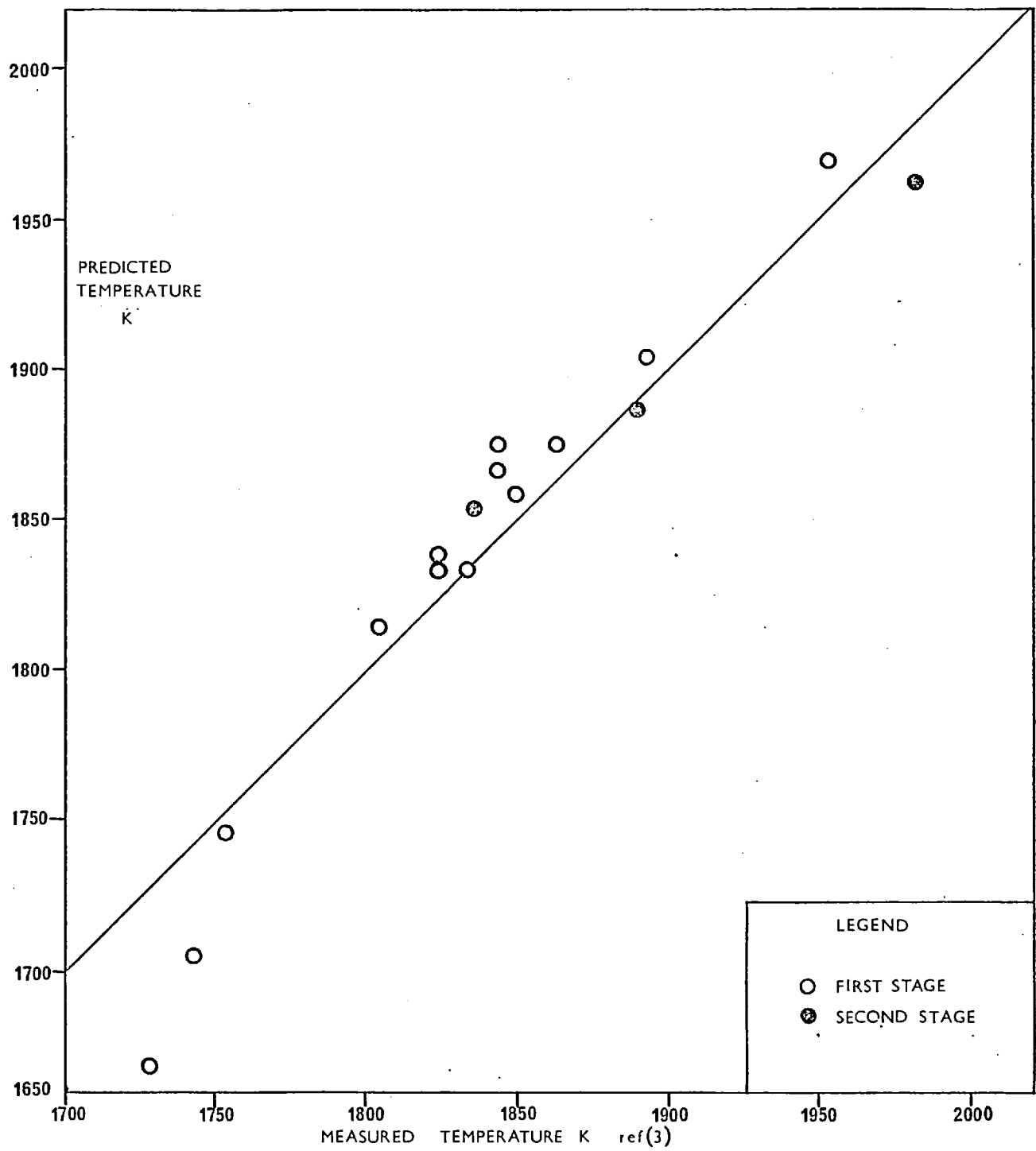


FIG. 5-2 COMPARISON OF MEASURED AND PREDICTED COMBUSTION GAS TEMPERATURES

$$m_s \pm 0.0061 = F_1^2 (0.0184 - 0.161 \times 10^{-4} T) \\ + 0.417 \times 10^{-4} F_1 T - 0.1132 \frac{\text{kg solid}}{\text{kg fuel}} \quad 5.1$$

The standard deviation is ± 0.0030 , the error in m_s is expressed above at the 95% confidence level for 25 degrees of freedom.

The first stage solids measured by HTUN (2) for both air and steam atomisation have been compared with the results predicted using the mechanistic model developed in Section 4.5 in conjunction with equation 5.1. The results are tabulated in Table A4-1 in Appendix 4 and are compared graphically in Fig.5-3. The 95% confidence limits in Fig.5-3 are indicated by the broken lines.

The amount of soot formed was not significantly affected by equivalence ratio, reverse/forward air flow rates or [OH] concentration. As most of the experimental measurements were made for a constant air flow rate, the effect of equivalence ratio is expressed by the fuel flow rate terms. The effect of [OH] concentration may be included in the temperature terms since [OH] concentration is increased by increasing steam flow rate which also reduces temperature.

Three of the experimental results indicated by (*) in Table A4-1 were considered to be in error as a result of collection inefficiency and were ignored when constructing the model.

Detailed comparison with respect to particle mass-size distributions, between the solids collected and those predicted using the model is not possible. A theoretical mass distribution of solids from the first stage is shown

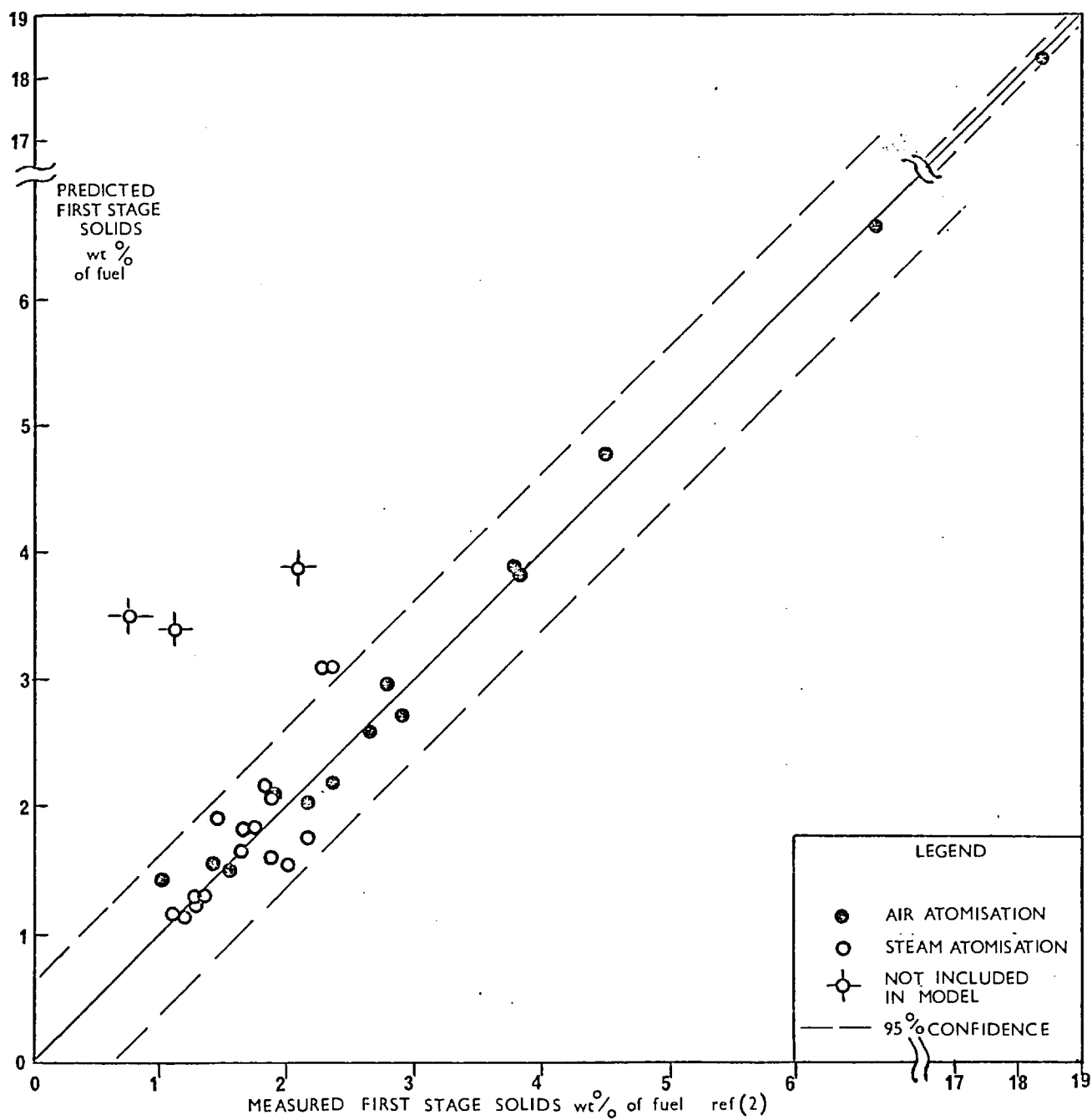


FIG. 5-3 COMPARISON OF MEASURED AND PREDICTED FIRST STAGE SOLIDS

in Fig.5-4. This distribution, shown as the solid curve, represents approximately twice as much soot (first peak) as char (second peak).

It is noted that no particles larger than $10\text{ }\mu\text{m}$ were found during photographic examination of the samples (3) which compares favourably with the predicted size distribution which indicates that less than 2.5% of the char particles formed were larger than $15\text{ }\mu\text{m}$. No cenospheres were found by photographic analysis of the first stage solids, hence their formation is not considered.

5.3. Heat Transfer.

The heat transfer rates measured in the heat transfer section between the first and second stages are compared in Fig.5-5, with those calculated from the models developed in Section 4-7. The initial comparison is indicated by the open circles. It was found that the best possible correlation was achieved using the Colburn equation for convective heat transfer and equation 2.49 to predict soot emissivity with $k_b = 550\text{ m}^{-1}\text{ K}$. It can be seen that there is a difference between the measured and predicted heat transfer rates, the predicted results being 50-90% too high. A possible source of this error is the under-estimation of cooling tube wall temperature as discussed in Appendix 5.

To improve the correlation, empirical means were used in conjunction with the mechanistic model. Using regression analysis, the absolute differences $\Delta\dot{Q}$, between the measured heat removal rates $\dot{Q}_m$ and those predicted using models $\dot{Q}_p$ were correlated with furnace operating conditions. The resulting correlation is given by equation 5.2

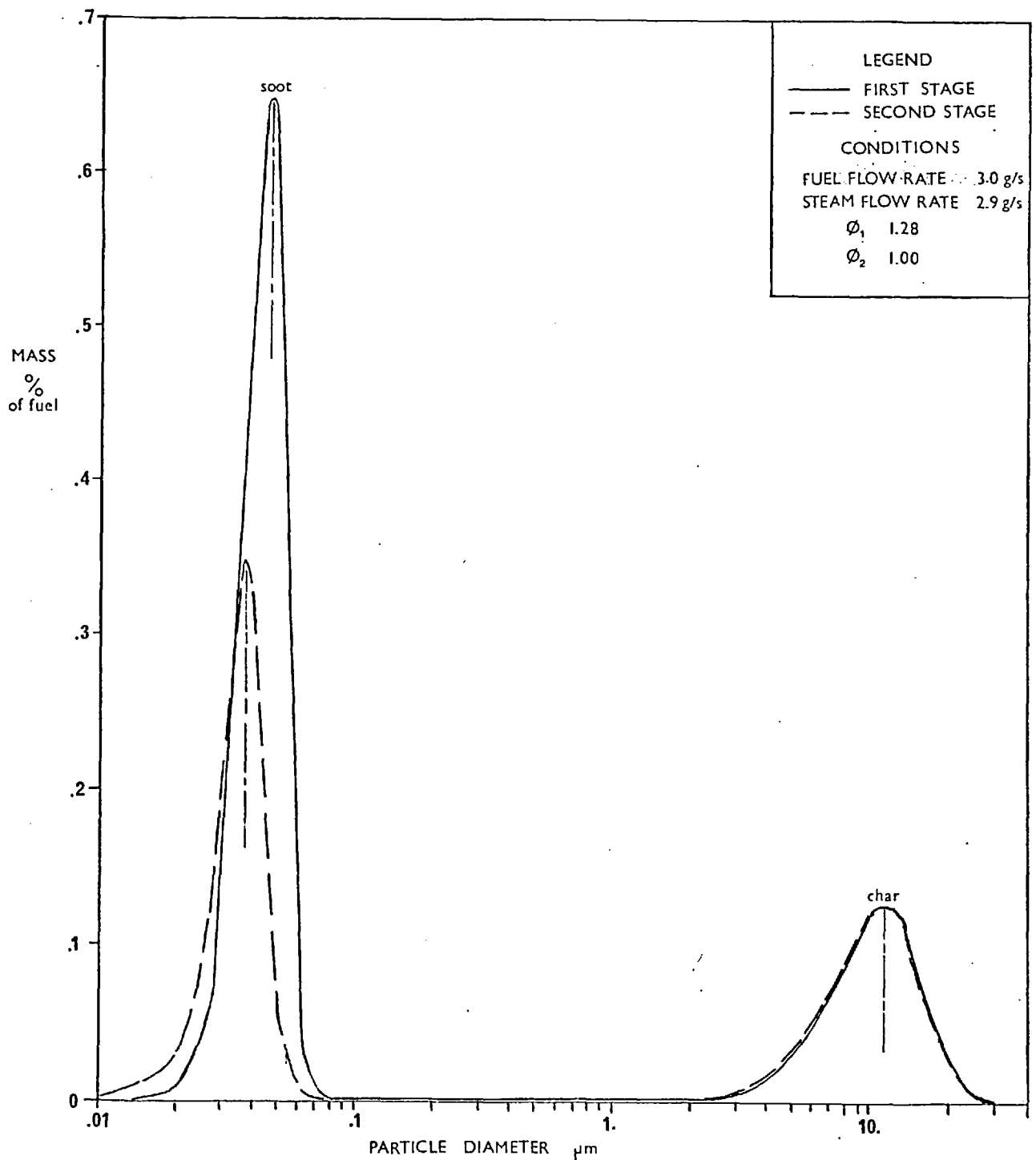


FIG. 5-4 MASS DISTRIBUTION OF SOLIDS FROM FIRST AND SECOND STAGES

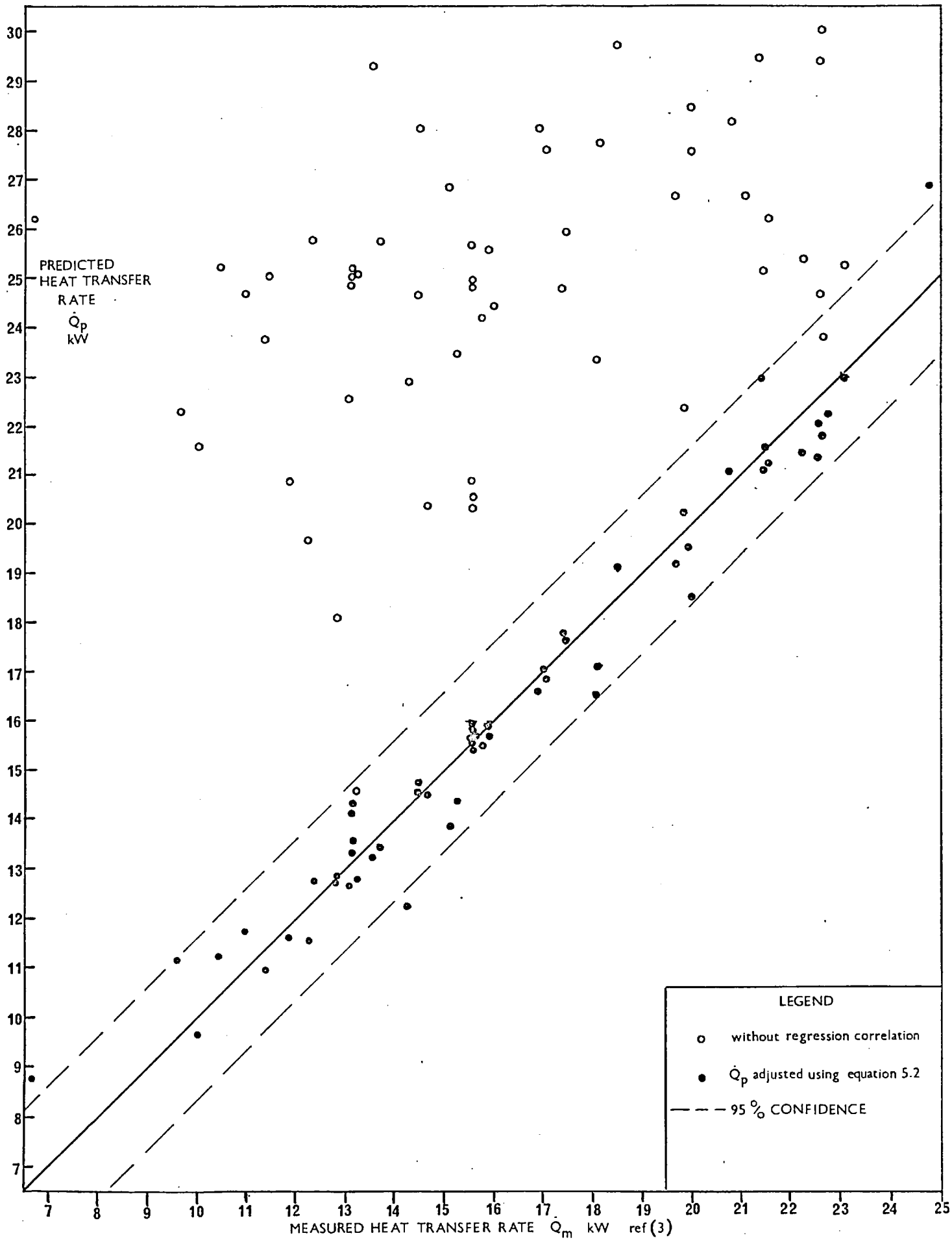


FIG.5-5 COMPARISON OF MEASURED AND PREDICTED INTERSTAGE HEAT TRANSFER RATES

$$\Delta \dot{Q} \pm 1.63 = \dot{Q}_p - 0.0546 T_{IN} + 0.0620 T_{OUT} + 4.972 \text{ kW}$$

5.2.

where the standard deviation is ± 0.81 and the error in $\Delta \dot{Q}$ above is given at the 95% confidence level for 52 degrees of freedom.

Solid burden in the gas stream did not significantly affect $\Delta \dot{Q}$. The strong dependence on temperature indicates that the use of T_{cm} or errors in the estimation of the wall temperature may be the cause of the differences. The correlation is shown in Fig.5-5 by the solid circles. The 95% confidence limits are shown by the broken lines.

5.4. Solid Burning in the Second Stage.

Previously (3), the quantity of carbon burned in the second stage was determined by subtraction using equations 3.1 and 3.9. The improved method of first stage solid prediction has allowed the direct comparison of the predicted first stage solids with the measured second stage solids. First and second stage solid burdens are compared in Table A4-2. The errors associated with these results are discussed in detail in Appendix 5. The amount of solid present in the second stage exhaust shown in Table A4-2 includes the solids which were not removed in the venturi-scrubber system. These solids were estimated as detailed in Appendix 7, to be a maximum of 0.0007 kg per kg fuel. This amount was added to the measured second stage solids.

Solid burnout in the second stage was predicted using the model developed in Section 4.6. Good agreement between measured and predicted second stage solid burdens could not be achieved by consideration of the gas phase and solid reactions consecutively.

If the carbonaceous solids were allowed to react in parallel with the gaseous combustibles, it was found that some of the soot was burned during the 3-10 ms that the system required to reach water gas shift equilibrium. Using time increments of 0.2 ms increasing to 2 ms after an elapsed time of 2 ms, the reactions were allowed to proceed from inlet concentrations to water gas shift equilibrium conditions. The progress of these reactions under typical operating conditions is shown in Fig.5-6. The rapid decrease in solids indicates the period of soot combustion.

It can be seen from Table A4-2 that the second stage solids predicted using the combined mechanistic and empirical models are the same magnitude as those measured. Most results correspond within experimental error, which in some cases is very large. A more detailed analysis and explanation of these errors is given in Appendix 5.

In Fig.5-4, the mass distribution of solids from the second stage at $\phi_2 = 1.0$ is indicated by the broken line. The corresponding first stage solids are indicated by the solid line. It can be seen that the soot has burned appreciably whilst the char has not. In this example, where overall carbon combustion efficiency is 36%, 53% of the soot but only 2% of the char has been burned.

Although some cenospheres were identified in the second stage solids, they comprise a small fraction of the total mass of solids and have been included with the char when considering solid burning.

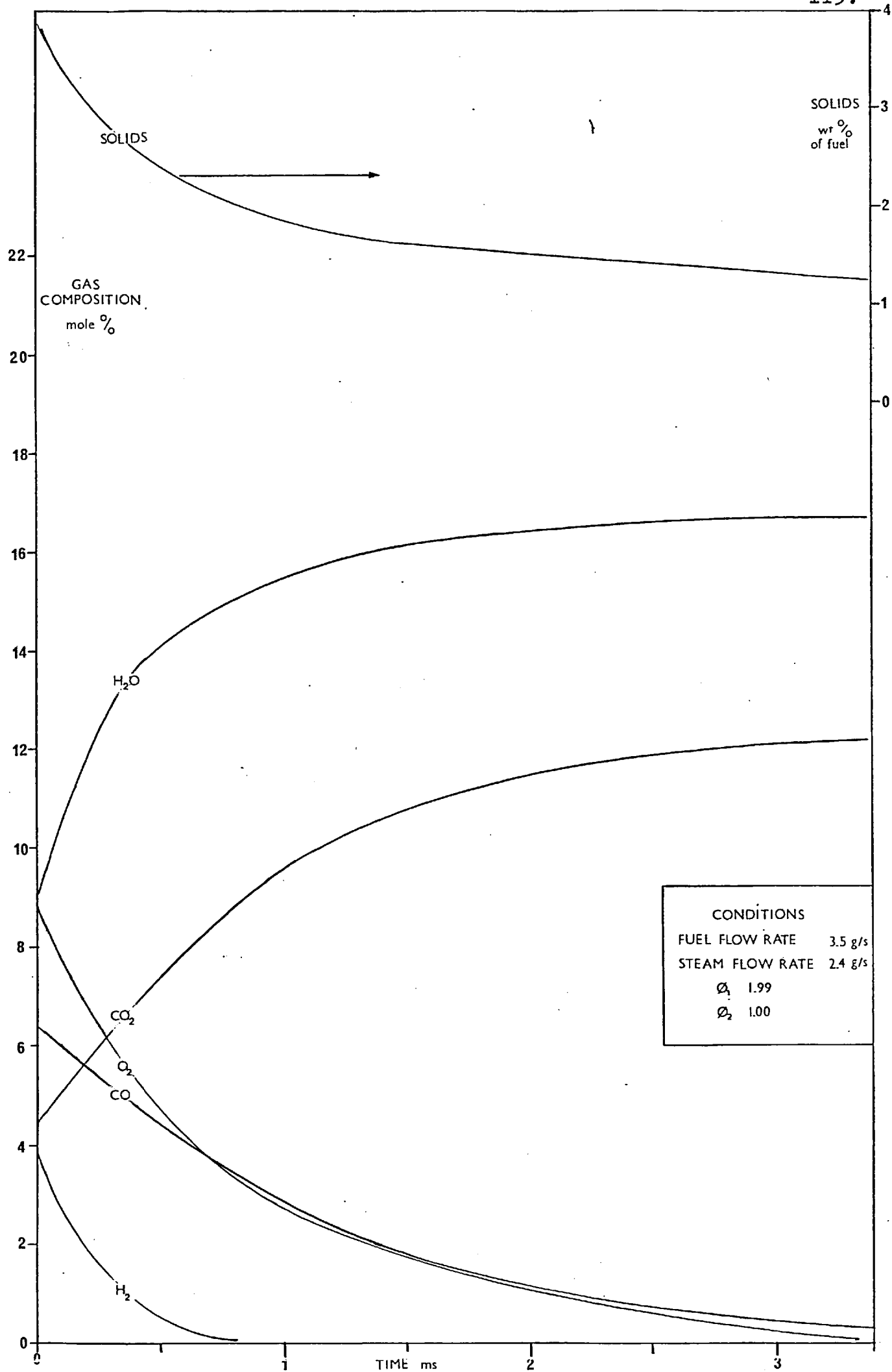


FIG. 5-6 PROGRESS OF SECOND STAGE COMBUSTION REACTIONS

5.5. Thermal Stability.

The experimental results from the blowout measurements were used to determine the operating limits of the second stage chamber and are shown in Tables A4-3 and A4-4. Using the method described in Section 4.8, the inlet gas stream temperature and combustion chamber temperature as a function of oxygen consumption efficiency were calculated. A typical curve based on experimental operating conditions is shown in Fig.5-7.

It can be seen that the theoretical curve ACD and straight line T_1A based on these experimental blowout conditions do not predict the blowout condition, but indicate stable operation at point A. The difference between the measured and predicted blowout condition cannot be entirely explained either by experimental errors, or by errors in the choice of constants in equations 2.26, 2.27 and 4.22.

The second stage temperature was not measured experimentally at the blowout condition, but was calculated using the measured interstage heat removal and calculated gas enthalpy at the cross piece. The most significant experimental error is the uncertainty in the spray water flow rate, which amounts to about $\pm 5\%$ of the spray water or about $\pm 5\%$ of the interstage heat removal. This error can be related to Fig.5-7 by shifting line T_1A by a maximum of $\pm 50K$. These errors are discussed in detail in Appendix 5.

In order to explain the differences between the experimental and theoretical results, the aerodynamics of the well mixed section must be considered. Work on mixing limitations in imperfectly stirred combustors (42) has shown that mixing intensity is related to inlet jet design by

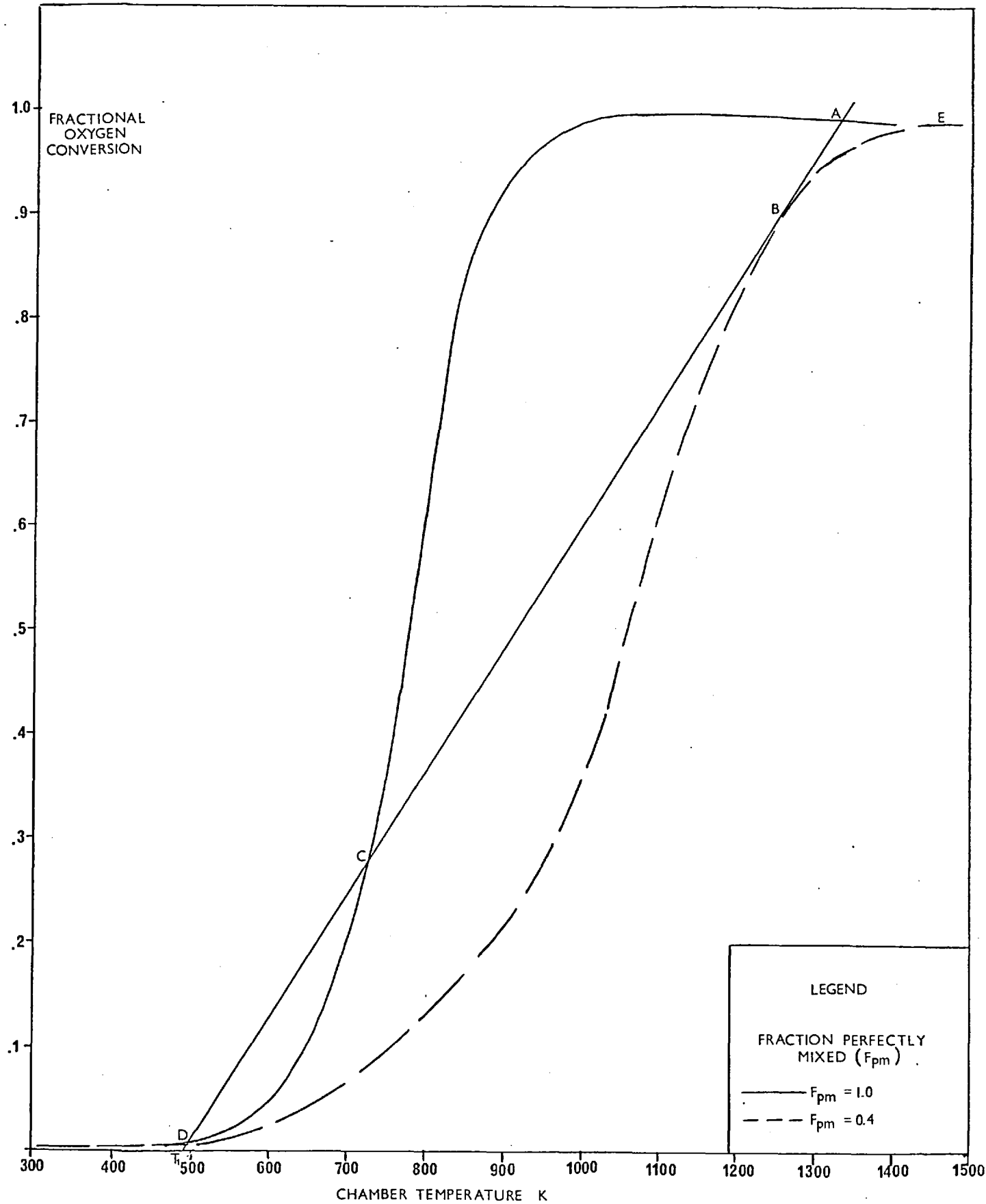


FIG.5-7 THE EFFECT OF MIXING ON COMBUSTION STABILITY

equation 2.41. In the well mixed section of the second stage chamber, the inlet jet design parameter is only 3.2, which is an order of magnitude smaller than values quoted in the literature for high intensity equipment. This is because the high velocity inlet air jets are recessed in such a way that considerable energy is dissipated by impaction of the air on the bend of the air inlet port (see Fig.3-3). This results in a lower degree of turbulence in the chamber, and consequently, the total volume of the well mixed section does not approach the perfectly stirred micromixed condition. The mixing can be improved by placing the air jets so that they project directly into the chamber (see Fig.3-3). To allow for the effects of imperfect mixing in the theoretical analysis, the mixing term $\beta_T \tau$ defined by equation 2.41 was reduced by approximating the well mixed section to a perfectly stirred volume less than the geometric volume. It can be seen that this approach reduces the mean residence time τ , but not the mixing intensity β_T , the reaction rate however, responds in the desired manner. The fraction of the geometric volume that was assumed to be perfectly micromixed was denoted as F_{pm} . The values of F_{pm} required for each operating condition to achieve agreement between the measured and predicted blowout conditions are shown in Table A4-5.

A corrected reaction rate curve is shown as the broken line DBE in Fig.5-7. The values of F_{pm} were analysed using regression techniques to determine their relation to such operating conditions as inlet velocity, kinetic energy and momentum. No meaningful correlation was obtained, and it was concluded that the use of the F_{pm} correction embodied undetermined errors as well as the mixing

limitations.

The maximum interstage heat removal rate $\dot{Q}_R$ determined experimentally using the spray cooling probe has been related to furnace operating conditions viz.

$$\dot{Q}_R \pm 1113 = 29290 - 8750 \phi_1 + 454 L_A + 7610 F_{pm} \frac{\text{kJ}}{\text{kg fuel}} \quad 5.3$$

where L_A is the air loading term expressed as mole of air per mole of combustible gas entering the second stage chamber. The standard deviation is ± 537 , the error in $\dot{Q}_R$ above is expressed at the 95% confidence level for 21 degrees of freedom.

The simple linear correlation with ϕ_1 is

$$\dot{Q}_R \pm 2190 = 37759 - 11700 \phi_1 \frac{\text{kJ}}{\text{kg fuel}} \quad 5.4$$

The standard deviation is ± 1060 , the error in $\dot{Q}_R$ above is expressed at the 95% confidence level for 23 degrees of freedom.

The experimental results shown as the open circles are plotted as a function of first stage equivalence ratio in Fig.5-8 for $0.95 < \phi_2 < 1.05$. Equation 5.4 is shown as the broken line.

It can be seen from Fig.5-8 that the maximum interstage heat removal increases as ϕ_1 decreases. The lowest value of ϕ_1 investigated experimentally under blowout conditions was 1.28, but it can be seen from theoretical considerations that when $\phi_1 = 1.0$, no combustible will enter the second stage chamber, hence the flame will be extinguished for all rates of interstage heat removal. The locus of the maximum heat removal as a function of ϕ_1 must approach zero as ϕ_1 approaches 1.0. This is shown by the dotted lines in

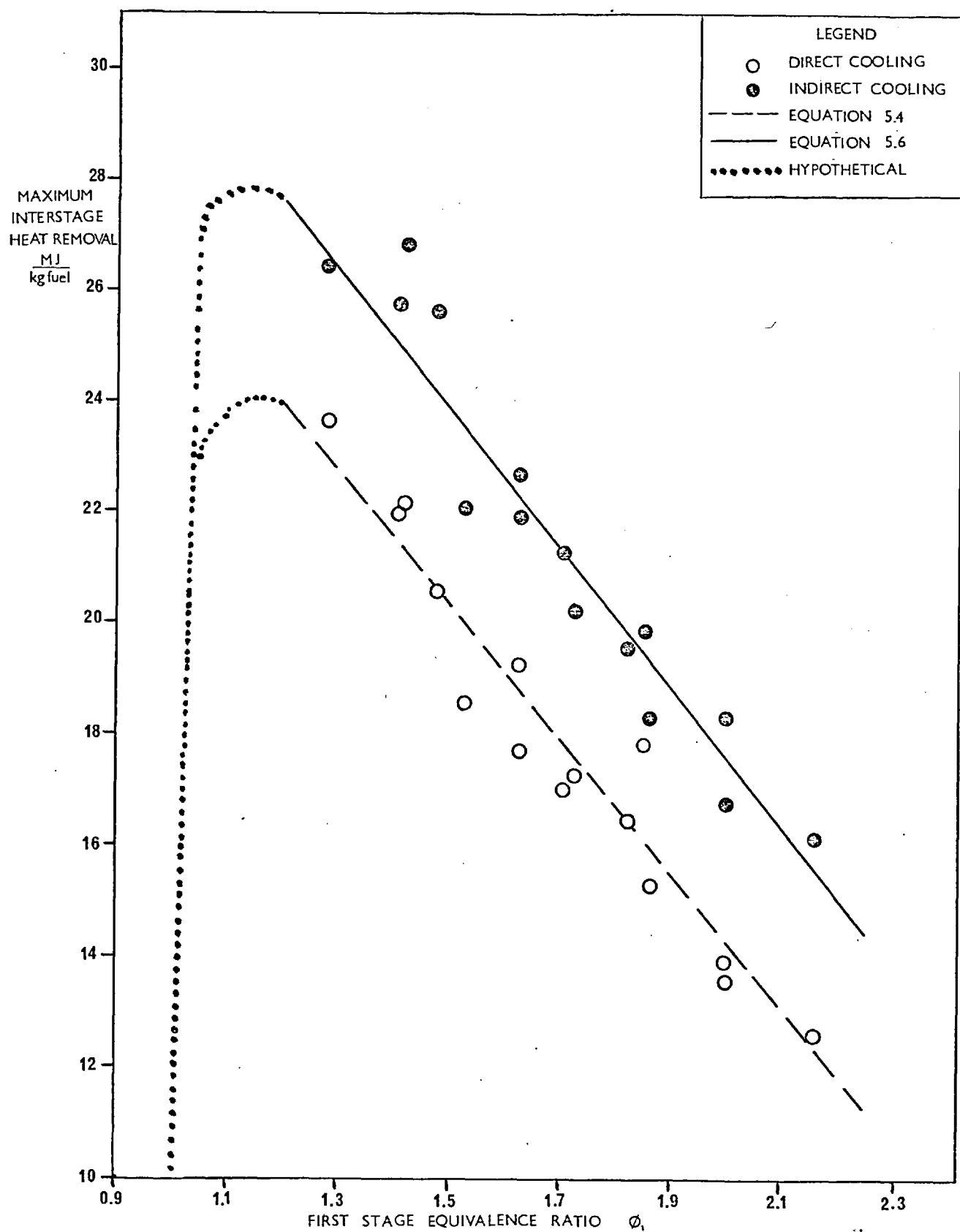
FIG.5-8 MAXIMUM INTERSTAGE HEAT REMOVAL FOR $0.95 < \phi_2 < 1.05$

Fig.5-8. The resulting peak in the curve has not been verified experimentally.

The maximum interstage heat removal that may be achieved by indirect cooling only (i.e. without spray water cooling) was calculated by the method described in Section 4.8. It can be seen from Fig. 4-8 that a decrease in water vapour mole fraction may increase or decrease overall reaction rate for a fixed reaction temperature. The decrease in water vapour in the inlet combustion gas will decrease the slope of line T_1A in Fig.5-7. In order to achieve blowout, therefore, the inlet temperature must be reduced since the curve ABCD does not vary appreciably with water vapour mole fraction.

To calculate the blowout conditions achieved by indirect cooling alone, at conditions corresponding to the experimentally determined ones, the value of F_{pm} in each case has been reduced so that the mean residence time in the perfectly mixed section is the same for indirect cooling conditions as that for water spray cooling. That is, it was reduced in the ratio of the total mass flow rates. The values of F_{pm} used in each case are given in Table A4-5.

This calculation involves a trial and error procedure whereby a value of interstage heat removal is assumed. If the heat generation and heat removal curves are not tangent, the heat removal is adjusted and the calculation repeated until the curves become tangent.

The calculated results for indirect cooling are shown as the solid circles in Fig.5-8. The maximum interstage heat removal $\dot{Q}_R'$ achieved by indirect means has been related to furnace operating conditions by equation 5.5

$$\dot{Q}_R' \pm 1648 = 49330 - 14400 \phi_1 - 166 v_i \frac{\text{kJ}}{\text{kg fuel}} \quad 5.5$$

where v_i is inlet combustion gas velocity (m/s). The standard deviation is ± 799 , the error in $\dot{Q}_R'$ above is expressed at the 95% confidence level for 22 degrees of freedom. The dependence on inlet velocity is as expected from theory (42).

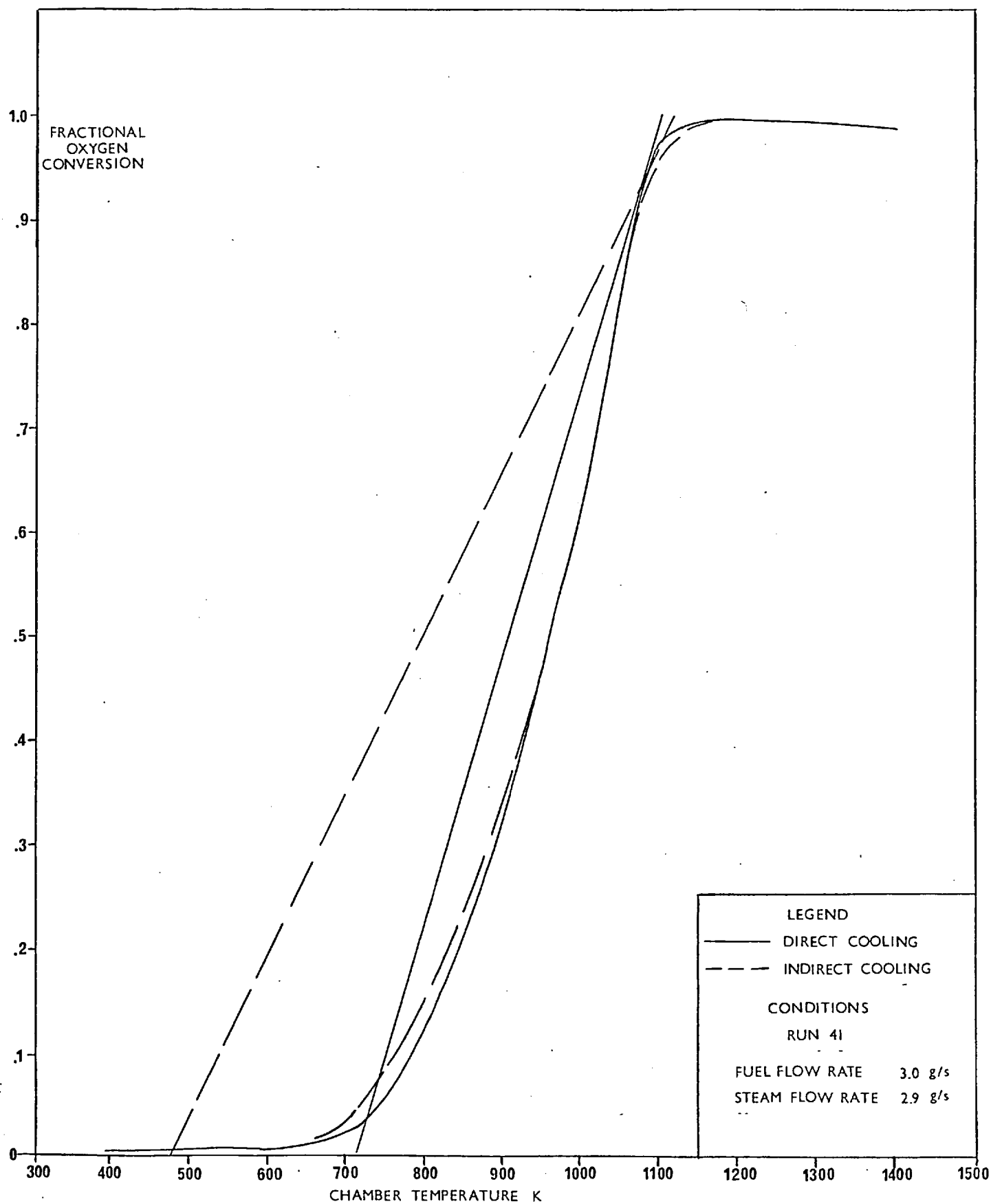
The best linear correlation with ϕ_1 is

$$\dot{Q}_R' \pm 1964 = 42464 - 12500 \phi_1 \frac{\text{kJ}}{\text{kg fuel}} \quad 5.6$$

and is plotted in Fig.5-8 as the solid line. The standard deviation is ± 927 , the error in $\dot{Q}_R'$ above is expressed at the 95% confidence level for 23 degrees of freedom. The coefficients for ϕ_1 in equations 5.4 and 5.6 are not significantly different.

Two typical temperature-conversion diagrams for direct and indirect cooling are shown in Fig.5-9 and Fig.5-10. In both cases solid lines indicate direct cooling, broken lines correspond to indirect cooling. The second stage equivalence ratio is 1.0 in both cases, whilst $\phi_1 = 1.28$ in Fig.5-9 and $\phi_1 = 2.00$ in Fig.5-10. Considering Fig.5-9 first, it can be seen that the oxygen conversion at blowout for both direct and indirect cooling is about 0.95. The low temperature rise of 600 K from inlet to combustion temperature for the indirect cooling case is a result of relatively high interstage heat removal at low ϕ_1 .

In fig.5-10, the oxygen conversion at blowout is only about 0.85. This reduction is a result of the lower mean residence time in the well mixed section of the chamber since the mass flow rate and chamber temperature are higher. The temperature rise in this case is about 1050 K as a result

FIG.5-9 TEMPERATURE-CONVERSION DIAGRAM AT $\phi_1=1.28, \phi_2=1.00$

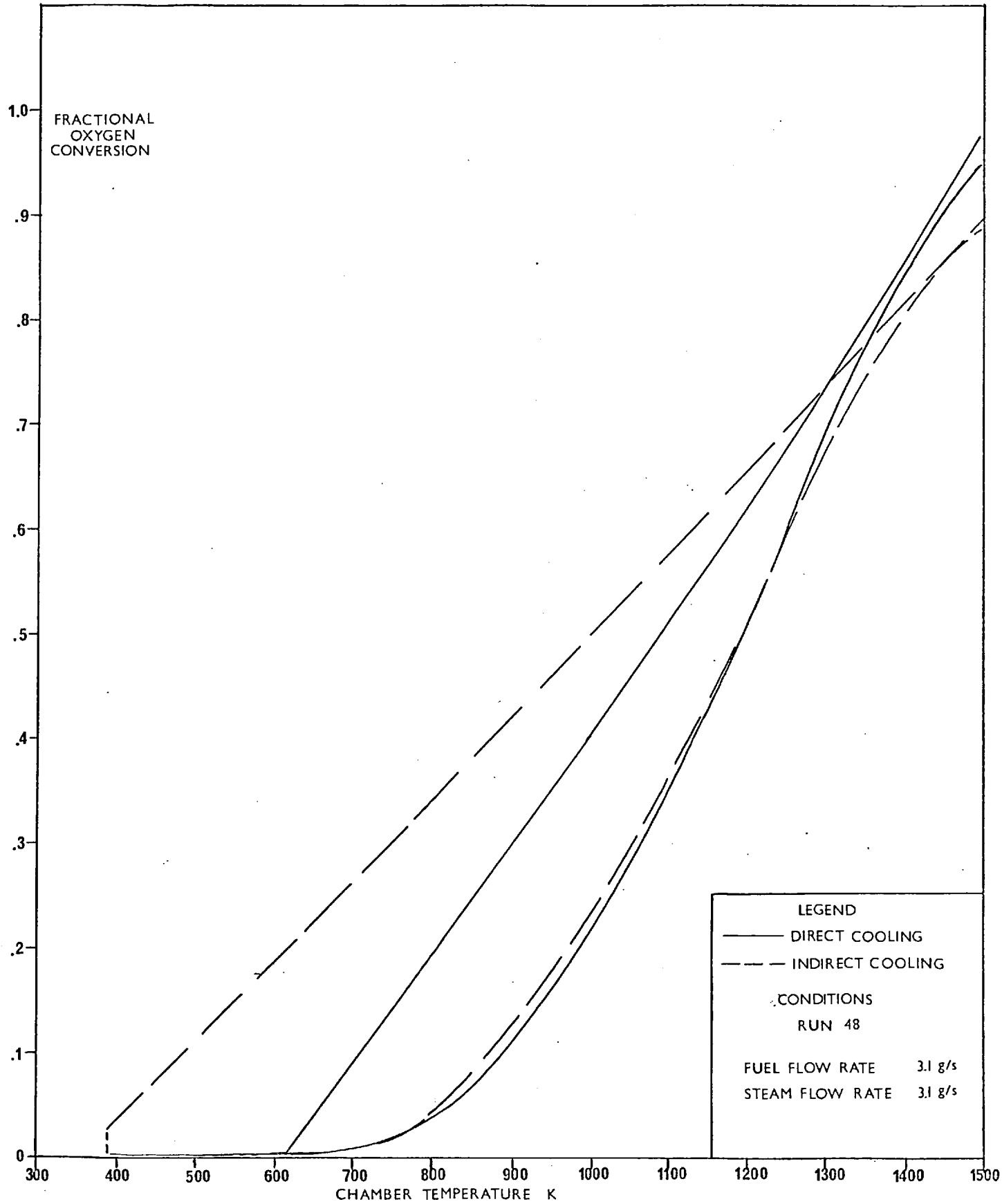


FIG.5-10 TEMPERATURE- CONVERSION DIAGRAM AT $\phi=2.00$, $\phi=1.00$

of the lower interstage heat removal at higher ϕ_1 . The heat removal curve for the indirect cooling case rises vertically at 373 K as some of the water vapour in the inlet gas has been condensed, and must be vapourised before the temperature of the gas begins to rise.

CHAPTER 6.

SYSTEM OPTIMISATION.

The general complexity of this combustion system indicates that the evaluation of any objective function will be time consuming. It is also evident that the system models constructed in Chapter 4 are highly non-linear with respect to most system variables. The choice of optimising techniques is limited, therefore, by the requirement that the method be applicable to non-linear systems and be rapidly convergent in terms of the number of function evaluations required to attain the optimum.

Several minimisation techniques were investigated, however the most promising method is that due to MURTAGH and SARGENT (54) which has quadratic convergence characteristics and can be used with linear and non-linear constraints.

6.1. Objectives.

The principal objective of the system optimisation is the determination of the operating conditions that allow maximum annual profit arising from the sale of high and low pressure steam. Calculations based on the equipment and raw materials costs shown in Tables 4-1, 4-2, and 4-3, indicated however, that for a small installation of the size studied, the capital and operating costs of ancillary equipment were so high that the marginal cost of the system always exceeded the marginal return, and that even when labour costs were ignored, the annual operating costs and plant burden exceeded the revenue from steam production at all operating conditions. In the light of these findings, studies have

been made to minimise annual operating losses for the two stage system. A similar study has been made on the combined first stage and heat exchange section but excluding labour costs, plant burden and the solid burden constraint. The conditions yielding minimum first stage solid emission have also been determined.

System operation has also been studied to determine the maximum first stage intensity and optimal heat transfer surface required for two stage operation conforming to the solid emission limitation of 0.0040 kg solids/kg fuel.

The advantages of air or steam atomisation have been compared, and the effects of combustion chamber size on system performance have also been evaluated.

6.2. General Minimisation Program.

This method of minimisation utilises the assumption that the objective function can be approximated by a quadratic over a small region. If this approximation is valid, the function $f(\underline{x})$ in n variables with continuous second derivatives may be expanded about the point k using a Taylor series.

$$\begin{aligned} \Delta f(\underline{x}_k) &\equiv f(\underline{x}_k + \Delta \underline{x}) - f(\underline{x}_k) \\ &= \sum_{j=1}^n \left(\frac{\partial f_k}{\partial x_j} \right) \Delta x_j + \frac{1}{2} \sum_{j=1}^n \sum_{i=1}^n \left(\frac{\partial^2 f_k}{\partial x_j \partial x_i} \right) \Delta x_j \Delta x_i + O(\Delta \underline{x}^3) \end{aligned}$$

6.1

where $f_k \equiv f(\underline{x}_k)$ and $O(\Delta \underline{x}^3)$ represents all higher order terms of the expansion. If the quadratic approximation is exact, the higher order terms will vanish. The term $\Delta \underline{x}$ is a column vector of small displacements

$$\partial \underline{x} = (\partial x_1, \partial x_2, \dots, \partial x_n) \quad 6.2$$

Equation 6.1 can be written in terms of a displacement vector $\underline{p}_k$

$$\underline{p}_k = \underline{x}_k - \underline{x}_{k-1} \quad 6.3$$

a gradient vector $\underline{g}_{k-1}$ evaluated at the point $k-1$

$$\underline{g}_{k-1} = \sum_{i=1}^n \left(\frac{\partial f_{k-1}}{\partial x_i} \right) \quad 6.4$$

and the Hessian matrix $\underline{H}$ which is a $n \times n$ matrix of second partial derivatives of f_k , hence

$$f_k = f_{k-1} + \underline{g}_{k-1}^T \underline{p}_k + \frac{1}{2} \underline{p}_k^T \cdot \underline{H} \underline{p}_k + O(\partial x^3) \quad 6.5$$

The change in gradient between points $k-1$ and k is defined by $\underline{q}_k$ where

$$\underline{q}_k = \underline{g}_k - \underline{g}_{k-1} = \underline{H} \cdot \underline{p}_k \quad 6.6$$

At maxima and minima, the gradients become zero, hence if $\underline{x}^*$ represents the co-ordinates of a local minimum at point k , equation 6.6 becomes

$$\underline{x}^* = \underline{x}_{k-1} - \underline{H}^{-1} \cdot \underline{g}_{k-1} \quad 6.7$$

To distinguish between a minimum and other maxima or points of inflection, the term $\underline{p}_k^T \cdot \underline{H} \underline{p}_k$ embodying the second derivatives is evaluated. Where a minimum occurs, this term is positive and the Hessian matrix is said to be positive definite.

The only unknown in the right hand side of equation 6.7 is the inverse Hessian matrix $\underline{H}^{-1}$. Hence if it can be determined and the quadratic approximation is valid, the local minimum $\underline{x}^*$ can be found directly from $\underline{x}_{k-1}$. One method of approximating $\underline{H}^{-1}$ is from equation 6.6 since

$$p_j = H^{-1} \cdot g_j \quad 1 \leq j \leq k \quad 6.8$$

At each new point, the value of p_j and g_j is calculated and after n such steps, the $n \times n$ matrices P_n and Q_n of p_j and g_j respectively are formed. Then

$$H^{-1} = P_n \cdot Q_n^{-1} \quad 6.9$$

If the initial quadratic assumption is valid, the Hessian matrix, and the inverse Hessian will be invariant over n steps and the minimum $\underline{x}^*$ will be found in exactly n steps using equation 6.7. Since the quadratic approximation is probably not exact, new steps are made from $\underline{x}_k$ and the inverse Hessian is updated at each step. The methods of updating the inverse Hessian are given in ref. (54); in general, estimates for corresponding p_j and g_j which are the oldest and contain the largest changes in gradients are replaced first.

6.3. Variables.

The independent variables considered in the optimisation study were fuel flow rate F_1 , steam flow rate F_2 , steam temperature T_2 , forward and reverse air flow rates F_3 and F_4 respectively, interstage heat removal rate $\dot{Q}_1$ and net second stage air flow rate $F_7 - F_8$. Although there is some experimental evidence indicating that solid formation may be reduced by increasing steam temperature (2), this effect is small and is apparent only when fuel atomisation is very poor, i.e. at low F_2/F_1 ratios. As steam temperature does not significantly alter the performance of the system, it has been held constant at 443 K for all optimisation studies.

To simplify the optimisation problem (see Section

6.4), the independent variables were expressed in terms of ϕ_1 , steam/fuel mass ratio F_2/F_1 , reverse/forward air ratio F_4/F_3 , fuel flow rate F_1 , fraction of maximum interstage heat removal $\dot{Q}_1/\dot{Q}_R$ and ϕ_2 .

6.4. Constraints.

In most practical applications, the objective function must satisfy side conditions in the form of equations or inequalities relating the independent variables. These constraints may be both linear and non-linear. In the case of strict equality constraints, it is usually possible to modify the objective function so as to satisfy these constraints before minimisation is attempted. This procedure reduces the degrees of freedom from n , the number of variables, to $n - m_1$ where m_1 is the number of strict equality constraints, both linear and non-linear.

In the system considered, any strict equality constraints such as overall stoichiometric operation ($\phi_2 = 1.0$), have been combined in the system model, and do not appear in the formulation of the optimisation problem.

Inequality constraints in real systems are usually of the form

$$a \leq c(\underline{x}) \leq b \quad 6.10$$

where a and b are constants and $c(\underline{x})$ is a linear or non-linear function of the independent variables.

Linear inequality constraints can be written in the form

$$\underline{g}_i^T \underline{x} - b_i \leq 0 \quad 1 \leq i \leq m_2 \quad 6.11$$

where m_2 is the number of linear inequality constraints.

Non-linear inequality constraints of the form of

equation 6.10 require the inclusion of slack variables $\underline{x}_s$ to form an equivalent problem

$$c(\underline{x}) - \underline{x}_s = 0 \quad 6.12$$

where

$$\underline{x}_s \leq 0 \quad 6.13$$

and the objective function is suitably redefined

$$f'(\underline{x}) = f(\underline{x}) + \sum_{i=1}^{m_3} (c(\underline{x}) - \underline{x}_s)^2 \quad 6.14$$

where m_3 is the number of non-linear inequality constraints. The slack variables are then constrained as linear inequalities as indicated by equation 6.11. The gradient projection technique used to maintain the search in the feasible region involves the use of a projection operator and is described in detail in reference (54).

The constraints imposed on this system arise from equipment limitations in the form of maximum and minimum fuel and air flow rates, as well as operational limitations including maximum and minimum steam/fuel mass ratios to maintain effective atomisation, and maximum interstage heat removal. The interstage heat removal is limited either by combustion stability in the second stage or by an arbitrarily chosen outlet gas stream temperature from the cooling tube based on the derived temperature of the steam generated.

The system constraints are enumerated in Table 6-1 and are classified as to type i.e. equipment or operating limitation. The constraints formed by equipment limitations are shown graphically in Fig.6-1 and define the region of possible system operation. Constraint 7 in Fig.6-1 is not truly an equipment limitation, but is included so as to give a closed operating region. The operating region is

TABLE 6-1.

SYSTEM CONSTRAINTS.

CONSTRAINT NUMBER	CONSTRAINT	TYPE OF LIMITATION	SYMBOL	VALUE
1	Maximum fuel flow rate	equipment	F_1	4.0 g/s
2	Minimum fuel flow rate	equipment	F_1	2.0 g/s
3	Maximum first stage air	equipment	$F_3 + F_4$	33.0 g/s
4	Minimum first stage air	equipment	$F_3 + F_4$	18.0 g/s
5	Maximum net second stage air	equipment	$F_7 - F_8$	29.0 g/s
6	Minimum net second stage air	equipment	$F_7 - F_8$	7.0 g/s
7	Maximum first stage equivalence ratio	operating	ϕ_1	2.3
8	Maximum interstage heat removal	operating	$\dot{Q}_R$	equation 5.5
9	Minimum interstage heat removal	operating	-	0
10	Maximum steam/fuel mass ratio	operating	F_2/F_1	1.3
11	Minimum steam/fuel mass ratio	operating	F_2/F_1	0.5
12	Maximum reverse/forward air ratio	operating	F_4/F_3	1.3
13	Minimum reverse/forward air ratio	operating	F_4/F_3	0.7
14	Minimum second stage equivalence ratio	process	ϕ_2	1.0
15	Maximum second stage solid emission	process	m_{T2}	0.0040 $\frac{\text{kg solid}}{\text{kg fuel}}$
16	Maximum interstage heat removal	process	$\dot{Q}_{\max}$	gas outlet temperature = 600 K
17	Constant steam temperature	constant	T_2	443 K
18	Constant second stage outlet air	constant	F_8	5.0 g/s

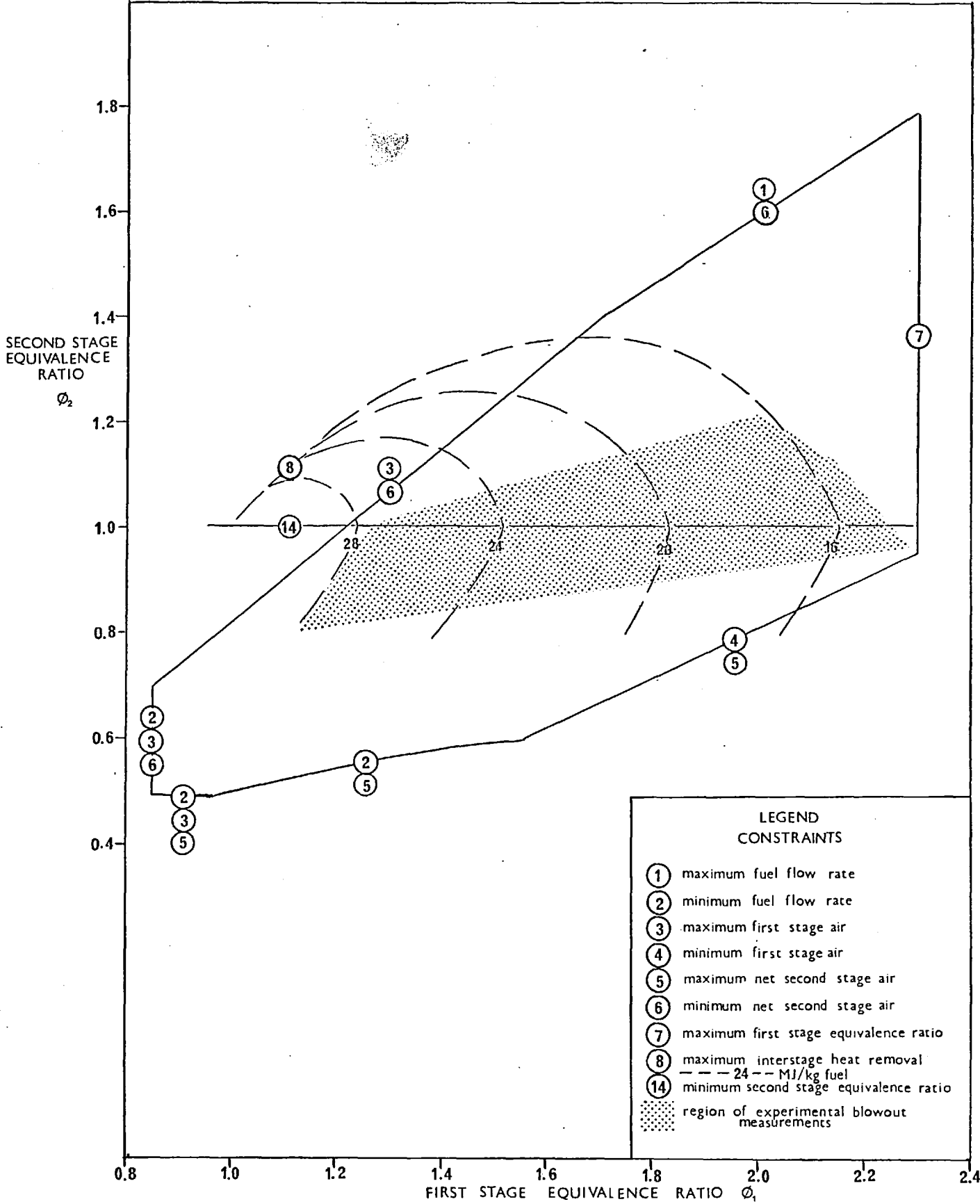


FIG.6-1 SYSTEM OPERATING REGION

considerably reduced by the requirement of excluding excess air i.e. $\phi_2 \geq 1.0$, which limits operation to the upper portion of the feasible region. It can be seen that this requirement limits ϕ_1 to values greater than 1.22 as the minimum net second stage air flow rate is 7.0 g/s.

The shaded area in Fig.6-1 indicates the region for which experimental blowout measurements have been obtained. The blowout limits, shown by a series of broken lines, indicate maximum interstage heat removal. The values of interstage heat removal shown in Fig.6-1 (MJ/kg fuel) are correct for $\phi_2 = 1.0$ only, and are obtained from Fig.5-8.

The parabolic shape of the curves is obtained from theory, but the values of heat removal for $\phi_2 \neq 1$ are not necessarily true. As second stage air loading increases with decreasing ϕ_2 , it is known that the heat removal at blowout decreases. Similarly, as ϕ_2 increases, the maximum interstage heat removal must approach zero as ϕ_2 approaches ϕ_1 , hence the contours approach the $\phi_2 = \phi_1$ line asymptotically as ϕ_1 decreases. The range of experimental results was not wide enough to confirm this behaviour. Fortunately, the only region of interest in the present study is $\phi_2 = 1$, for which the heat removal contours have been verified.

For generation of high pressure steam (690 kN/m², 450 K), the minimum working combustion gas temperature at the outlet from the heat exchange section is taken to

be 600 K. This lower temperature limit imposes a second constraint on the maximum interstage heat removal (the first one is imposed by thermal stability). These two heat removal constraints are compared in Fig.6-2 as a function of ϕ_1 . The solid line corresponds with the blowout limit (equation 5.6) whilst the broken line represents the 600 K limit for $F_2/F_1 = 0.9$. Changes in F_2/F_1 do not change the curves significantly. It can be seen that for $\phi_1 > 1.2$, the 600 K limit is the active constraint on the system. The maximum heat removal consistent with the 600 K restriction is denoted as $\dot{Q}_{\max}$. If the maximum temperature were 500 K, the constraint would be active only for $\phi_1 > 1.4$. This constraint is also shown in Fig.6.2.

All of the system constraints could not be expressed as linear inequalities. Depending on the choice of independent variables, the constraints taking non-linear form could be chosen accordingly; e.g. if fuel flow rate F_1 and steam flow rate F_2 are independent variables, the constraint corresponding to steam/fuel mass ratio is non-linear in terms of F_1 and F_2 .

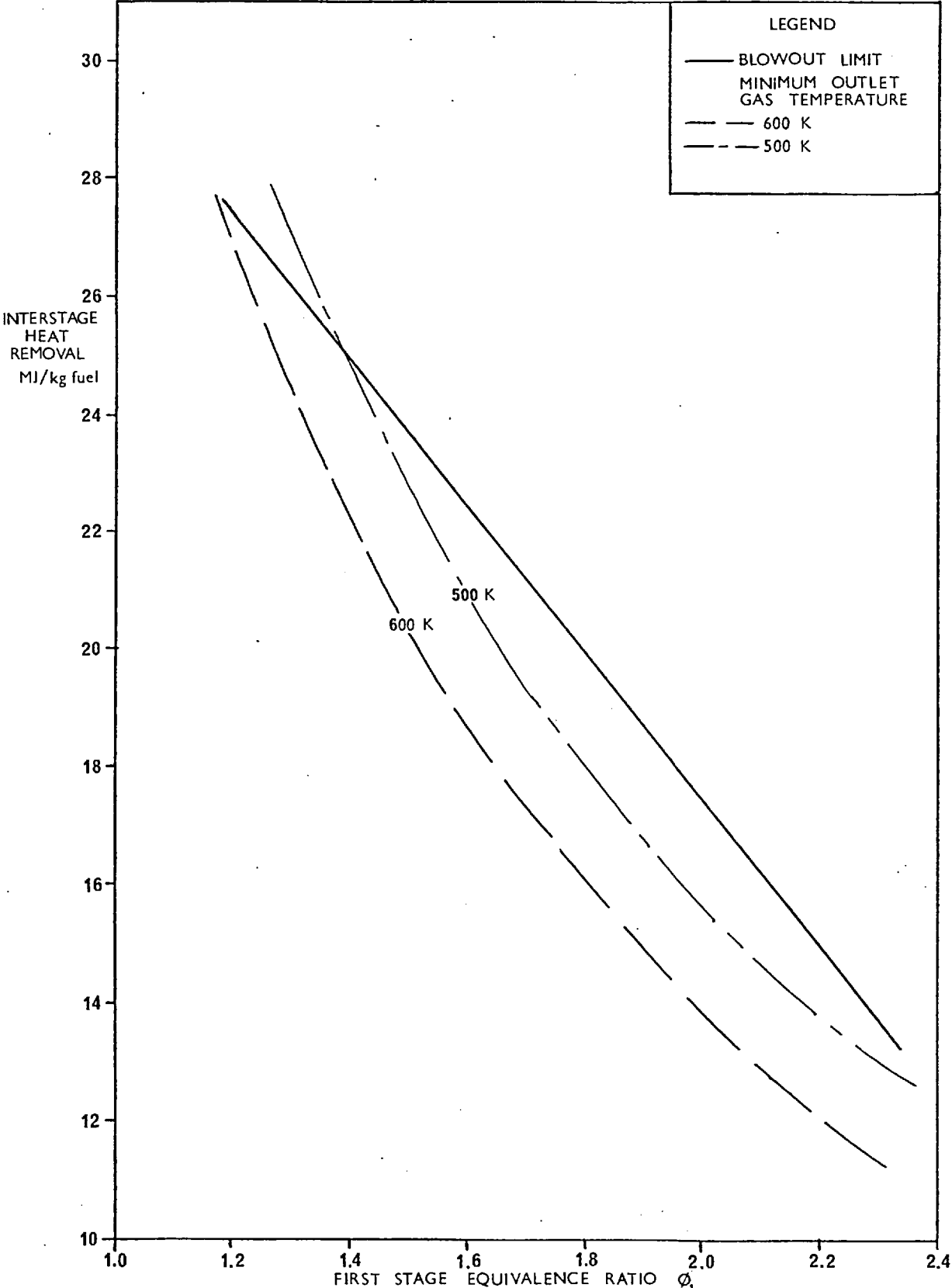


FIG.6-2 MAXIMUM INTERSTAGE HEAT REMOVAL CONSTRAINTS

The constraints on first and second stage air flow rates were considered in non-linear form using the combined forward and reverse air flow rate in the first stage, and net second stage air as slack variables.

Both independent and slack variables were normalised between 0 and 1 to facilitate matrix operations.

6.5. Modifications.

Indirect minimisation techniques are known to be very powerful when the gradient of the objective function can be calculated analytically and accurately. In the present case where gradients cannot be determined analytically, numerical methods must be employed to evaluate the derivatives of the objective function. The numerical method of evaluating function gradients becomes inaccurate in the vicinity of the minimum where gradients approach zero, and may be affected by rounding errors.

A second, more serious limitation is that the second derivatives of the objective function are not truly continuous. The discontinuities arise from the method of solution of the heat and mass balances in the system model, which require a trial and error type solution. Each heat balance is determined to an accuracy of $\pm 1\%$ as more accurate solutions require disproportionately more computational time. Small changes in system variables may occur without changing the calculated heat and mass balances within the above accuracy, hence system performance will not change. The objective function is not smooth, but is a series of steps, wide or narrow depending upon the gradient at that point. Where the minimum is relatively flat, the steps will be wide,

hence the true minimum may not be found as the search may stop on a wide step where the gradients appear to be zero.

This problem has been partially overcome in this instance by making a sensitivity analysis of the objective function and scaling the step size used to evaluate gradients accordingly. The requirement of this preanalysis weakens the optimisation method.

As with all minimisation techniques, the determination of a minimum of an objective function does not mean that the global minimum has been found. In this case where the objective function is not known to be unimodal, either a designed scan of the region, or successive minimisations from different starting points have been made.

The accuracy to which the required minimum is found is limited by the accuracy of the model. In the present study, a careful investigation has been made of the response surface corresponding to solid emission from the first stage (see Section 7-3). Using this as a test surface, the convergence of the minimisation program using numerical derivatives was verified.

CHAPTER 7.

INTERPRETATION OF RESULTS.

Results predicted using the mathematical models developed in Chapter 4 are shown to correspond with experimental results within experimental error (see Chapter 5 and Appendix 5). In this chapter, the mathematical models are used to determine the effects of individual variables on system performance, and to construct response surfaces for various system criteria. These response surfaces are essential if on-line control is required.

7.1. Designed Model Experiments.

It was proposed that factors F_1 , F_2 , F_3 , F_4 , F_7 , F_8 , T_2 and $\dot{Q}_1$ (interstage heat removal rate) should be considered in the determination of system response surfaces. After preliminary analysis, it was found that steam temperature T_2 had no effect on system performance in the range 413 K - 473 K, and that the effects of second stage outlet air F_8 could be neglected. The remaining six operating factors were expressed in terms of fuel flow rate F_1 , steam/fuel mass ratio F_2/F_1 , reverse/forward air ratio F_4/F_3 , first stage equivalence ratio $\phi_1, (14 F_1/(F_3+F_4))$, second stage equivalence ratio $\phi_2, (14 F_1/(F_3+F_4+F_7-F_8))$, and fraction of the maximum interstage heat removal $\dot{Q}_1/\dot{Q}_{\max}$ (for $\dot{Q}_{\max}$ see section 6.4).

Initially a second order factorial design was constructed, however this indicated that the relevant response surfaces could not be determined satisfactorily, and the design was expanded to include intermediate points.

The level of the factors studied was determined from known operating constraints. The designed tests are given in Table 7-1. The values are equally spaced and have been normalised between -2 and +2 corresponding to the range of settings indicated in Table 7-2. It is seen from Table 7-1 that a $5 \times 5 \times 3 \times 3 \times 3 \times 5$ factorial experiment was designed but only 175 preferred combinations of different conditions were considered. The maximum interstage heat removal $\dot{Q}_{\max}$ is limited in all cases by the minimum combustion gas outlet temperature of 600 K. Initial experiments showed that where economic optimisation was required, maximum interstage heat removal was essential, hence only limited tests were made using interstage heat removal rates other than the maximum.

A supplementary design based on this set for constant levels of $F_2/F_1 = 0.7$ at $\dot{Q}_1 = 20$ kW was also made.

A limited set of tests were made to determine the effects of both first and second stage combustion chamber volume on system performance.

7.2. Gas Composition and Temperature.

Combustion gas composition is shown as a function of equivalence ratio in Fig. 7-1 for a fuel flow rate of 3.0 g/s and a steam/fuel mass ratio of 0.9. The gas is in water gas shift equilibrium, and solids are indicated as a weight percent of the fuel. Nitrogen can be determined by difference. It can be seen that the mole fraction of hydrogen is lower than that of carbon monoxide over the whole range. Previously (1), methane has been ignored, hence

TABLE 7-1. SUMMARY OF CONDITIONS FOR DESIGNED MODEL TESTS.

MODEL TEST NUMBER	TEST CONDITIONS.					
	F_1^*	ϕ_1	ϕ_2	F_2/F_1	F_4/F_3	$\dot{Q}_1/\dot{Q}_{max}$
001	B	0	0	0	0	2
002	A	0	0	0	0	1
003	A	0	0	0	0	0
004	A	0	0	0	0	-1
005	A	0	0	0	0	-2
006	B	-1	0	0	0	2
007	A	-1	0	0	0	0
008	A	-1	0	0	0	-2
009	B	1	0	0	0	2
010	A	1	0	0	0	0
011	A	1	0	0	0	-2
012	B	0	-2	0	0	2
013	B	0	2	0	0	2
014	B	0	0	0	-2	2
015	A	0	-2	0	-2	2
016	A	0	2	0	-2	2
017	B	0	0	0	2	2
018	A	0	-2	0	2	2
019	A	0	2	0	2	2
020	B	0	0	-2	0	2
021	A	0	-2	-2	0	2
022	A	0	2	-2	0	2
023	B	0	0	2	0	2
024	A	0	-2	2	0	2
025	A	0	2	2	0	2
026	B	0	0	-2	-2	2
027	B	0	0	-2	2	2
028	B	0	0	2	-2	2
029	B	0	0	2	2	2
030	B	-1	0	-2	0	2
031	B	-1	0	2	0	2
032	E	1	0	-2	0	2
033	E	1	0	2	0	2
034	B	-2	0	0	0	2
035	B	-2	0	-2	0	2
036	B	-2	0	2	0	2
037	A	-2	-2	0	0	2
038	A	-2	2	0	0	2
039	B	2	0	0	0	2
040	B	2	0	-2	0	2
041	B	2	0	2	0	2
042	A	2	-2	0	0	2
043	A	2	2	0	0	2

* A = 1, 0, -1

B = 2, 1, 0, -1, -2.

TABLE 7-2. RANGE OF SETTINGS FOR MODEL TESTS.

FACTOR	MAXIMUM	MINIMUM
Fuel flow rate g/s F_1	4.0 g/s (2)	2.0 g/s (-2)
First stage equivalence ratio ϕ_1	2.3 (2)	1.1 (-2)
Second stage equivalence ratio ϕ_2	1.05 (2)	0.95 (-2)
Steam/fuel mass ratio F_2/F_1	1.3 (2)	0.5 (-2)
Reverse/forward air ratio F_4/F_3	1.3 (2)	0.7 (-2)
Interstage heat removal Q_i	$Q_{\max}$ (2)	0 (-2)

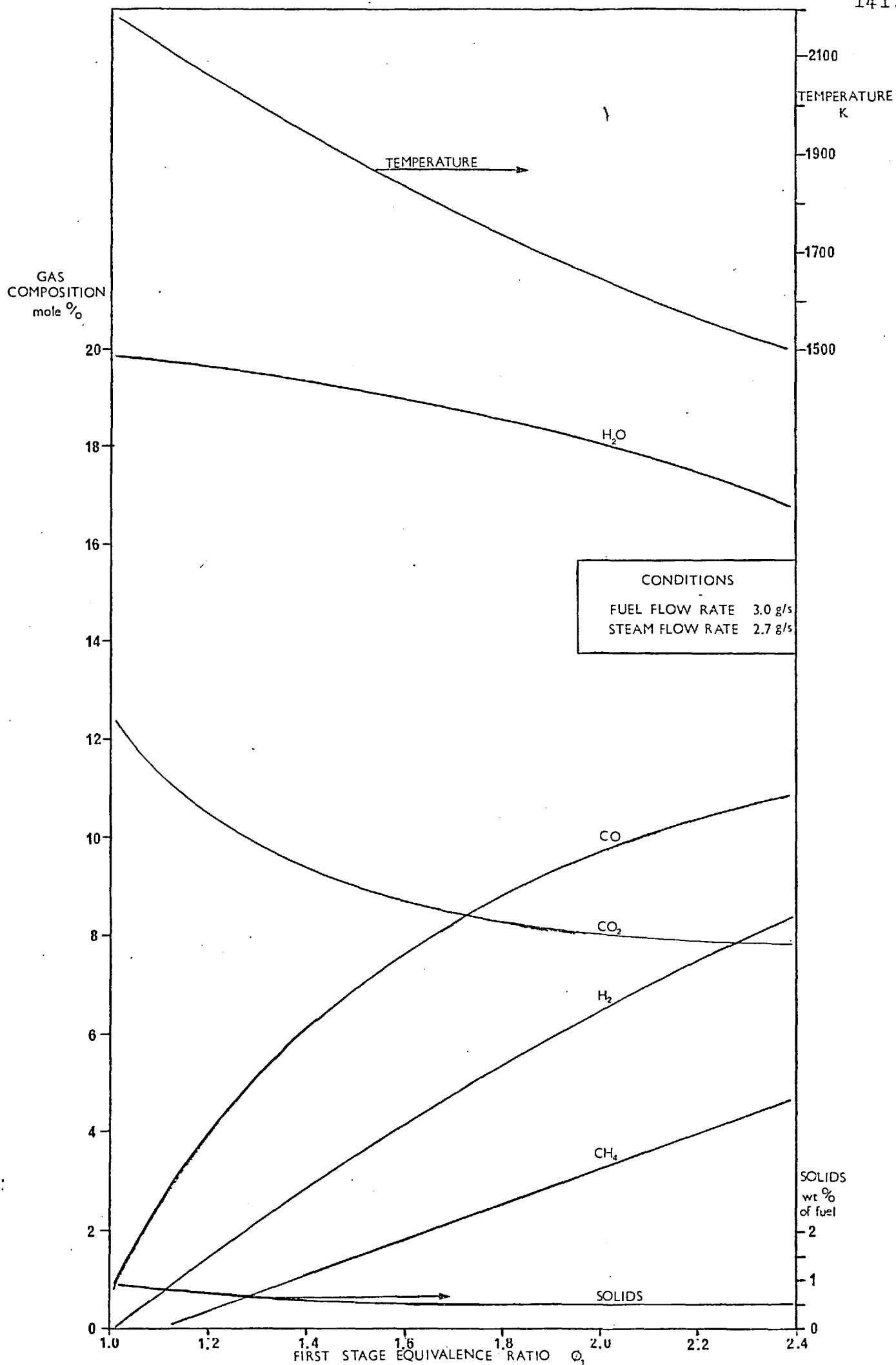


FIG.7-1 CALCULATED GAS COMPOSITION AND TEMPERATURE FOR STEAM ATOMISATION

increasing the hydrogen mole fraction above that of carbon monoxide. Corresponding gas compositions are shown in Fig.7-2 for air atomisation at an atomising air/fuel mass ratio of 0.9. It can be seen that the mole fractions of both hydrogen and carbon monoxide are substantially higher than the corresponding figures for steam atomisation. The solid burden in the gas stream is increased as well. The increase in hydrogen is due to the considerably lower amount of methane present, whereas the water vapour is reduced by the absence of steam atomisation. To satisfy the water gas shift equilibrium subject to these conditions, carbon monoxide is increased and carbon dioxide decreased. There is an interesting comparison between the gas compositions predicted in Fig.7-2 and those measured by HTUN (2). It is known that his reported dry gas compositions (2) indicated more carbon (as CO and CO₂) than was theoretically possible, however in all cases, the measured amounts of carbon monoxide and carbon dioxide combined were about 5 volume percent lower than the combined CO - CO₂ - H₂O predicted from theory. This error was not affected by equivalence ratio. It was further noted that the measured and predicted concentrations of carbon monoxide were approximately equal. These observations are interpreted to mean that the gas samples for analysis by gas chromatography were not properly dried and that the carbon dioxide and water vapour present were interpreted as carbon dioxide only. This interpretation is supported by the fact that carbon dioxide and water vapour are not likely to be separated by the type of gas chromatograph used in this analysis (2). If it is assumed

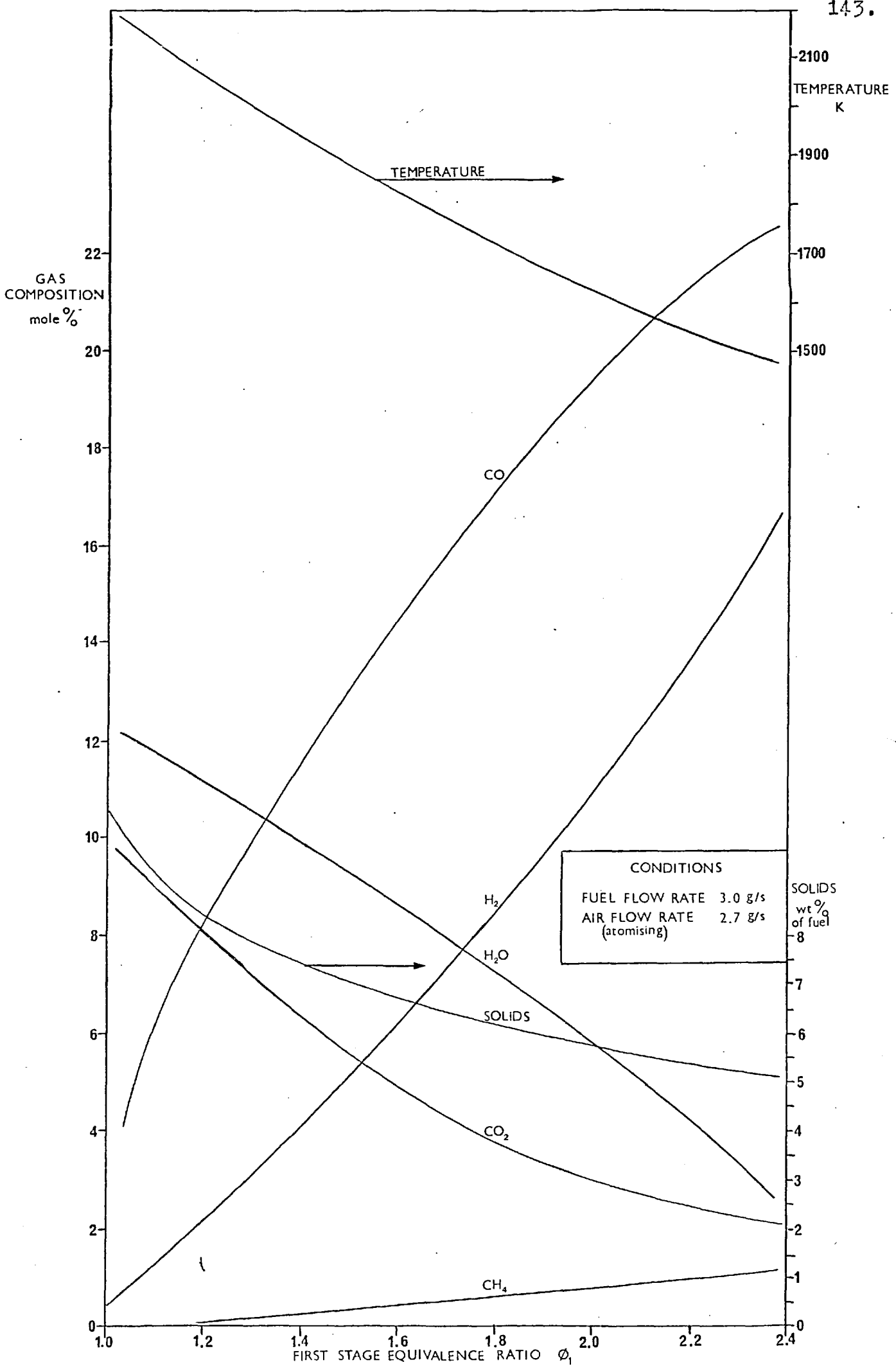
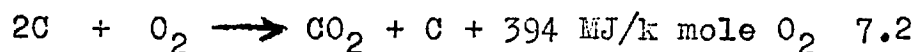
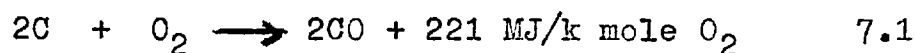


FIG.7-2 CALCULATED GAS COMPOSITION AND TEMPERATURE FOR AIR ATOMISATION

that drying the gas sample removed only 50% of the water vapour, and that the remaining water vapour was reported as CO_2 , then the measured results correspond with theory.

The combustion gas temperatures corresponding to the above gas compositions are shown in Fig.7-1 and Fig.7-2. It can be seen that the gas temperatures in both cases are approximately equal. The use of steam atomisation can be expected to lower the combustion gas temperature by as much as 200 K if all other conditions remain unchanged. It has been suggested (1,34) that the formation of carbon will tend to increase gas temperature as can be seen from heat release considerations viz.



From Fig. 7-1 and Fig. 7-2 it can be seen that there is substantially more carbon present under air atomised conditions, hence a greater temperature difference than that caused by steam atomisation would be expected. The anomaly can be resolved by considering Fig. 7-3 where the relative amounts of char and soot formed are indicated. It can be seen that soot formation is the same in each case whilst char formation varies appreciably. In all cases, the amount of char present under air atomisation is greater than that for steam. This is as expected since atomisation is improved by the use of steam which reduces the initial D_{sm} of the spray. Steam atomisation increases fuel evaporation efficiency, hence the number of moles of gasified reactants is increased. Since the system remains in water gas shift equilibrium in both cases, it can be seen from Fig. 7-1 and Fig. 7-2 that

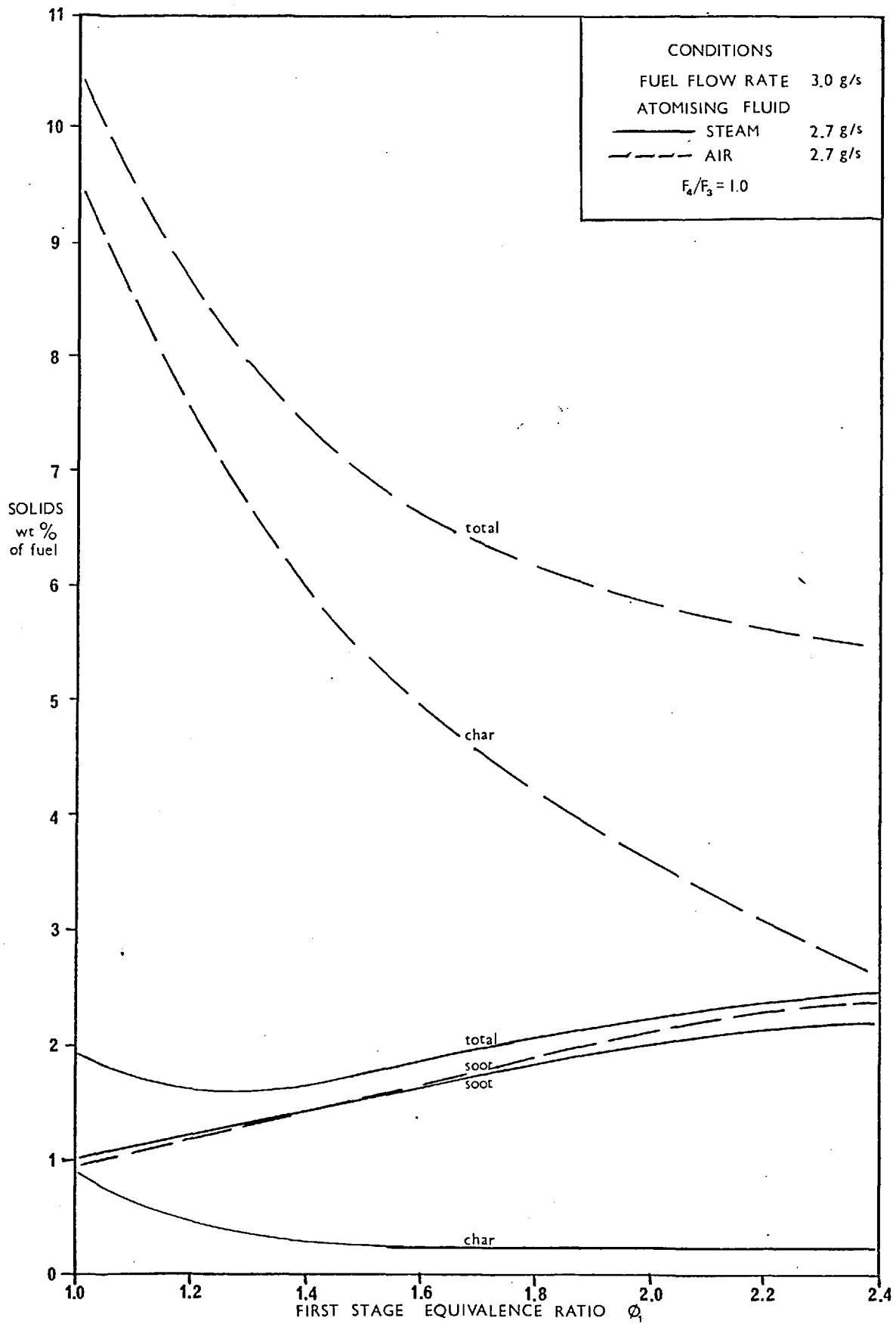


FIG.7-3 COMPARISON OF FIRST STAGE SOLIDS FOR AIR AND STEAM ATOMISATION

equation 7.1 predominates in the case of air atomisation whilst equation 7.2 predominates under steam atomised conditions. The result is that heat release is increased by the use of steam, and gas temperatures for both steam and air atomisation are approximately the same.

7.3. Solids Formation.

Solid formation takes place by two essentially independent mechanisms. Char is formed from incompletely evaporated fuel drops. The amount and size distribution of the char formed is influenced by the characteristics of the initial fuel spray and by the mean residence time in the first stage chamber.

The characteristic diameter D_{sm} of the fuel spray for both air and steam atomisation is indicated in Fig. 7-4 based on equations 2.5 and 2.6. Equation 2.5 has been used throughout this work. It is seen that equation 2.6 does not predict a reduction in spray D_{sm} by the use of steam rather than air. This is attributable to the effect of the gas density term ρ_g in equation 2.6 which was determined for small changes in the density of one gas only. Steam/fuel mass ratio determines spray characteristics, hence has a large influence on the amount of char formed.

Molar flow rate and gas temperature, which depend upon fuel and steam flow rates as well as equivalence ratio, determine the mean residence time in the chamber. Char formation is decreased both by increasing steam/fuel mass ratio and by increasing mean residence time. Mean residence time may be increased either by decreasing fuel flow at constant ϕ_1 , or increasing ϕ_1 at constant fuel flow. Thus,

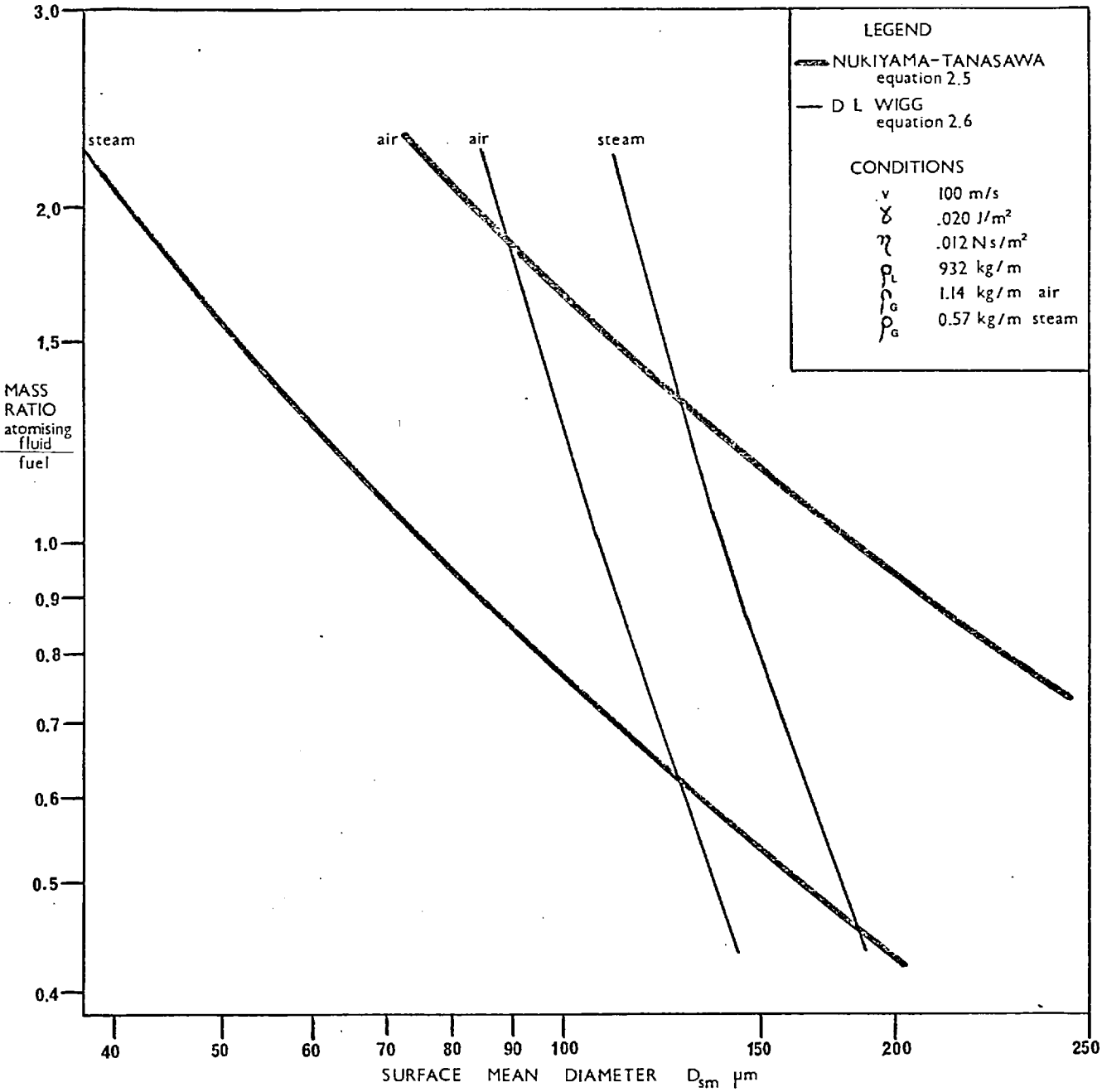


FIG. 7-4 SURFACE MEAN DIAMETER OF SPRAYS FROM TWIN FLUID ATOMISERS

in order to minimise char formation, low fuel flow rate, high steam/fuel mass ratio, and high equivalence ratio are required.

Soot formation is predicted by equation 5.1. It is seen from Fig. 7-3 that soot increases as ϕ_1 increases at constant fuel flow rate. This suggests that soot formation depends on relative fuel and oxygen concentrations as opposed to dependence on combustion temperature since it was expected that soot would decrease with decreasing gas temperature as a result of decreased hydrocarbon cracking. It is also found that soot formation increases as fuel flow rate increases (see equation 5.1).

The total amount of soot and char, i.e. total solids, formed is indicated for four steam/fuel mass ratios in Figs. 7-5, 7-6, 7-7 and 7-8. The contours indicate a constant amount of solid. The broken lines are system constraints and enclose the feasible operating region. The constraint numbers correspond with those given in Table 6-1.

Within the feasible region, it can be seen that solid formation increases with fuel flow rate under all conditions. The decreases observed in Fig. 7-7 and Fig. 7-8 above a fuel flow rate of about 3.5 g/s may arise from the regression equation for soot formation which should not be extrapolated outside the feasible region. As atomisation is improved by the use of higher steam/fuel mass ratios, the total solid burden is reduced and the presence of soot becomes more evident. At $F_2/F_1 = 0.5$, an increase in ϕ_1 decreases total solids for any fuel flow rate. This indicates that the reduction in char formation is greater than the increase in soot formation. As F_2/F_1 increases, the rate

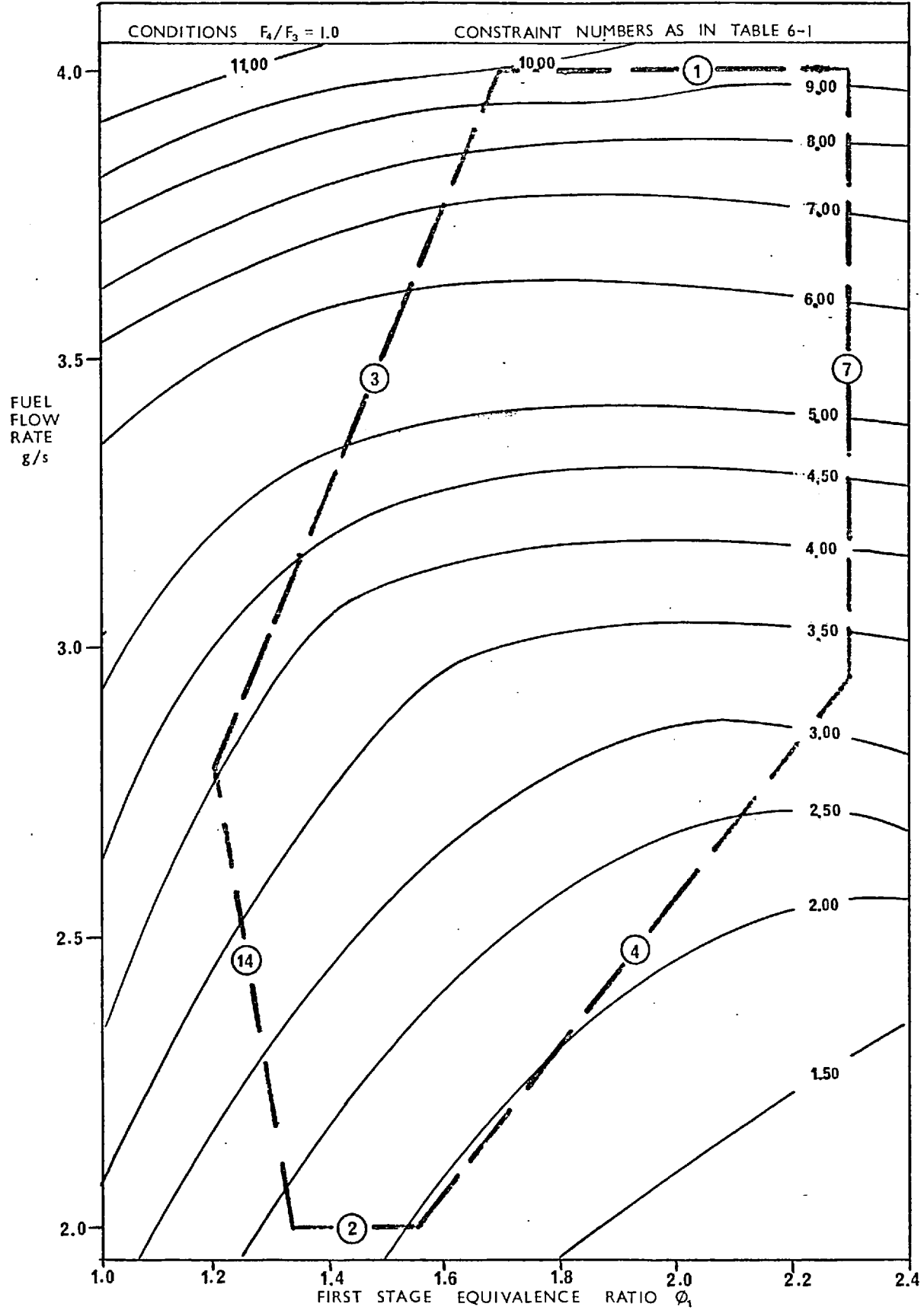


FIG.7-5 FIRST STAGE SOLIDS AS % OF FUEL AT $F_2/F_1 = 0.5$

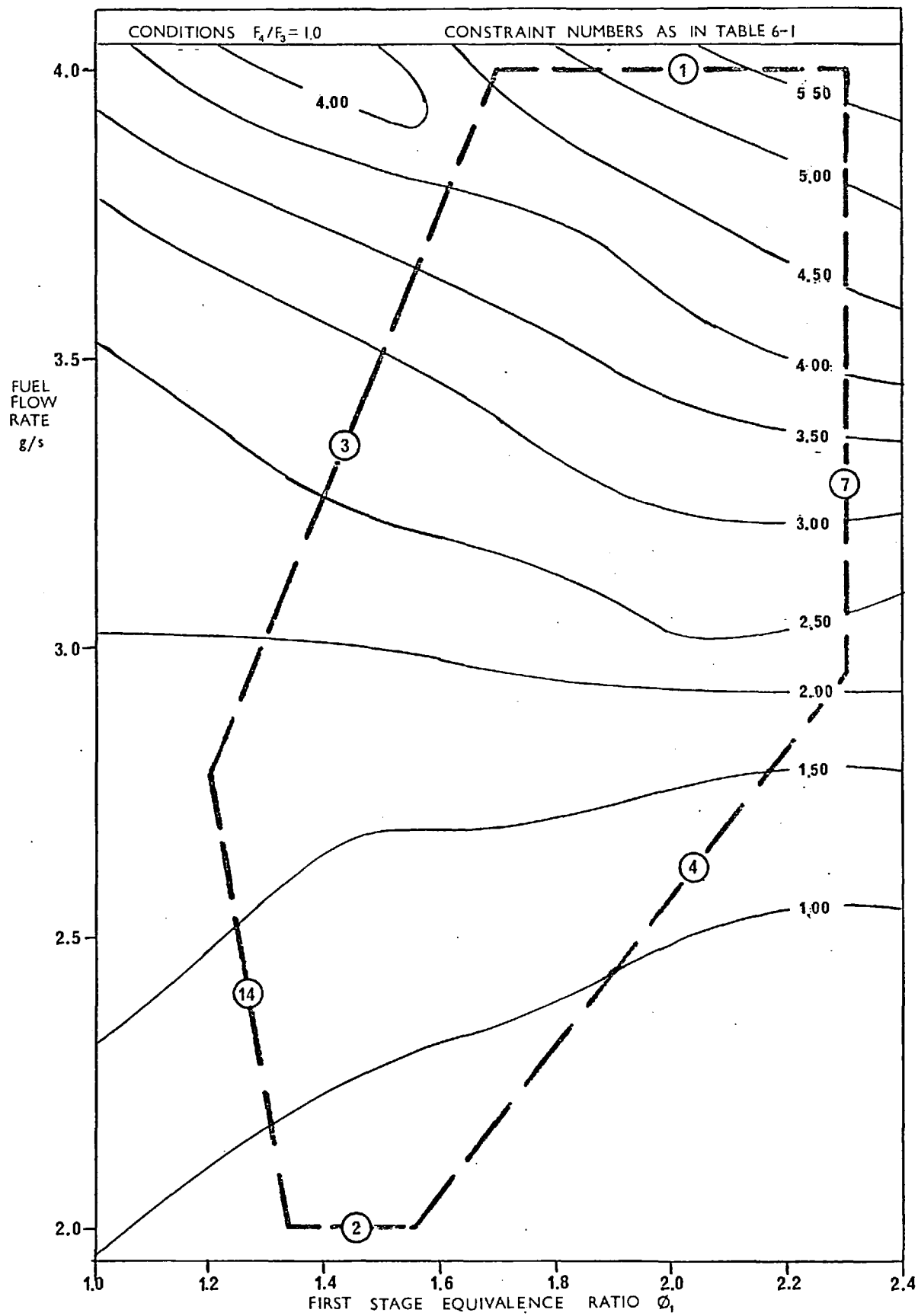


FIG.7-6 FIRST STAGE SOLIDS AS % OF FUEL AT $F_2/F_1 = 0.7$

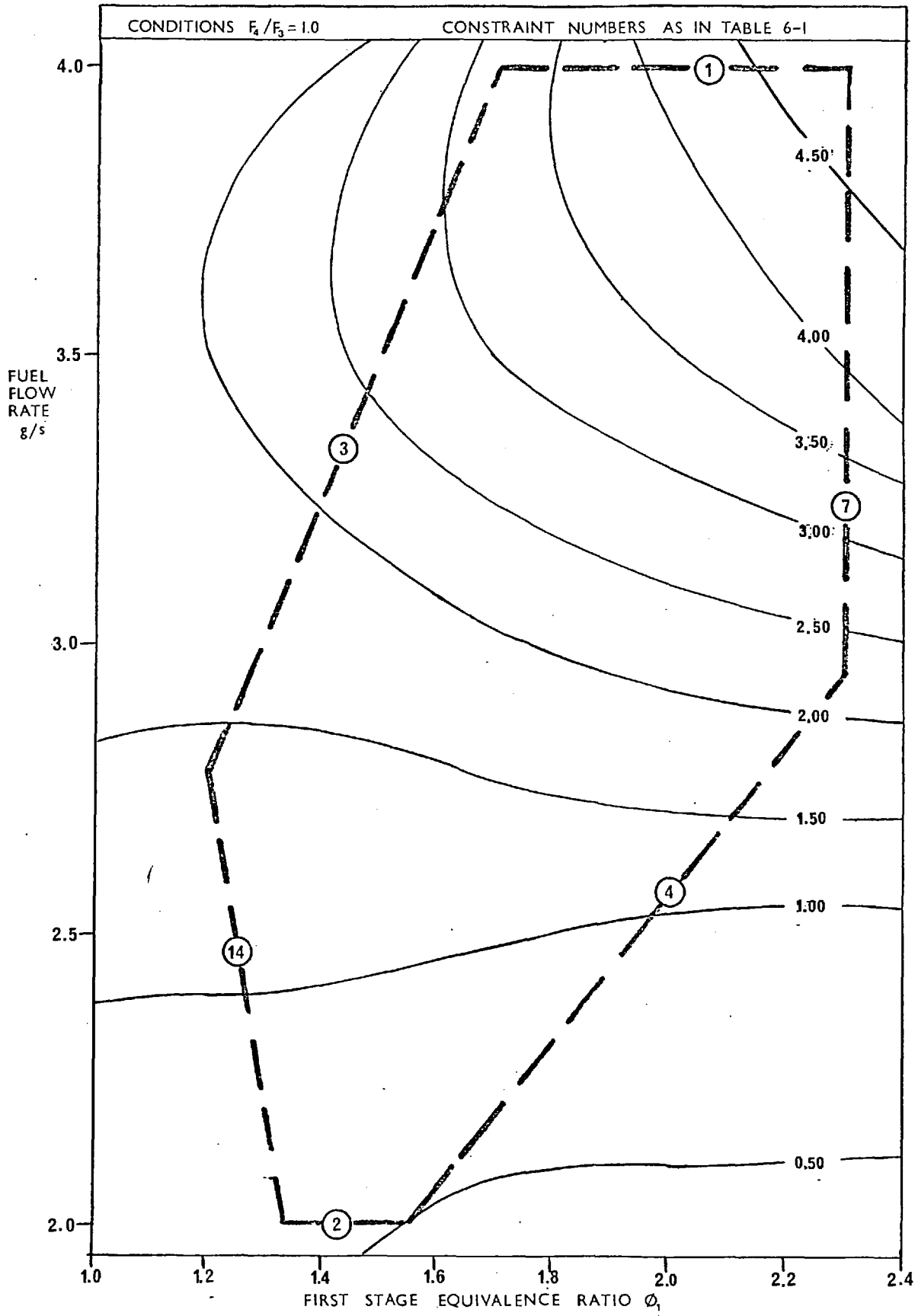


FIG. 7-7 FIRST STAGE SOLIDS AS % OF FUEL AT $F_2/F_1 = 0.9$

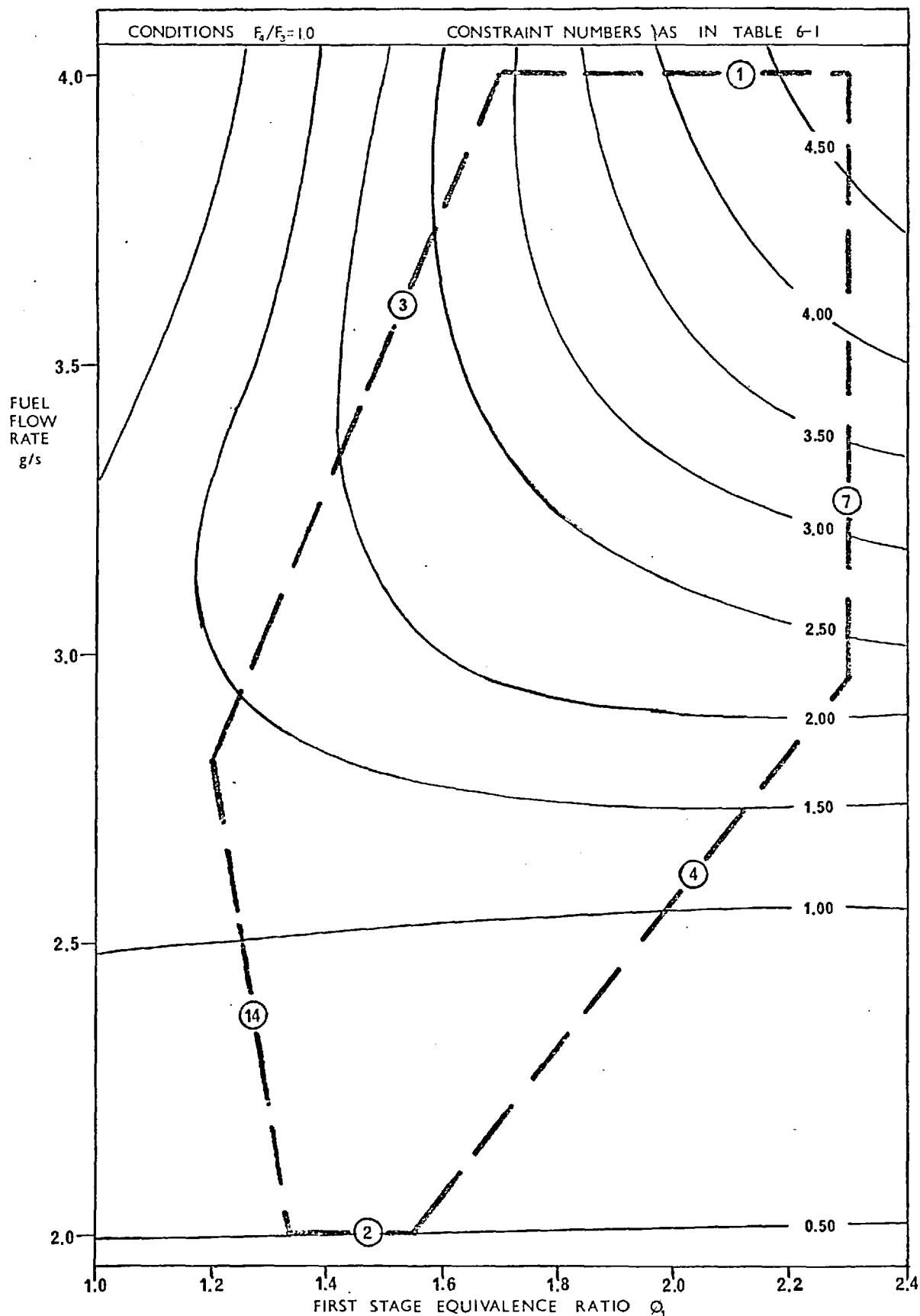


FIG.7-8 FIRST STAGE SOLIDS AS % OF FUEL AT $F_2/F_1=1.3$

of char reduction with ϕ_1 becomes smaller until at $F_2/F_1 = 0.7 - 0.9$ no significant improvement is made and it is seen that solids now increase with increasing ϕ_1 which is characteristic of soot formation. The effect of F_2/F_1 on total solid burden is shown in Fig. 7-9 for a fuel flow rate of 3.00 g/s at three levels of ϕ_1 . For $F_2/F_1 < 0.6$, solid burden decreases with increasing ϕ_1 . This is because at constant fuel flow rate, first stage mean residence time increases with increasing ϕ_1 resulting in higher fuel evaporation efficiencies and less char. At higher F_2/F_1 , fuel evaporation efficiency is less dependent on equivalence ratio, but soot formation increases with increasing ϕ_1 ; hence an overall increase in solid loading is observed.

The dependence of particle mass distribution on steam/fuel mass ratio is shown in Fig. 7-10.

Very little change in the shape of the soot particle distribution occurs in each case. The char particle distribution changes considerably; it will be shown however, in sections 7.4 and 7.5, that size distribution changes of this type have little effect on system performance.

7.4. Interstage Heat Transfer.

Interstage heat transfer rates have been predicted using equations 4.76 and 4.77, with a correction term calculated from equation 5.2. The total heat transfer rate $\dot{Q}_T$ can be expressed in terms of a single heat transfer coefficient h_T viz.

$$\dot{Q}_T = h_T A_x \Delta T_{lm} \quad 7.3$$

where

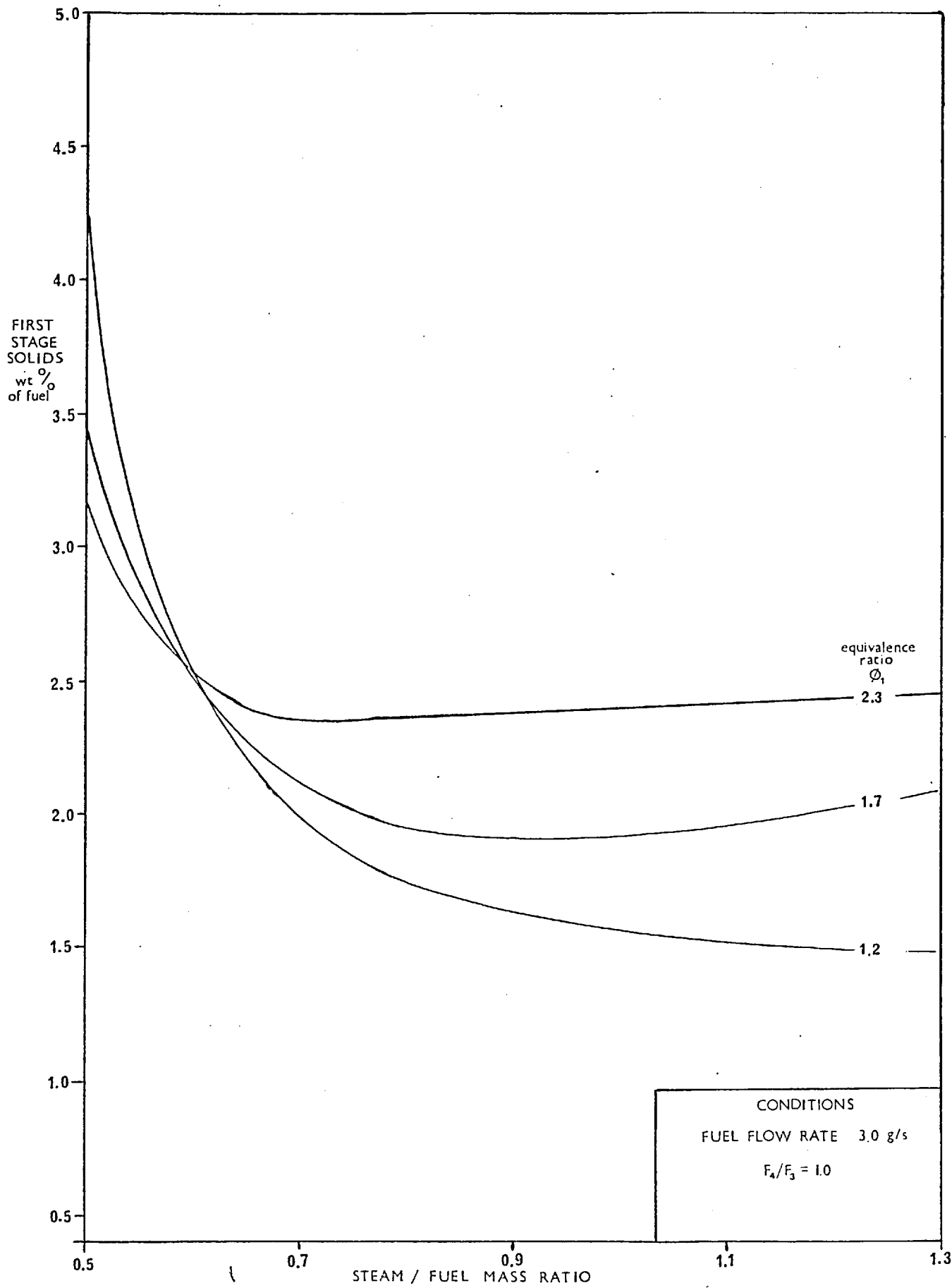


FIG.7-9 THE EFFECT OF STEAM / FUEL MASS RATIO ON FIRST STAGE SOLIDS

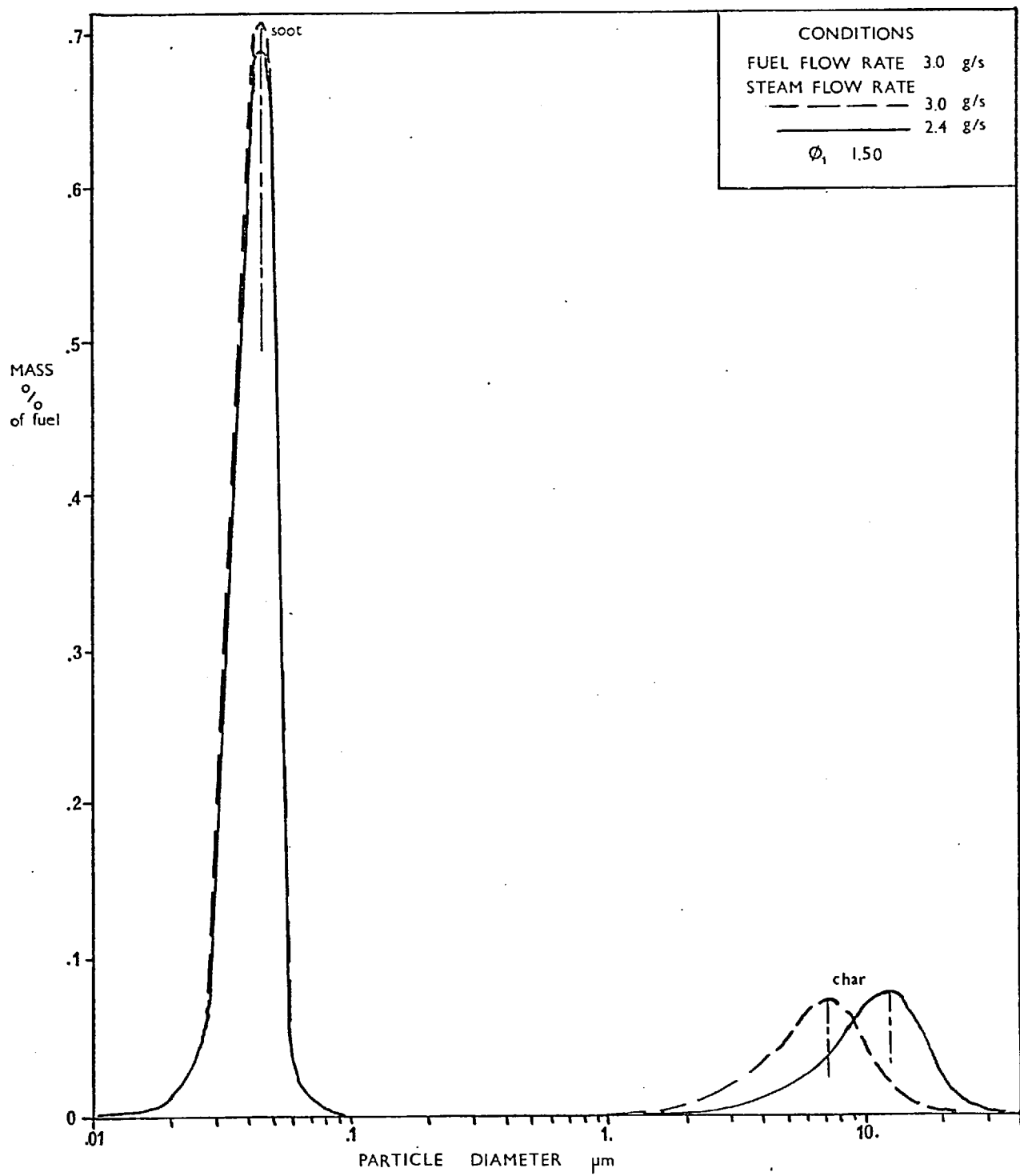


FIG.7-10 THE EFFECT OF STEAM / FUEL MASS RATIO ON PARTICLE MASS DISTRIBUTION

$$\Delta T_{lm} = \frac{T_{IN} - T_{OUT}}{\ln \left(\frac{T_{IN} - T_w}{T_{OUT} - T_w} \right)} \quad 7.4$$

The overall heat transfer coefficient can be determined by combining equations 4.76, 4.77, and 7.3 to eliminate A_x .

$$h_T = \frac{\sigma \epsilon_T (T_{rm}^4 - T_w^4) + h_c (T_{cm} - T_w)}{\Delta T_{lm}} \quad 7.5$$

Values of h_c are calculated from the Colburn equation for convective heat transfer, whilst ϵ_T is determined using equation 4.78.

Thus, for any $\dot{Q}_T$, a value of h_T can be calculated and the corresponding heat transfer area A_x found from equation 7.3. Interstage heat transfer surface areas calculated in this way are shown in Fig. 7-11 and Fig. 7-12 as functions of equivalence ratio and fuel flow rate. The results shown in Fig. 7-11 are for maximum interstage heat removal $\dot{Q}_{max}$ at a steam/fuel mass ratio of 0.9. The steam/fuel mass ratio is 1.3 in Fig. 7-12. The permitted operating region is shaded in each figure and is enclosed by system constraints. The constraint numbers correspond with those shown in Table 6-1.

7.5. Solids Burning.

The burnout of soot and char has been calculated using the model developed in Section 4.6 with the modification of Section 5.4. The progress of both the gas and solid reactions in the second stage chamber is shown in Fig. 5-6 as a function of time. It can be seen that while the oxygen concentration is high considerable quantities of solid are burned. The solid burned is soot. The char

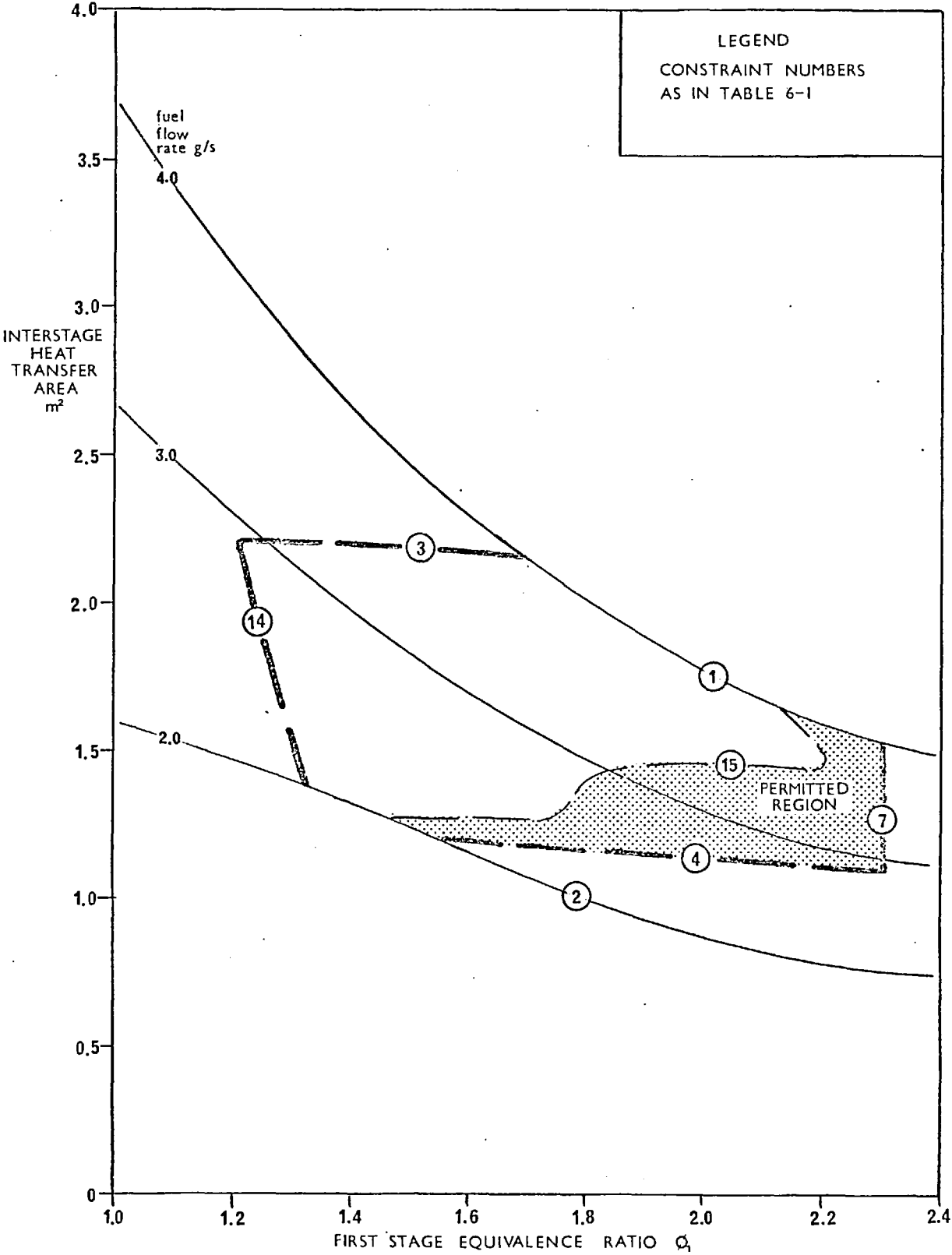


FIG.7-11 HEAT TRANFER SURFACE FOR MAXIMUM INTERSTAGE HEAT REMOVAL AT $F_2/F_1=0.9$

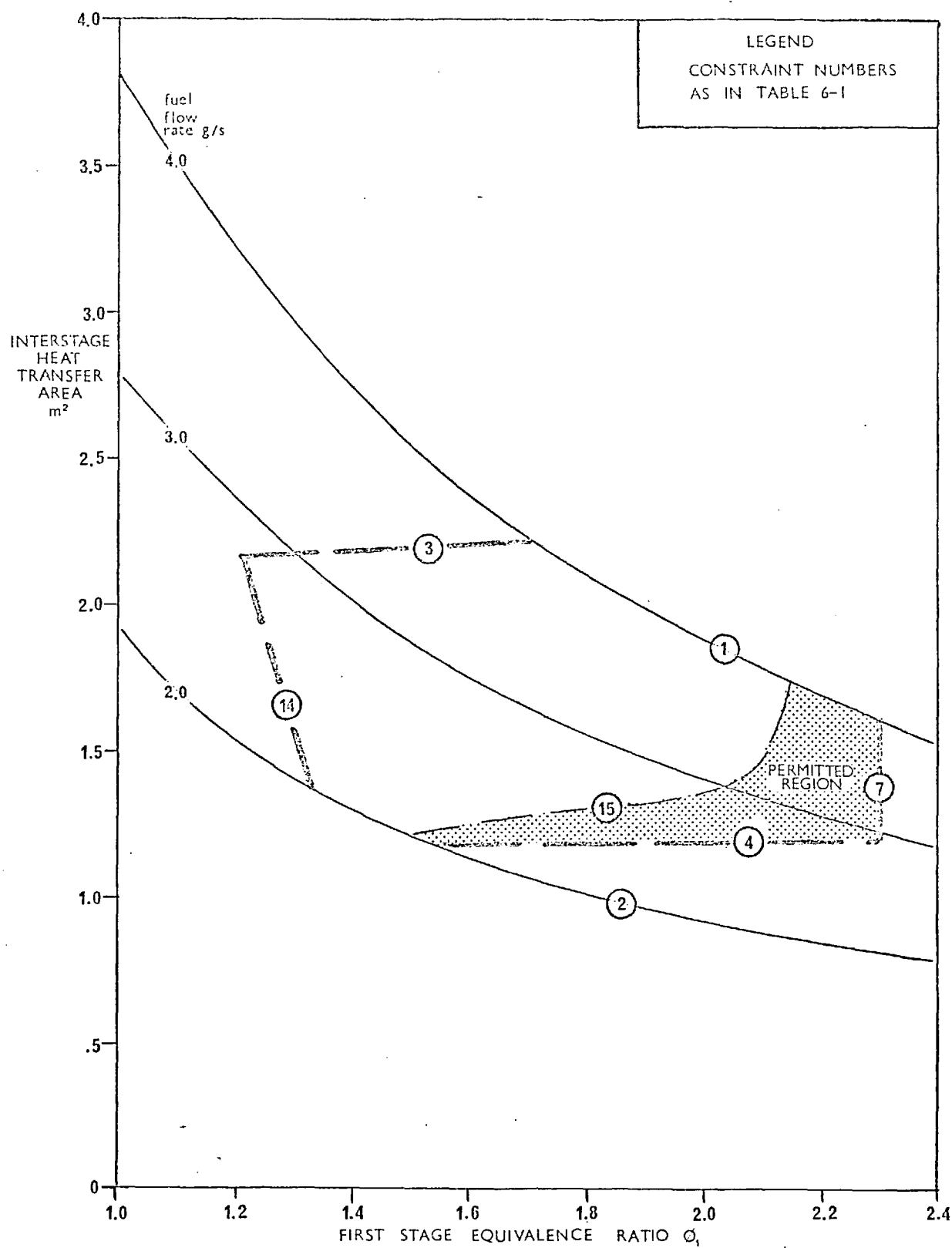


FIG.7-12 HEAT TRANSFER SURFACE FOR MAXIMUM INTERSTAGE HEAT REMOVAL AT $F_2/F_1 = 1.3$

present reacts very slowly and is not appreciably burned away in the chamber. The gas-solid reactions involving CO_2 and H_2O proceed very slowly in comparison with the oxygen-carbon reaction. The particle mass distributions before and after combustion at the conditions shown in Fig.5-6 are shown in Fig.7-13. It is evident that the soot rather than the char is burned. Further, even where atomisation in the first stage is good, yielding char particles of small diameter ($<10 \mu\text{m}$), the char particles are not appreciably burned even at stoichiometric conditions in the second stage.

The amount of solid that can be burned in the second stage depends on the type of solids formed in the first stage, as well as the temperature and oxygen concentration in the second stage. Second stage temperature is determined by interstage heat removal and second stage equivalence ratio. As overall stoichiometric operation is required, the results presented are for $\phi_2 = 1.0$ only. The solid burden in the second stage exhaust gas, at the first stage conditions indicated in Figs. 7-5, 7-6, 7-7 and 7-8, is shown in Figs. 7-14, 7-15, 7-16 and 7-17 respectively. The broken contour line in each case represents the limiting solid emission of $0.0040 \text{ kg solid/kg fuel}$, whilst the other broken lines are system constraints defined in Table 6-1. Interstage heat removal is constant for Fig.7-15 at 20 kW , whilst the maximum heat $\dot{Q}_{\text{max}}$ is removed in the other cases. In each case the permitted operating region is shaded. It can be seen from Fig.7-14 that a steam/fuel mass ratio of 0.5 in the first stage and maximum heat removal between stages does not allow the solid emission criterion to be met in the second stage exhaust gas. The second stage solids can be reduced either

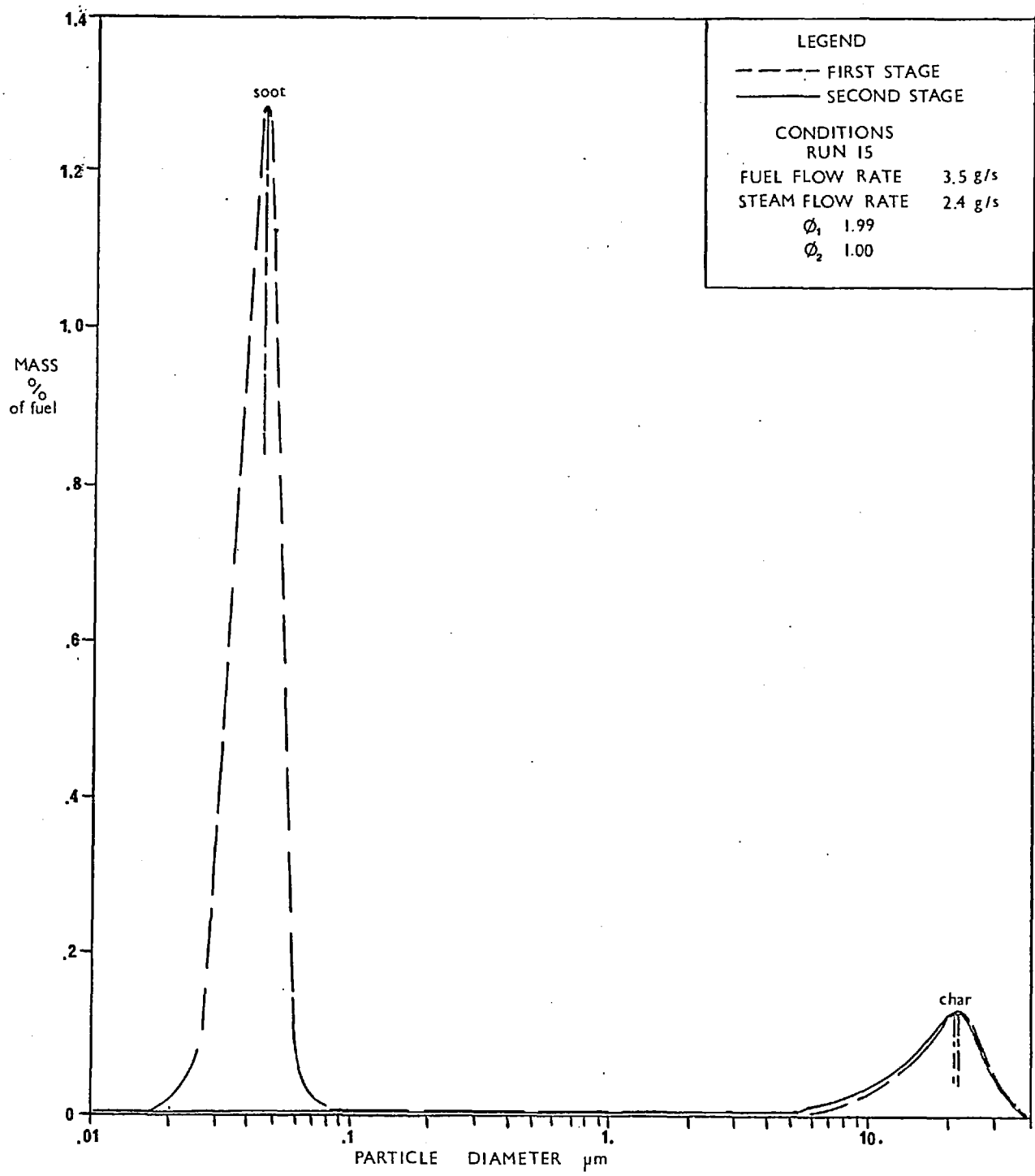


FIG.7-13 PARTICLE MASS DISTRIBUTION AFTER FIRST AND SECOND STAGES

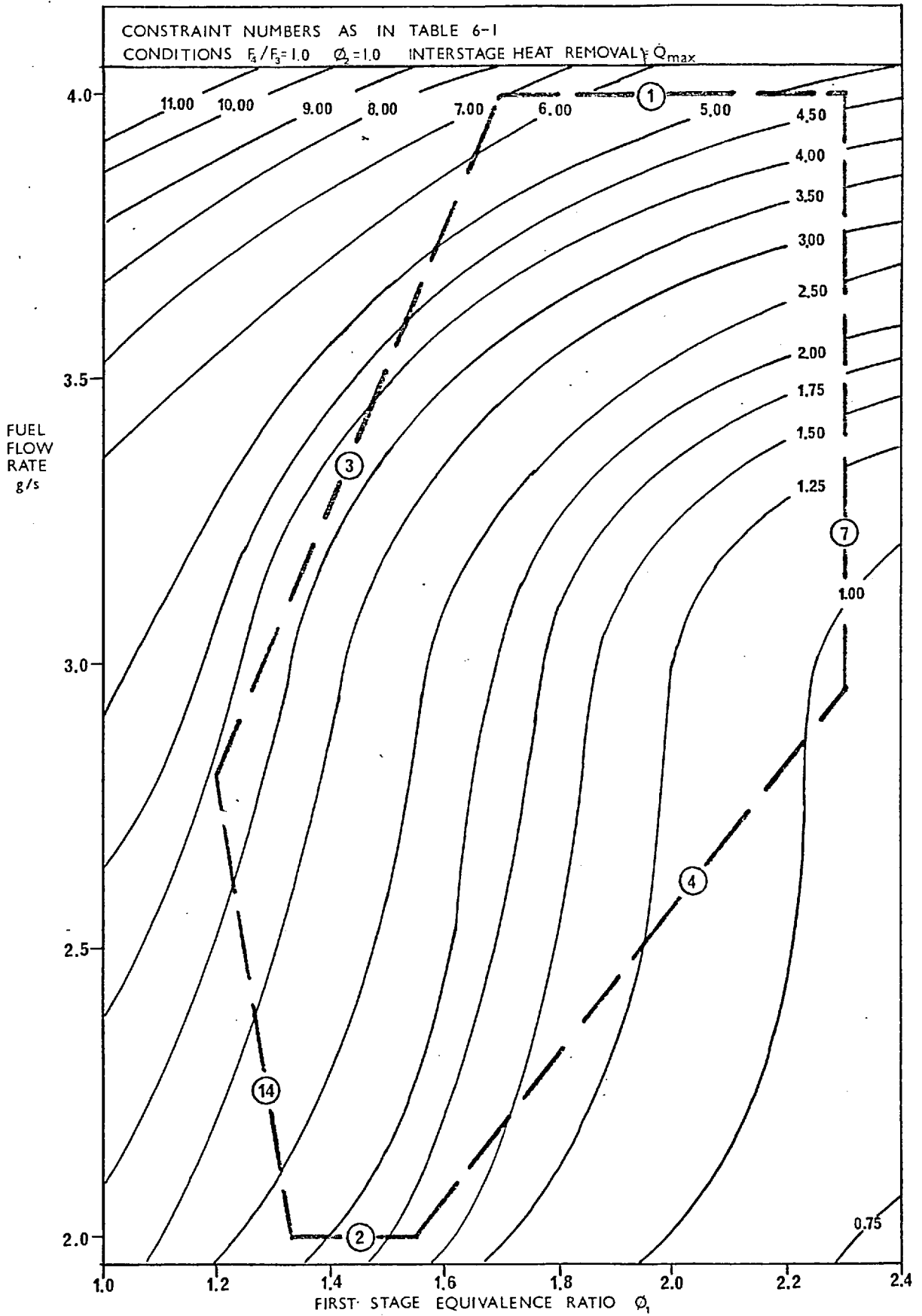


FIG.7-14 SECOND STAGE SOLIDS AS % OF FUEL AT $F_2/F_1 = 0.5$

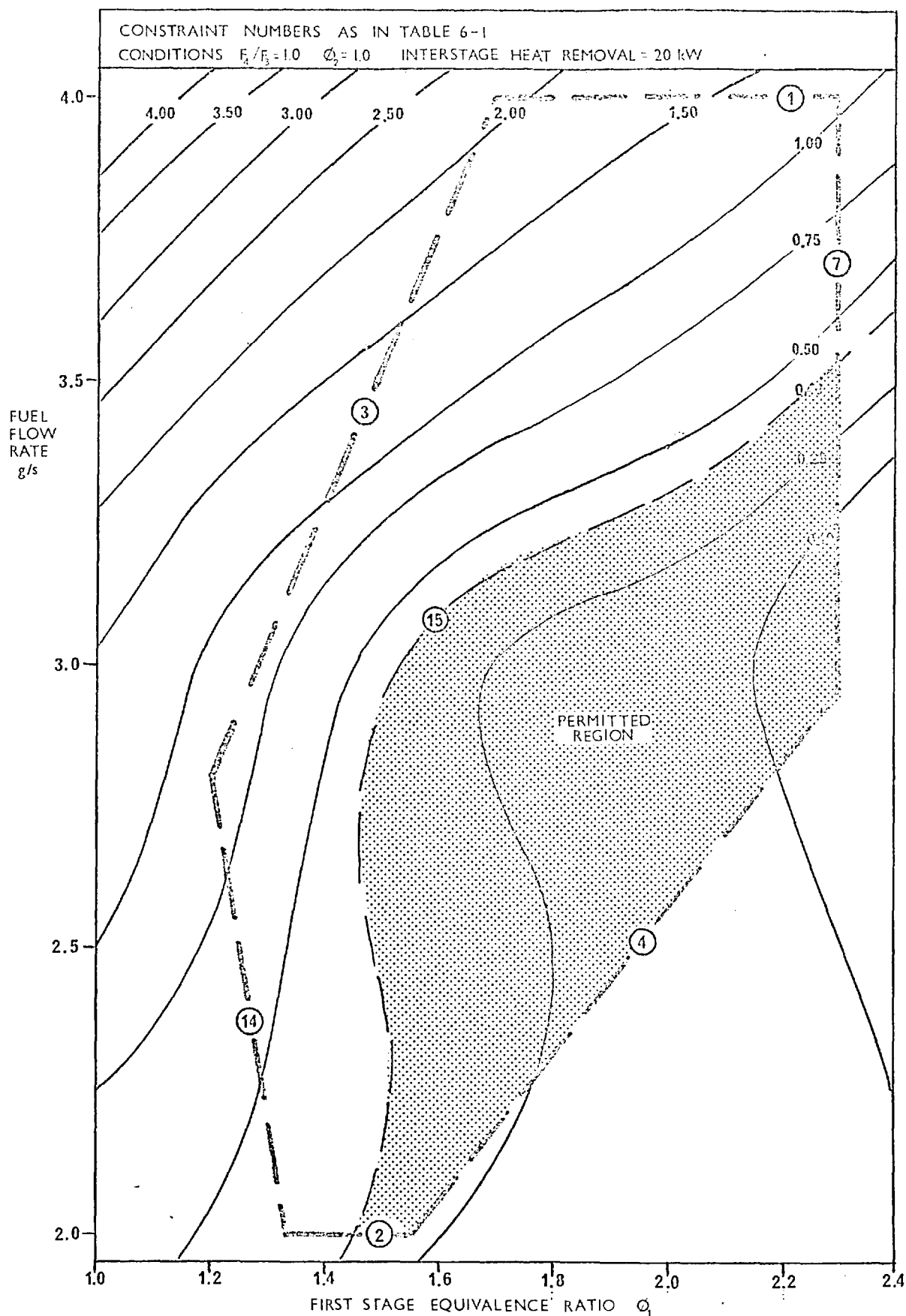


FIG.7-15 SECOND STAGE SOLIDS AS % OF FUEL AT $F_2/F_1 = 0.7$

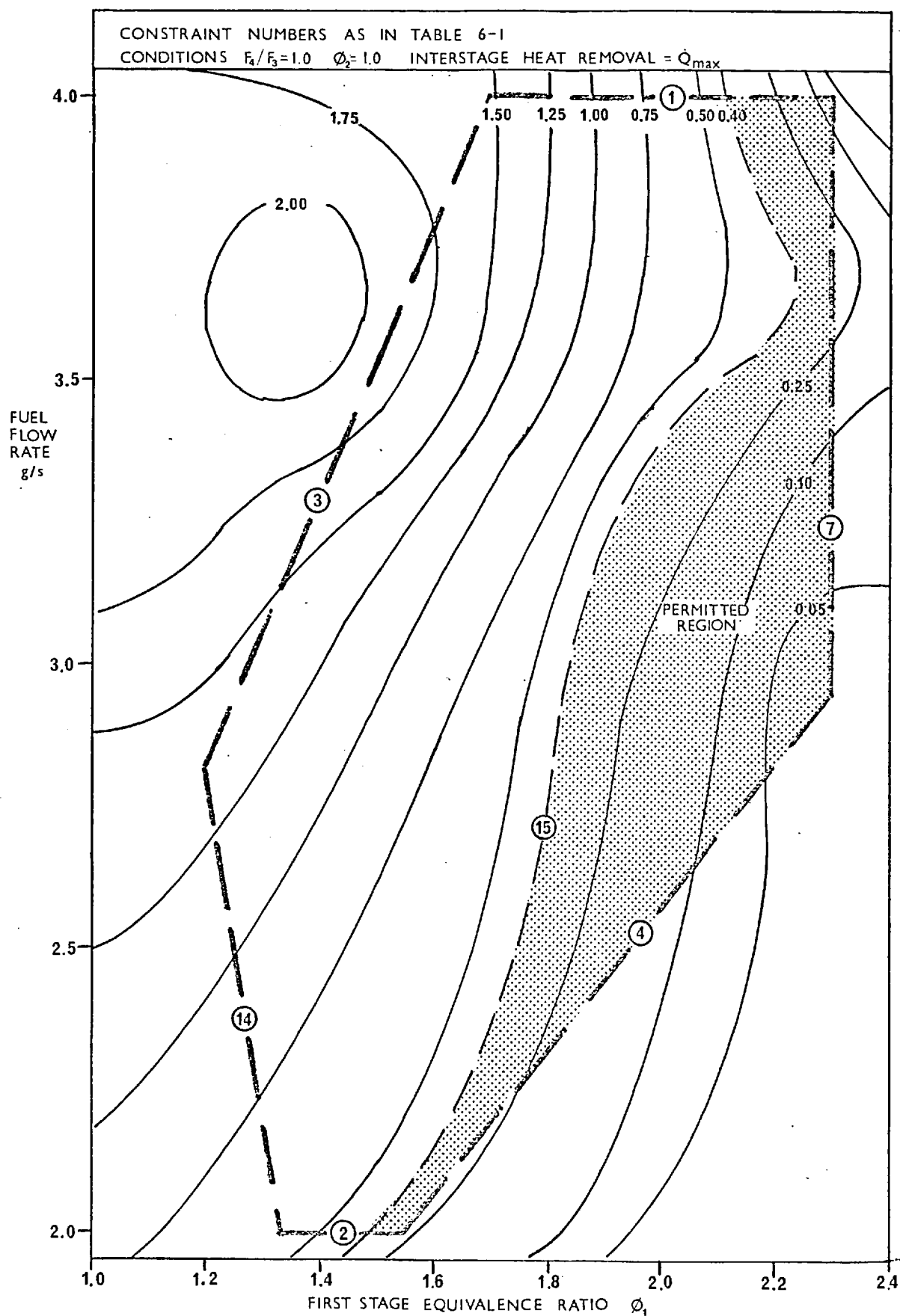


FIG.7-16 SECOND STAGE SOLIDS AS % OF FUEL AT $F_2/F_1 = 0.9$

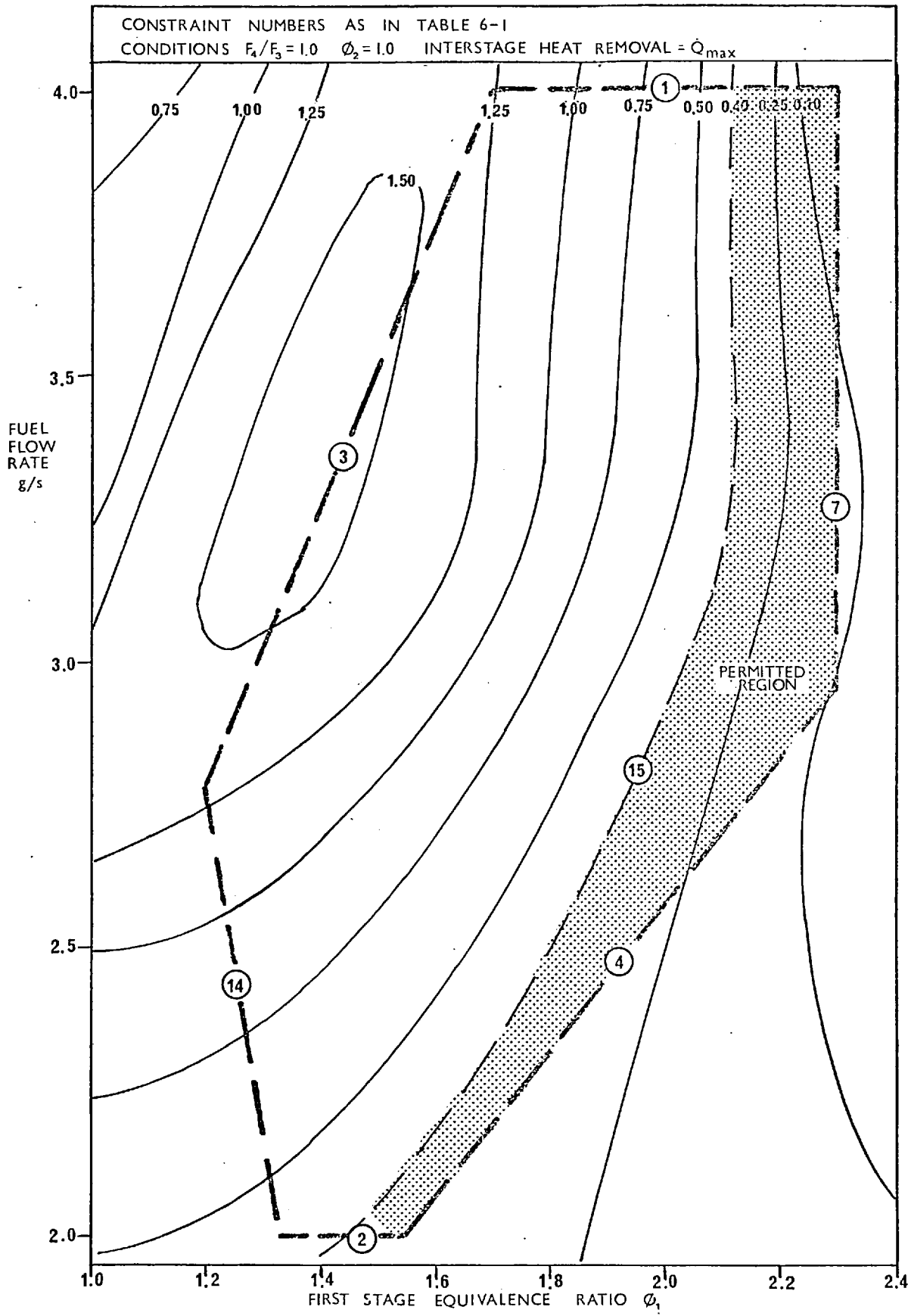


FIG.7-17 SECOND STAGE SOLIDS AS % OF FUEL AT $F_2/F_1 = 1.3$

by increasing the steam/fuel mass ratio in the first stage, or by reducing the interstage heat removal rate. The effect of reducing interstage heat removal is indicated in Fig. 7-18 for a steam/fuel mass ratio of 0.9. It is seen that an improvement of about 25% in solid combustion efficiency, defined as the percent of the solids entering the second stage that are burned, may be achieved by reducing interstage heat removal by 50%. The general effect of reducing interstage heat removal is to shift the contours in the lower half of Figs. 7-14, 7-15, 7-16 and 7-17 to the left. The contours in the upper halves of these figures represent high char weight fractions and are not appreciably affected by second stage operating conditions. It can be seen by comparison of Fig. 7-16 and Fig. 7-17 that little improvement in the second stage solid burden is achieved by increasing the steam/fuel mass ratio in the first stage from 0.9 to 1.3. This is because the atomisation is not significantly improved (see Fig. 7-4). The relation between Fig. 7-7 and Fig. 7-16 is shown in Fig. 7-19 in terms of solid combustion efficiency.

As ϕ_1 decreases, the maximum interstage heat removal increases as indicated in Fig. 5-8. For a fixed fuel flow rate, and $\phi_2 = 1.0$, the second stage temperature will increase as ϕ_1 decreases, and solid combustion will increase. The variation of combustion efficiency with fuel flow rate where $\phi_1 > 2.0$ within the permitted region is explained as follows. At low fuel flow rates, char and soot are formed in approximately equal amounts. As fuel flow rate increases, relatively larger amounts of soot are formed and subsequently burned, thus giving higher efficiencies. As fuel flow

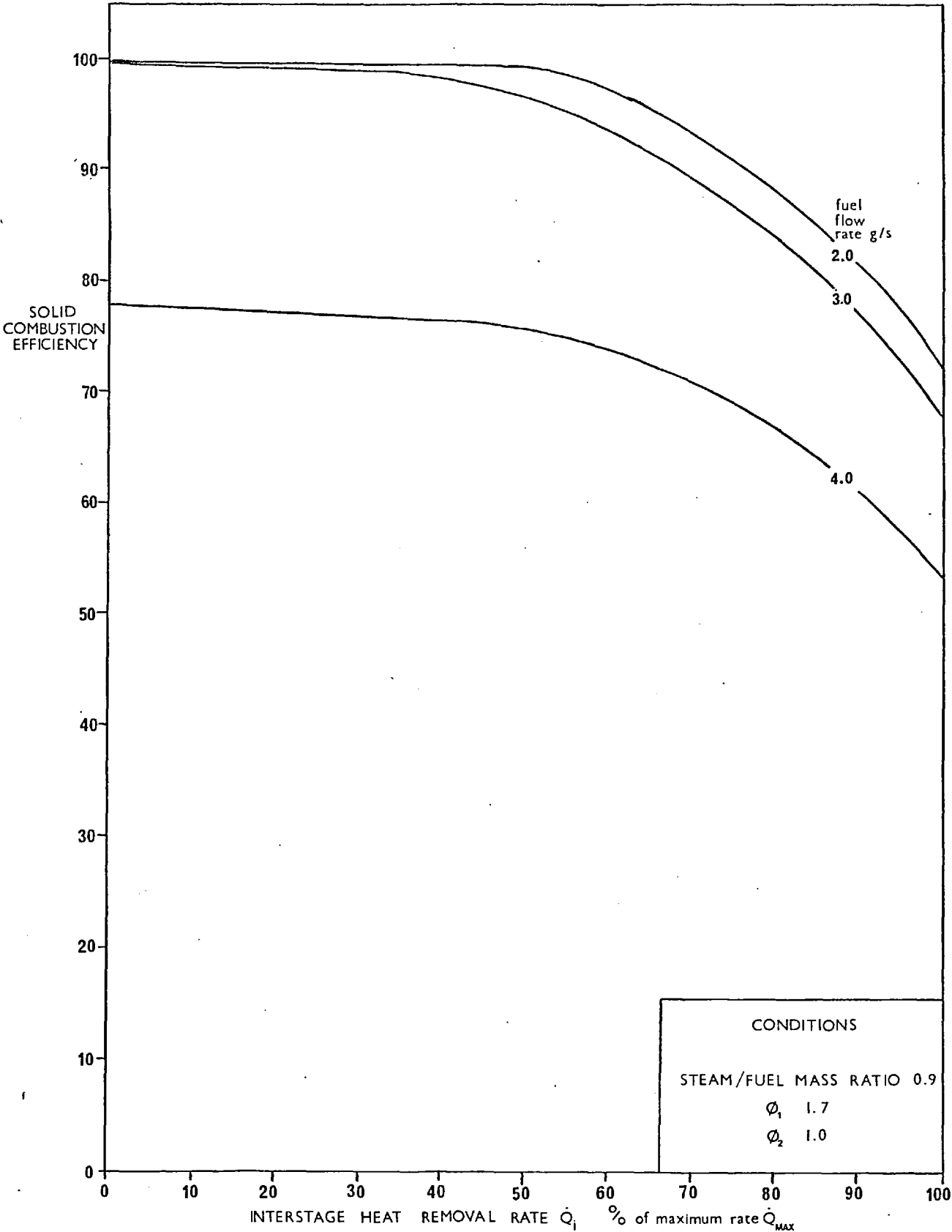


FIG.7-18 THE EFFECT OF INTERSTAGE HEAT REMOVAL ON SOLID COMBUSTION IN THE SECOND STAGE

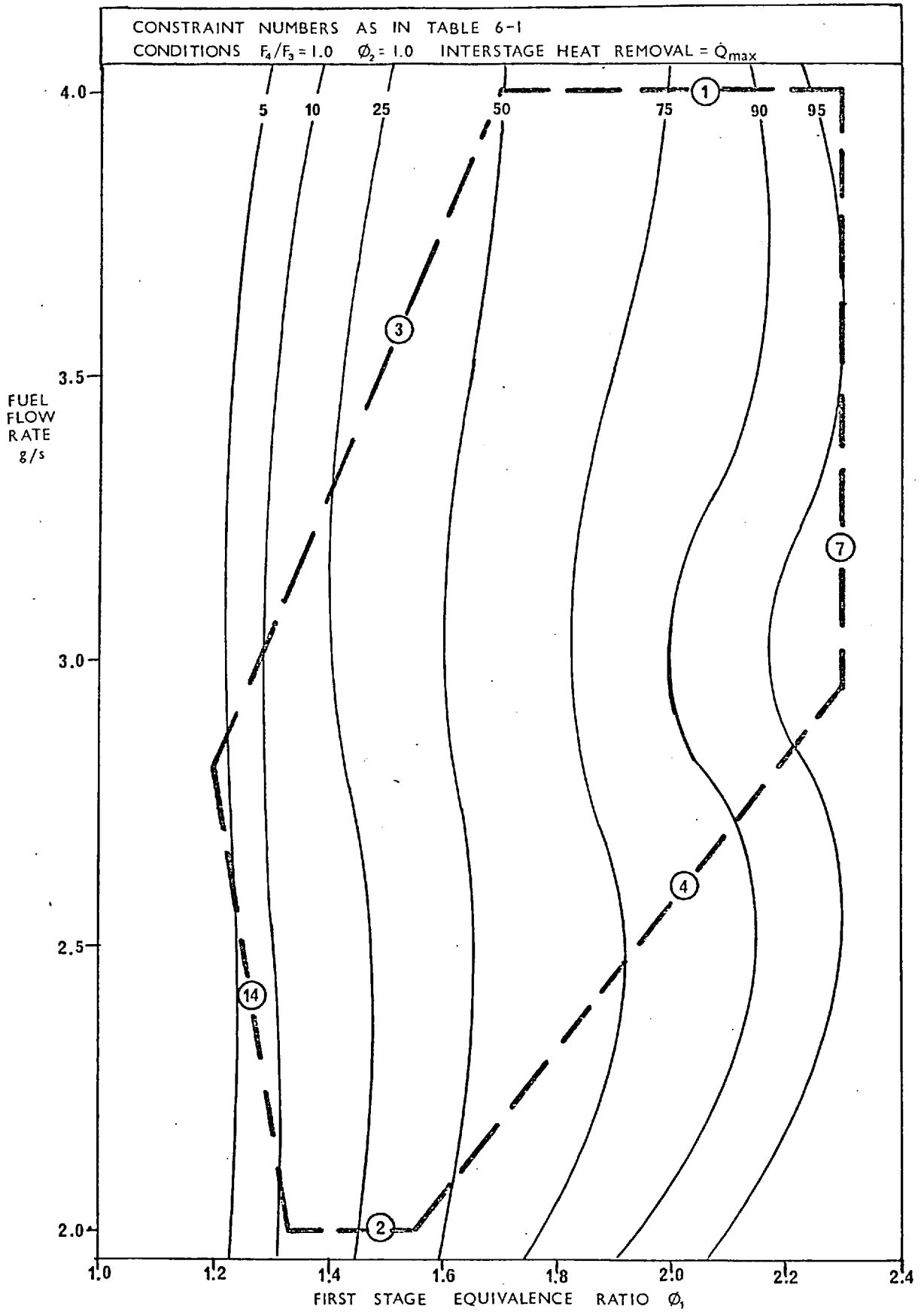


FIG. 7-19 SOLID COMBUSTION EFFICIENCY AT $F_2/F_1 = 0.9$

increases further, char formation predominates and combustion efficiency is subsequently reduced.

The combustion efficiencies for corresponding air and steam atomised conditions are shown in Fig. 7-20. The solid and broken lines denote steam and air atomisation respectively. In all cases, combustion efficiencies with air atomisation are lower because of the relatively higher fraction of char present in the total solids. The highest combustion efficiency at any conditions of about 99% is achieved at a fuel flow of 2.92 g/s and $\phi_1 = 2.3$. These conditions correspond to low maximum interstage heat removal, high second stage temperatures and relatively large percentages of soot in the solids formed in the first stage.

7.6. Combustion Intensity and Thermal Stability.

First stage combustion intensity is shown as a function of first stage equivalence ratio and fuel flow rate in Figs. 7-21, 7-22 and 7-23 and for steam/fuel mass ratios of 0.7, 0.9 and 1.3 respectively. It can be seen from these figures that F_2/F_1 does not change chamber intensity appreciably. Constraints are shown as broken lines and correspond with those indicated in Table 6-1. Constraint 15 indicates a solid emission of 0.0040 kg solid/kg fuel at maximum interstage heat removal $\dot{Q}_{\max}$. It is noted that the permissible operating region shown as the shaded area is considerably reduced by the solid emission criterion. The permissible operating region is enlarged by reducing the level of interstage heat removal as shown in Fig. 7-21, however as F_2/F_1 is reduced, the interstage heat removal must be reduced proportionately more to maintain the permitted region.

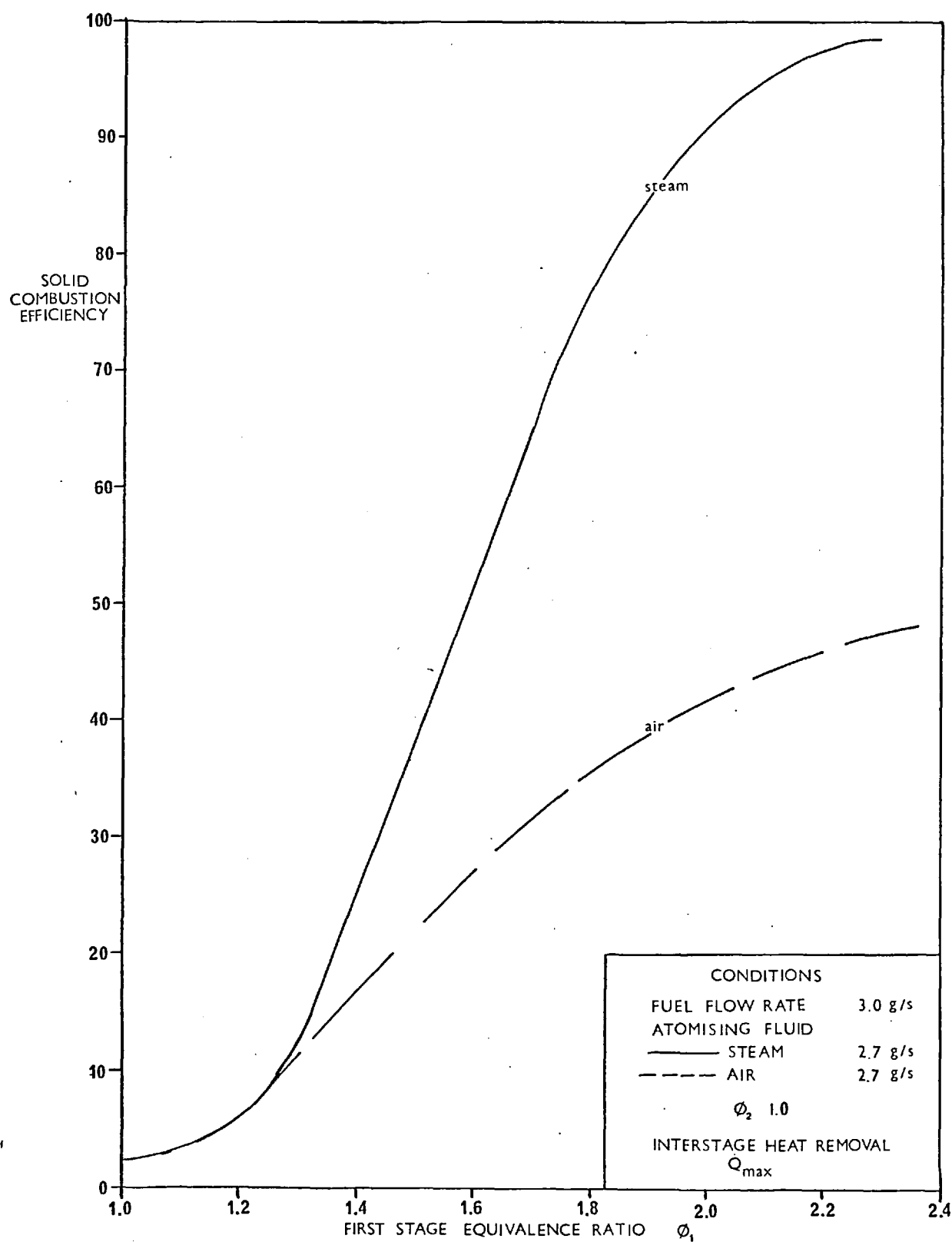
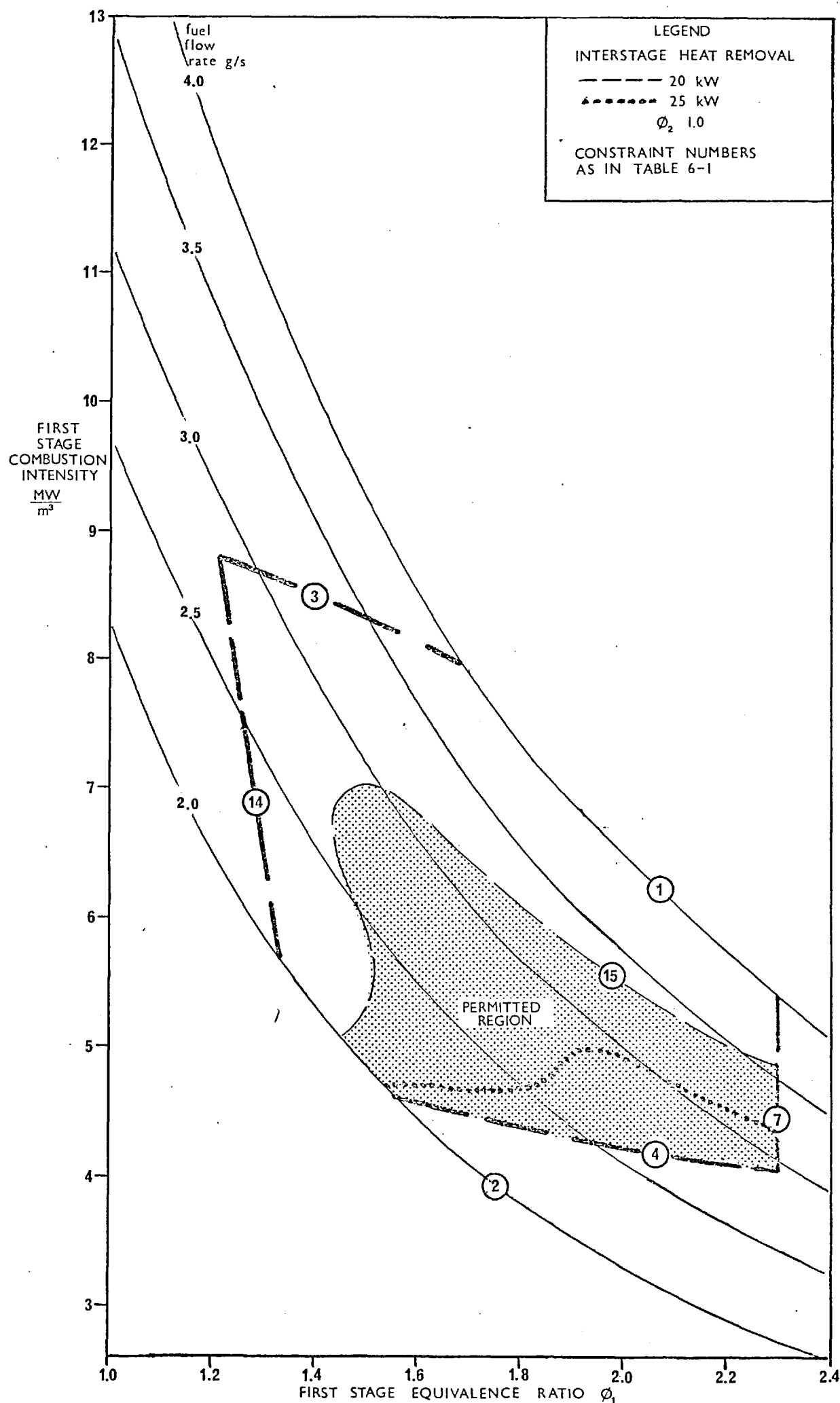
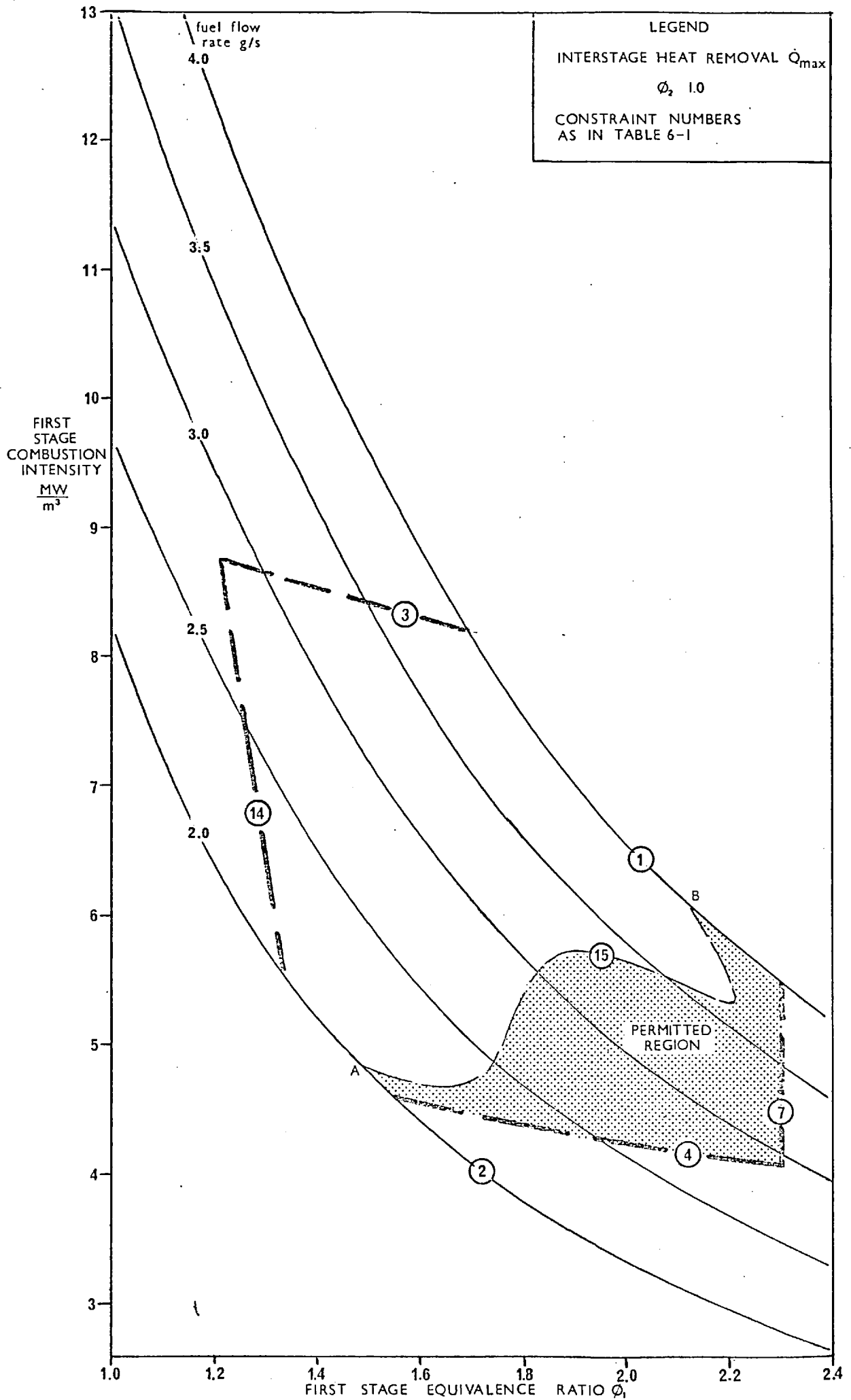
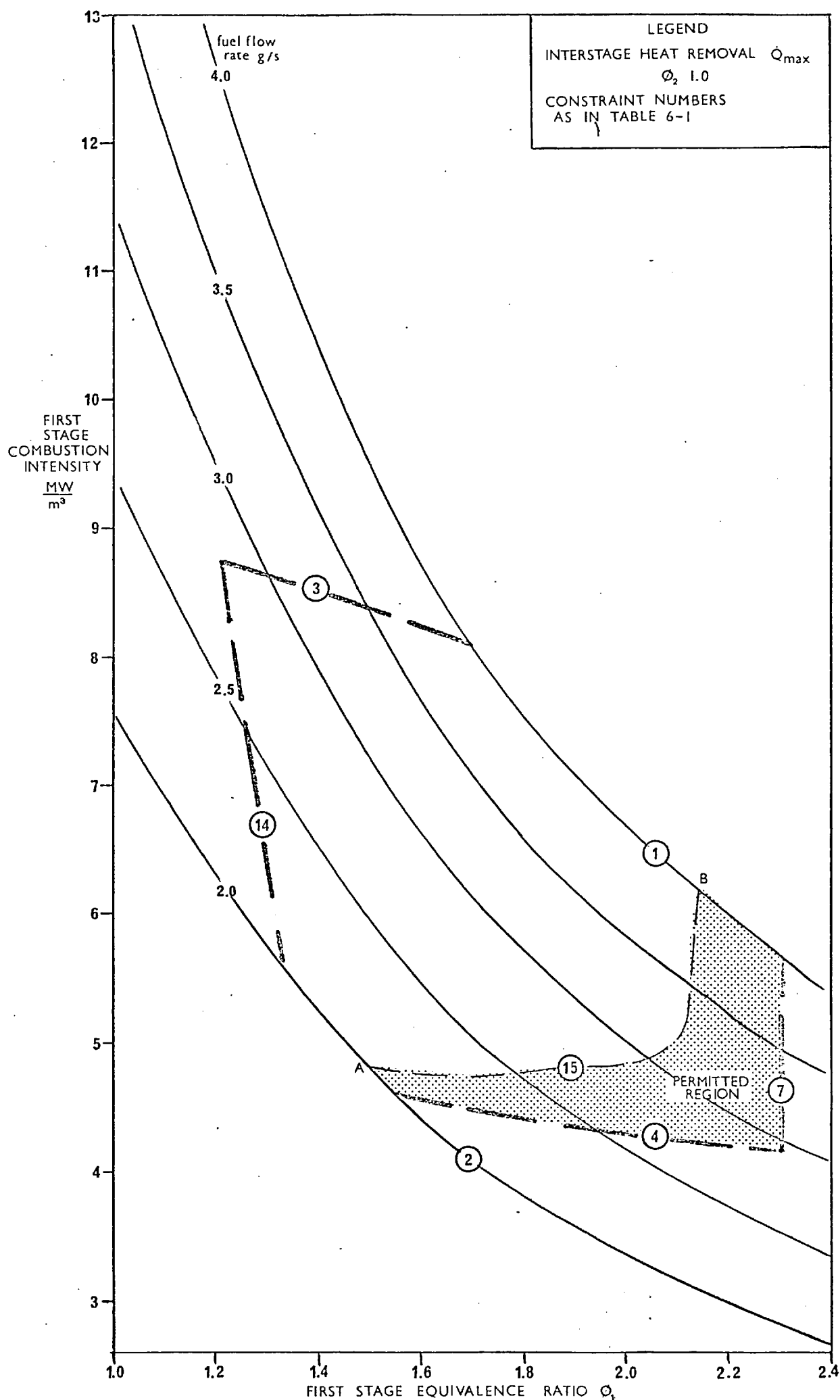


FIG. 7-20 COMPARISON OF SOLID COMBUSTION EFFICIENCY FOR AIR AND STEAM ATOMISATION

FIG.7-21 FIRST STAGE COMBUSTION INTENSITY AT $F_2/F_1 = 0.7$

FIG. 7-22 FIRST STAGE COMBUSTION INTENSITY AT $F_2/F_1 = 0.9$

FIG. 7-23 FIRST STAGE COMBUSTION INTENSITY AT $F_2/F_1 = 1.3$

At $F_2/F_1 = 0.5$, the solid emission criterion cannot be met even when there is no interstage heat removal. The steam/fuel mass ratio must be increased to about 0.55 before the solids can be reduced to 0.0040 kg solid/kg fuel in the exhaust gas.

Second stage combustion intensity is expressed in a similar way in Fig. 7-24 for $\phi_2 = 1.0$, $F_2/F_1 = 0.9$ and maximum interstage heat removal $\dot{Q}_{\max}$. The permitted region is again shown by the shaded area.

It can be seen from equations 5.6 and 5.4 that maximum interstage heat removal $\dot{Q}_R$ by either indirect or direct cooling depends predominantly on first stage equivalence ratio. In Fig. 7-25 the maximum interstage heat removal at blowout as a function of ϕ_1 is expressed as a percent of the heat release in the first stage for both direct and indirect cooling modes based on the heat removal shown in Fig. 5-8. The upper solid curve in Fig. 7-25 corresponds to indirect cooling, the lower solid curve to direct cooling. It can be seen in Fig. 7-25 that the maximum percent of the heat released in the first stage is removed at $\phi_1 = 1.90$. The broken line in Fig. 7-25 indicates maximum interstage heat removal $\dot{Q}_{\max}$ limited by the 600 K constraint, as a percent of the first stage heat release. In this case, the maximum is approached as ϕ_1 approaches 1.0.

7.7. Air Atomisation.

The use of air as an atomising fluid is economically attractive as it is less expensive than steam. It is found that the use of air does not change first stage combustion temperatures appreciably (see Figs. 7-1 and 7-2).

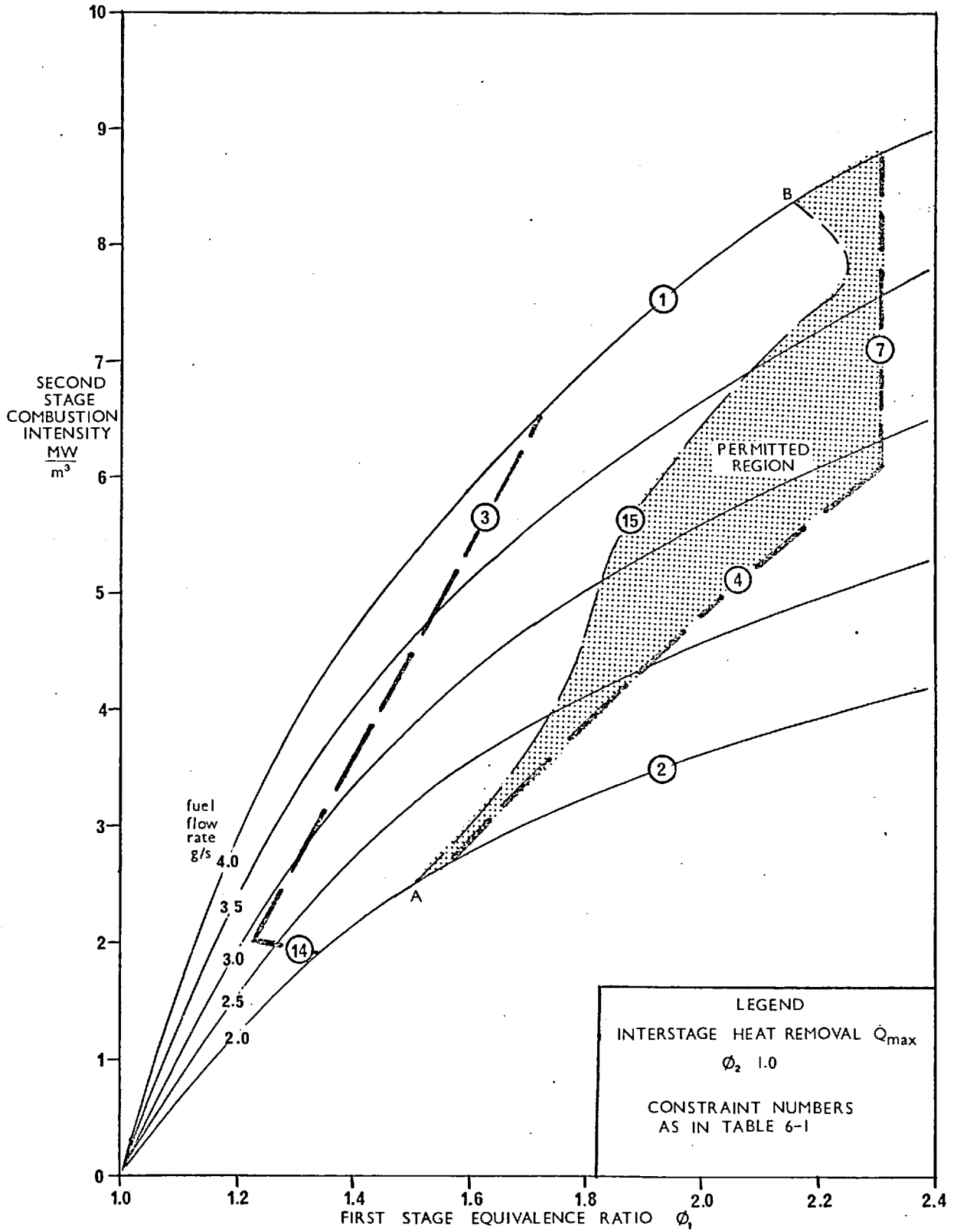


FIG.7-24 SECOND STAGE COMBUSTION INTENSITY AT $F_2/F_1=0.9$

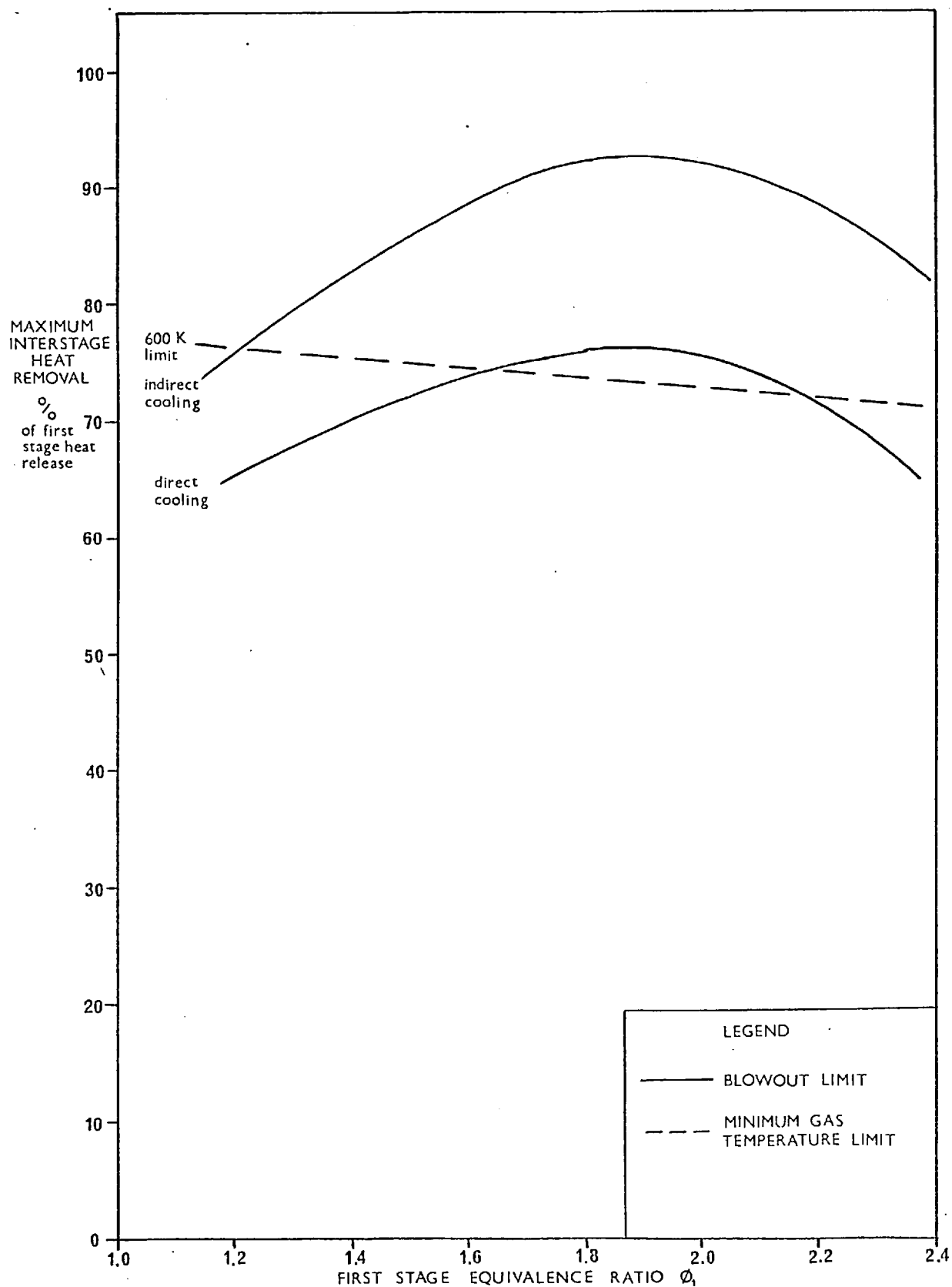


FIG.7-25 MAXIMUM INTERSTAGE HEAT REMOVAL AS % OF FIRST STAGE HEAT RELEASE

When air is used at the same atomising fluid/fuel mass ratio as for steam, it is found that the solid emission from the first stage is increased by three to six times (Fig.7-3). This is a result of poorer fuel atomisation hence most of the increased solid burden is char. The minimum solid emission of 0.0103 kg solid/kg fuel from the first stage is achieved at an atomising air/fuel mass ratio of 1.3 with $F_1 = 2.00$ g/s and $\phi_1 = 1.55$.

The increased solid emission achieved using air atomisation improves the heat transfer characteristics of the gas, mainly by increasing gas emissivity by about 10%. It is not expected that the blowout limits determined under steam atomised conditions will apply to the air atomised case as both gas compositions and mass flow rates are considerably changed. Where the maximum interstage heat removal is limited by a minimum outlet temperature of 600 K for air or steam atomisation it is found that 10-15% more heat can be removed under steam atomised conditions because of the higher first stage combustion efficiency and higher mass flow rates.

At maximum interstage heat removal conditions $\dot{Q}_{\max}$, it is found that the minimum solid emission constraint in the second stage cannot be met using air atomisation in the first stage. The minimum solid emission from the second stage is achieved at $\phi_2 = 1.0$ with first stage operating so as to give minimum first stage solids as above.

7.8. Optimisation of the First Stage.

The optimisation of first stage operation was attempted using various criteria.

7.8.1. Minimum Solid Emission.

The first run was made with the aim of minimising first stage solid emission. This minimisation was essentially a test of the minimisation program as the response surfaces were well known (see Figs. 7-5 to 7-8 inclusive). Minimisation was attempted with respect to fuel flow rate, steam/fuel mass ratio, reverse/forward air ratio, and first stage equivalence ratio. Other variables did not affect first stage solid emission. Five test runs were made starting from different points. The results of these tests are summarised in Table 7-3. The convergence criterion was taken as ± 0.00001 kg solid/kg fuel. It can be seen that in most cases, a minimum solid emission of 0.0051 kg solid/kg fuel was attained, at a fuel flow rate of 2.00 g/s. The value of steam/fuel mass ratio at the minimum varied between 1.02 at $\phi_1 = 1.54$, and 1.30 at $\phi_1 = 1.47$. At these conditions, reverse/forward air ratio was not significant. Reference to Figs. 7-7 and 7-8 indicates that the 0.5% contour lies at the boundary of the feasible region at both these conditions. It was found that small changes in the variables at these points produced no improvement in the solid burden, hence it was assumed that an expected discontinuity in the objective function had been reached. It was further discovered from the designed tests that F_4/F_3 only affected solid loadings at high fuel flow rates and at steam/fuel mass ratios less than 0.9, hence the variable F_4/F_3 remained unchanged once fuel flow rate was reduced or F_2/F_1 increased.

It was concluded from these tests that the first stage could not be operated within the system constraints so

as to satisfy the minimum solid requirement of 0.0040 kg solid/kg fuel in the exhaust gas from the first stage.

7.8.2. Economic Optimisation.

The economic optimisation of the first stage and heat exchange section was attempted using two bases. The first basis was one year (7200 hr.) of operation. Using the economic factors defined in Table 4-3, it was found that the system could not be operated profitably under any conditions. The operating conditions for minimum loss without regard to solid emission were achieved at $F_1 = 2.0$ g/s, $F_2/F_1 = 0.50$, $F_4/F_3 = 0.70$ and $\phi_1 = 1.55$. The interstage heat removal was at the maximum $\dot{Q}_{\max}$ corresponding to an outlet gas stream temperature of 600 K. The operating loss excluding labour costs was £ 683.40.

As fuel was the most expensive raw material, its effect on the system was removed by attempting the economic optimisation on the basis 1 kg of fuel. In this case, labour costs and plant burden were excluded, hence only operating costs were compared with the value of the export steam and increased capital costs arising from increased plant capacity were not considered. The operating costs and export steam values are plotted in Fig. 7-26 as a function of ϕ_1 for an electrical power cost of £ 4.17×10^{-3} /unit, atomising steam cost of £ $.14 \times 10^{-3}$ /kg and an export steam value of £ $.92 \times 10^{-3}$ /kg. It can be seen from Fig. 7-26 that the best operating conditions are at $\phi_1 = 1.0$ for $F_2/F_1 = 0.5$.

The optimum corresponding to a profit of £ $.44 \times 10^{-3}$ /kg fuel, determined using the minimisation

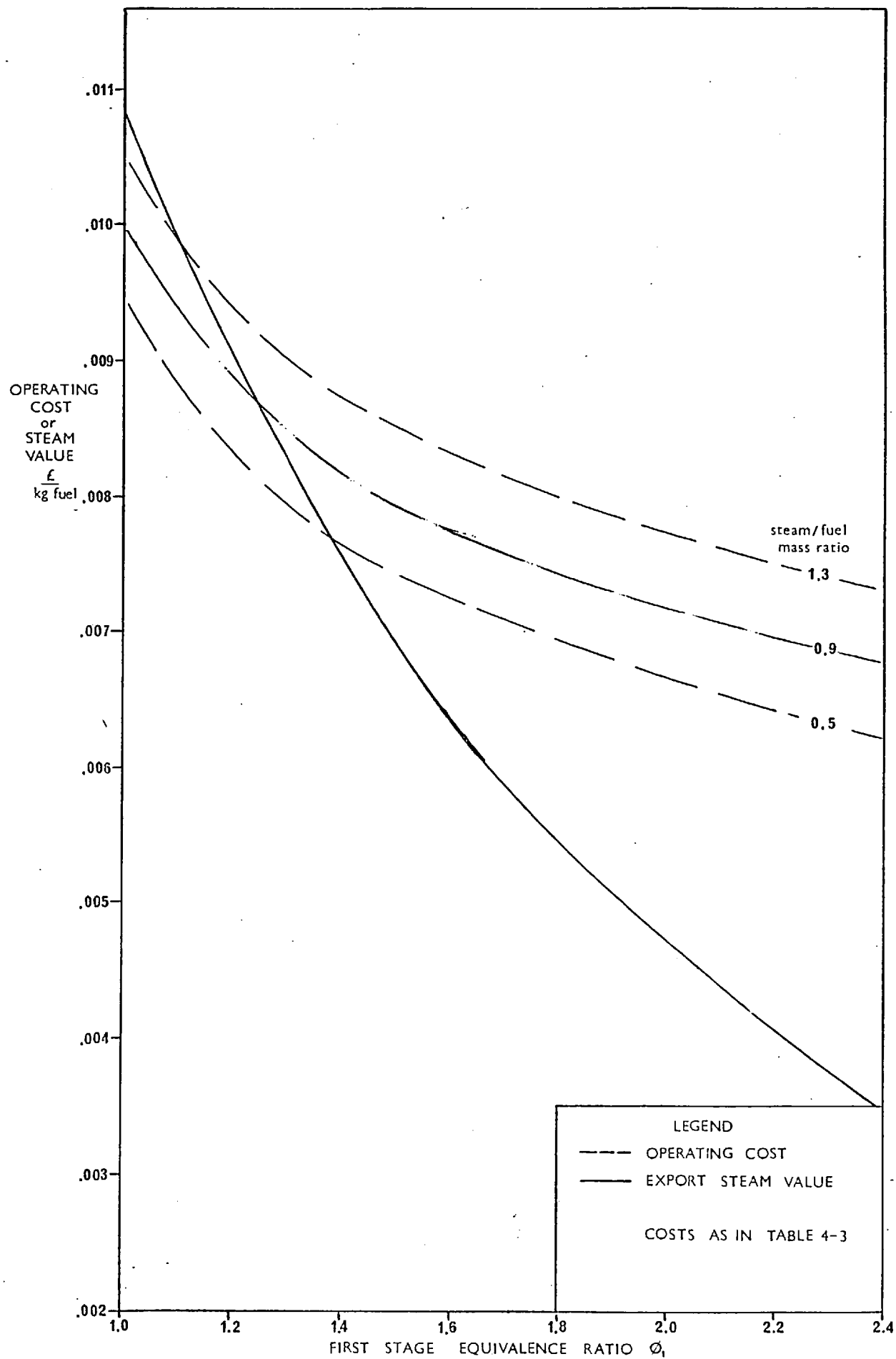


FIG.7-26 COMPARISON OF EXPORT STEAM VALUE AND OPERATING COSTS

program was at $\phi_1 = 1.26$ which lies on the intersection of constraints 3 and 14 in Table 6-1. The steam/fuel mass ratio was 0.5 and reverse/forward air ratio was 0.7. The fuel flow rate was 2.8 g/s. As the basis was 1 kg of fuel, it is surprising to find fuel flow rate significant. This may be explained by the fact that as fuel flow rate increases, evaporation efficiency hence molar flow rate of the exhaust gases per kg of fuel decreases thus lowering extraction costs. It is also noted that at $\phi_1 = 1.26$, equivalence ratio is minimised within the feasible region and maximum interstage heat removal $\dot{Q}_{\max}$ per kg of fuel is achieved.

As the value of export steam increases, the break-even point defined by the intersection of the curves in Fig. 7-26 moves to the right. Optimal operation is still achieved at $\phi_1 = 1.26$. Increasing electrical power, cooling or feed water costs shift the break-even point to the left.

7.8.3. Combustion Volume and Intensity.

The effect of increasing or decreasing first stage combustion volume has been analysed. It was found that solid formation was the only variable significantly affected by these changes. The effect of first stage volume on solid formation is shown in Fig. 7-27 for a steam/fuel mass ratio of 0.9. The solid lines show the dependence on fuel flow rate at $\phi_1 = 1.7$, the broken lines the dependence on ϕ_1 at $F_1 = 3.00$ g/s.

It can be seen that solid emission can be reduced by as much as 15% by increasing chamber volume by 25% at high fuel flow rates. Little change is observed for lower fuel flow rates. It is noticed that slightly greater changes in

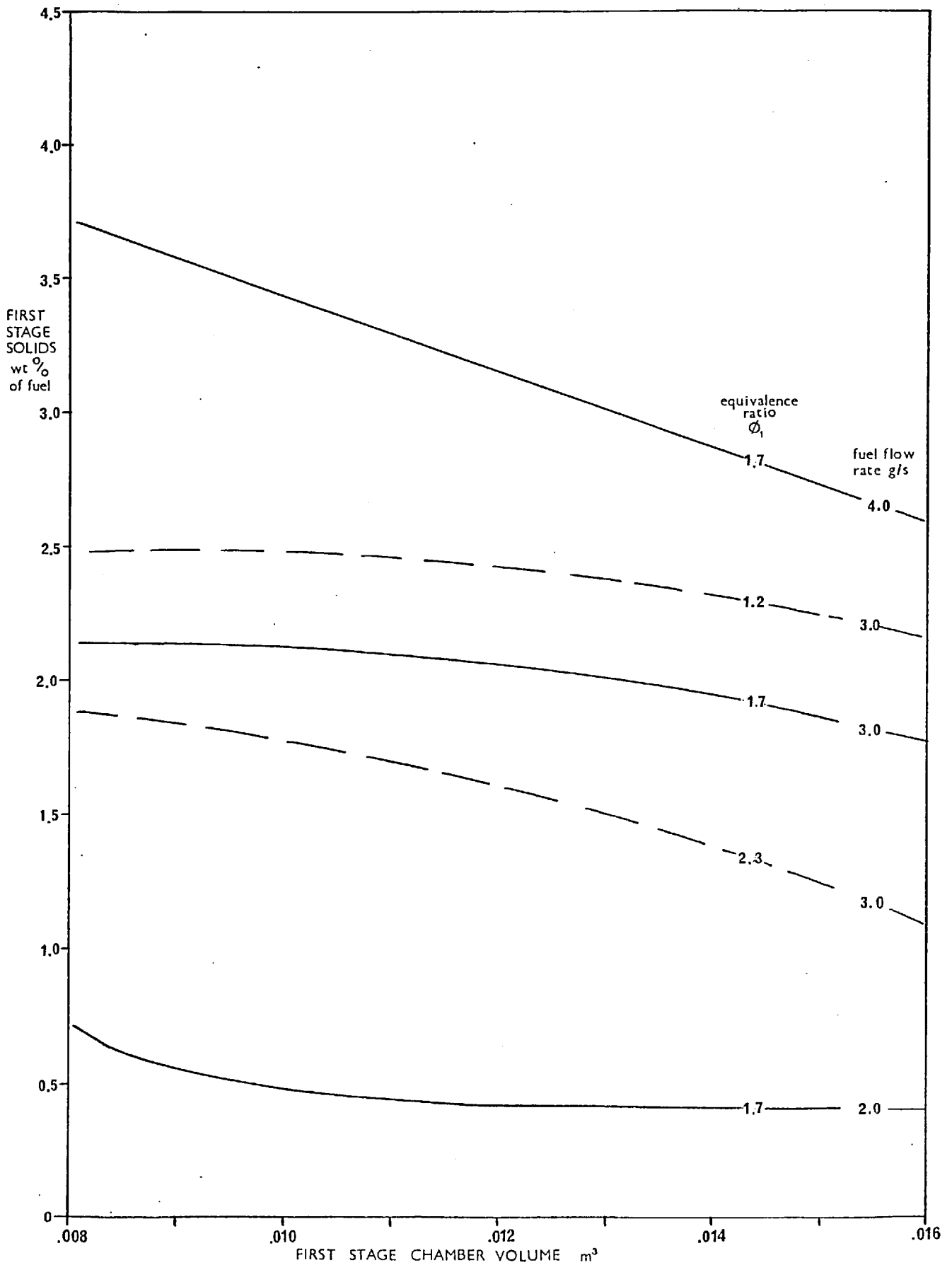


FIG.7-27 THE EFFECT OF FIRST STAGE VOLUME ON FIRST STAGE SOLIDS

solid formation occur at low ϕ_1 than at high values. Changing volume affects only char formation as the mean residence time in the chamber is altered.

Heat release rates in the first stage are not significantly affected by changes in chamber volume. Changes in evaporation efficiency result in changes of less than 1% in the heat released. Chamber intensity therefore varies hyperbolically as chamber volume. That is, an increase or decrease of 25% in chamber volume yields a 20% decrease or a 33% increase respectively in combustion intensity.

7.9. Optimisation of the Two Stage System.

The inclusion of the second stage in the optimisation problem increased the number of variables from four to six and increased the computational time required for one function evaluation by a factor of 3. As between 40 and 60 function evaluations were required to achieve the optimal first stage conditions involving 400 s to 600 s computation time, it was expected that the six variable problem would require at least 2000 s i.e. (0.55 hr.) to complete.

To simplify the problem and reduce the computational time, the number of variables was reduced to four by keeping $\phi_2 = 1.0$ and $F_4/F_3 = 0.7$ constant. These conditions were known to be satisfied at the economic optimum. Although F_4/F_3 only affects first stage solids when atomisation is poor, lowest solids may be achieved for any operating conditions when F_4/F_3 is a minimum, i.e. 0.7. Second stage intensity and solid combustion are maximised within the permitted region at $\phi_2 = 1.0$.

7.9.1. Economic Optimisation.

The economic optimisation was attempted as for the first stage. Equation 4.90, used to predict the scrubber water requirements, was included only when the solid burden in the second stage exhaust gases exceeded 0.0040 kg solid/kg fuel. Heat removal in or after the second stage was constrained by the minimum stack gas temperature of 500 K. As with the first stage, optimisation was attempted on a basis of one year's operation, or per kg of fuel.

Two results were immediately apparent. As before, when the basis was one year of operation, it was found that there were no conditions within the permitted operating region at which the system could be operated profitably. It was also found that the solids were not successfully constrained at $m_{T_2} \leq 0.0040$ kg solids/kg fuel. The constraint was not met because the penalty incurred by exceeding it consisted only of the inclusion of a venturi-scrubber system. It was more economic to operate the first stage optimally at $\phi_1 = 1.26$, and emit 0.0140 kg solids/kg fuel from the second stage, than to conform to the constraint for solid emission.

The problem was overcome by introducing a penalty function P_f to the objective function when $m_{T_2} > 0.0040$ kg solids/kg fuel.

$$P_f = K_p (0.0040 - m_{T_2}) \quad 7.1$$

where K_p is a positive constant. The size of the constant K_p must be determined by trial and error and depends on the magnitude of the objective function. It should be large enough to ensure that P_f is the same order of magnitude as the objective function when $m_{T_2} > 0.0040$ kg solids/kg fuel.

Analysis of the system with the penalty function included did not yield consistent results for the optimal operating conditions in all cases. As before when the system was analysed on an annual basis, a minimum loss of £ 668.60 corresponded with minimum $F_1 = 2.00$ g/s. In this case the steam/fuel mass ratio was between 0.86 and 0.89 and $\phi_1 = 1.49 - 1.51$. This result indicates that operating costs are minimised and steam production is not maximised.

Using 1 kg of fuel as a basis, the same minimum was found. In both cases, the interstage heat removal was at the maximum for these conditions. Referring to Fig.7-22, it can be seen that the operating conditions corresponding with the minimum operating loss lie in the leftmost part of the permitted region. These operating conditions also allow the maximum interstage heat to be removed per kg of fuel, as ϕ_1 takes the lowest permitted value (point A) but do not correspond to the maximum annual heat removal (point B). It can be seen from Fig.7-24 that these operating conditions give minimum second stage intensity. Export steam value and fuel cost are the most significant economic factors appearing in the analyses. As the operating costs associated with a small installation of this type are disproportionately high in comparison with product value, the value of export steam was arbitrarily increased from £ $.92 \times 10^{-3}$ /kg to £ 1.5×10^{-3} /kg to assess the effect on the optimal operation.

Depending on the starting point of the search, minimum losses were determined in the region $\phi_1 \approx 1.9$ at a steam/fuel mass ratio of 0.85 - 0.92 and a fuel flow rate $F_1 \approx 3.20$ g/s, as well as at $\phi_1 = 2.17$, $F_1 = 4.00$ and

$F_2/F_1 = 0.86 - 0.92$: In each case interstage heat removal was at the maximum $\dot{Q}_{\max}$. The former point appeared to be a global minimum with a net loss of £ 500.50 compared with a net loss of £ 541.80 at the latter conditions. This can be explained by reference to Fig. 7-22 and Fig. 7-23. It can be seen in Fig. 7-22 that within the permitted operating region there are three local/intensities at $\phi_1 = 1.49, 1.92$ and 2.17 . In Fig. 7-23, the maxima are at 1.51 and 2.17 , with a possible local maximum at $\phi_1 = 1.85$. It is seen that two of these maxima at $\phi_1 = 1.92$ and 2.17 , correspond with the minimum losses determined using the minimisation program. These results emphasize the need to have some knowledge of the shape of the objective function before the minimisation procedure is applied.

Similar multiple answers are obtained when maximum first stage intensity is the criteria, global maximum intensity being achieved at $\phi_1 = 2.17$, $F_1 = 4.00$ g/s and $F_2/F_1 = 0.92$ in the first stage. A single maximum at $\phi_1 = 2.3$, $F_1 = 4.00$ g/s and $F_2/F_1 = 0.85 - 0.93$ is achieved for overall stoichiometric conditions in the second stage.

Interstage heat transfer area is shown as a function of ϕ_1 and F_1 in Figs. 7-11 and 7-12. It can be seen that within the permitted region, minimum heat transfer surface of 1.14 m^2 is required for operation at $\phi_1 = 1.55$ and $F_1 = 2.0$ g/s with a steam/fuel mass ratio of 1.3 . It can also be seen that when the system is optimised at a steam value of £ $.92 \times 10^{-3}$ /kg, the required interstage heat transfer area is near the minimum even though the maximum interstage heat removal per kg of fuel is achieved at $\phi_1 = 1.50$.

7.9.2. Combustion Volume and Intensity.

Changes of up to $\pm 25\%$ in the second stage chamber volume at $\phi_2 = 1.0$ do not significantly alter system performance with the exception of chamber intensity which varies hyperbolically with chamber volume as before. A volume increase of 25% will reduce solid emission by only $1-2\%$ at any operating conditions. It is apparent therefore, that once the soot has been burned, no reduction in solid emission can be achieved by increasing chamber volume. It appears that most of the soot is burned in the well mixed section of the second stage, and that very little solid particle combustion occurs in the plug flow region as oxygen concentrations in this section are very low. There is no advantage to be gained by staged addition of air in the plug flow section as the majority of solids that are expected to burn (i.e. soot) have previously burned out in the well mixed section. It is evident that the best operational policy to be followed is to minimise char formation in the first stage by the use of steam/fuel mass ratios > 0.7 , and to attempt to burn soot in the second stage.

7.9.3. System Control.

From an operational point of view, the conditions giving the best economic return are not those that offer the best control possibilities. This can be seen clearly in Fig. 7-22 and Fig. 7-24 where the operating conditions corresponding to the optimum (points A) for a steam value of $\& .92 \times 10^{-3}/\text{kg}$ lie in relatively narrow peninsular regions of the permitted area. Almost any fluctuation in

fuel or air flow rates will cause the system to move outside the permitted operating region. This problem can be alleviated in this case by reducing the interstage heat removal thus widening the permitted region.

If maximum first stage intensity is required (Point B, Fig. 7-22) a similar control problem arises. In this case the problem can be solved by increasing the steam/fuel mass ratio to expand the permitted region in the vicinity of the maximum, as shown in Figs. 7-22 and 7-23.

CHAPTER 8.CONCLUSIONS.

The following conclusions can be formed, based on this study of residual fuel oil combustion in a two stage system.

GENERAL.

1. The system can be successfully described in terms of mathematical models based on experimental measurements. Models derived entirely from theory do not describe the system accurately in all cases, and require the support of experimental data.

2. Gas temperatures and compositions in both first and second stages correspond within experimental error with those predicted on the basis of water gas shift equilibrium. This indicates that this equilibrium is achieved in both stages. It can be shown from reaction rate data that equilibrium is attained in 3-10 ms in the second stage.

FIRST STAGE.

3. Two types of solid products are formed in the first stage. Char formation can be predicted from spray evaporation theory for burning drops. Soot formation m_s (kg solid/kg fuel) can be predicted by the use of the following equation

$$m_s \pm 0.0061 = (0.0184 - 0.161 \times 10^{-4}T) F_1^2 + 0.417 \times 10^{-4}T F_1 - 0.1132$$

where F_1 is fuel flow rate (g/s) and T is combustion temperature (K). The standard deviation is ± 0.0030 , the error in m_s above is expressed at the 95% confidence level for 25 degrees of freedom.

Char formation is reduced by increasing the steam/fuel mass ratio (F_2/F_1). However, little reduction in char formation is achieved by increasing F_2/F_1 beyond 0.9. Char formation is reduced by increasing the mean residence time in the chamber by reducing either the mass flow rate or the temperature. Soot formation is slightly reduced by decreasing F_1 .

4. System performance is improved by the use of steam as opposed to air atomisation at the same atomising fluid/fuel mass ratio. Better atomisation is achieved with steam, hence fuel evaporation efficiency and chamber intensity are increased.

First stage solid emission always exceeds the statutory limit of 0.0040 kg solid/kg fuel. Minimum first stage solid emission of 0.0051 kg solid/kg fuel is achieved by the use of steam atomisation at a first stage equivalence ratio $\phi_1 = 1.47 - 1.55$, a fuel flow rate $F_1 = 2.0$ g/s and a steam/fuel mass ratio $F_2/F_1 = 1.0 - 1.3$. At these conditions the reverse/forward air ratio F_4/F_3 is not significant. The minimum solid emission using air atomisation is 0.0103 kg solid/kg fuel at $\phi_1 = 1.55$, $F_1 = 2.0$ g/s, $F_2(\text{air})/F_1 = 1.30$ and $F_4/F_3 = 0.7$.

First stage solid emission can be reduced by up to 15% by increasing chamber volume by 25%, however combustion intensity is reduced by 20%. Changes in second stage volume do not significantly affect solid burnout in the second stage.

Maximum first stage intensity conforming with the statutory limit of 0.0040 kg solid/kg fuel solid emission from the second stage is achieved at $\phi_1 = 2.17$, $F_1 = 4.0$ g/s and $F_2/F_1 = 0.9$. Maximum second stage intensity at a second stage equivalence ratio $\phi_2 = 1.0$ is achieved at $\phi_1 = 2.3$, $F_1 = 4.0$ g/s, and $F_2/F_1 = 0.9$.

INTERSTAGE.

5. The absolute maximum interstage heat removal rates $\dot{Q}_R$ are limited by thermal instability in the second stage. Maximum heat removal rates can be measured by the use of spray water cooling.

The maximum heat removal rate per kg of fuel using spray cooling is

$$\dot{Q}_R \pm 2190 = 37759 - 11700 \phi_1 \frac{\text{kJ}}{\text{kg fuel}}$$

The standard deviation is ± 1060 , the error in $\dot{Q}_R$ above is expressed at the 95% confidence level for 23 degrees of freedom. Maximum heat removal as a percent of first stage heat release is 76% at $\phi_1 = 1.9$.

Chemical reactor theory can be used to predict maximum heat removal $\dot{Q}'_R$ achieved by indirect heat transfer (without spray cooling). The maximum indirect heat removal $\dot{Q}'_R$ which exceeds $\dot{Q}_R$ is

$$\dot{Q}'_R \pm 1964 = 42464 - 12500 \phi_1 \frac{\text{kJ}}{\text{kg fuel}}$$

The standard deviation is ± 927 , the error in $\dot{Q}'_R$ above is expressed at the 95% confidence level for 23 degrees of freedom. The maximum heat removal as a percent of the first stage heat release is in this case 92% at $\phi_1 = 1.9$.

6. Interstage heat removal is gas-side controlled, but cannot be successfully predicted on the basis of existing radiant and convective heat transfer expressions. Details of a method for predicting interstage heat transfer in this system are given in Chapter 5.

SECOND STAGE.

7. The amount of solid emitted from the second stage chamber is related to operating conditions in both chambers as well as to interstage heat removal.

Soot can be effectively burned in the second stage at stoichiometric conditions whilst char cannot. Most of the soot is burned in the well mixed portion of the second stage. Combustion of both soot and char is increased by reducing the interstage heat removal. Char can be more successfully controlled by limiting its formation in the first stage than by burning it in the second stage.

At interstage heat removal rates corresponding with the minimum second stage inlet gas temperature (600 K) suitable for generation of export steam, it is possible to reduce solids in the second stage exhaust to less than 0.0040 kg solid/kg fuel for $F_2/F_1 > 0.6$. This solid emission limitation in the second stage cannot be met using air atomisation in the first stage.

OPTIMISATION.

8. The system can be optimised on an economic basis for the generation of export steam at £ .92 per 1000 kg using existing non-linear minimisation techniques. It is necessary, however to have some knowledge of the shape of the objective function to be analysed.

Economic optimisation of the combined first stage and heat exchange section only, indicates that best operation corresponding to minimum annual loss is achieved at $\phi_1 = 1.55$, $F_1 = 2.0$ g/s, $F_2/F_1 = 0.5$, $F_4/F_3 = 0.7$ and maximum interstage heat removal of 19.4 MJ/kg fuel.

9. The optimisation of the two stage system on an annual basis within the constraint of solid emission less than 0.0040 kg solid/kg fuel, indicates that the system is not economically viable for the generation of export steam at £ .92 per 1000 kg under any conditions. Best operating conditions for minimum loss are achieved at $\phi_1 = 1.50$, $F_1 = 2.0$ g/s, $F_2/F_1 = 0.9$. Reverse/forward air ratio is not significant.

These optimal conditions are maintained up to a price level for export steam of approximately £ 1.50 per 1000 kg. Beyond this level, the best operating conditions for minimum loss become $\phi_1 = 1.9$, $F_1 = 3.2$ g/s and $F_2/F_1 = 0.9$.

CHAPTER 9.

RECOMMENDATIONS.

It should be noted that the preceding study has been made to illustrate the methods of modelling and optimising a combustion system, and not to provide an economic justification for the two stage process. The economics used in the analysis cannot be applied with a high degree of confidence to this pilot scale system, since the ratio of plant burden to the value of export steam is much larger than that for full scale plants. For this pilot scale system, the ratio is of the order of 3, whereas for a full scale generating plant, the ratio is of the order 0.3.

Based on the preceding study, the following recommendations can be made for the scale up of this system.

1. The system should be operated at conditions slightly removed from those determined at the economic optimum, in that the interstage heat removal should be decreased to allow easier system control at the boundary of the permitted operating region.
2. The interstage heat transfer section should be redesigned to give greater control over interstage heat removal.
3. The second stage chamber should be extensively redesigned. In particular, the air inlet jets should be placed so that combustion air is projected directly into the chamber to improve mixing. The plug flow section should be removed entirely as it does not improve combustion efficiency.

4. Further studies on control of such a two stage combustion system should be made by simplifying the system model to proportions suitable for on-line control.

APPENDIX 1.SPECIFICATION OF MINISTRY OF DEFENCE (NAVY)REFERENCE FUEL OIL 'VANILLA' (73).

<u>Fuel Composition</u>	% by weight
Carbon	86.1
Hydrogen	11.0
Sulphur	2.4
Ash	0.083
Uncombined Water	0.21
Unaccounted (probably nitrogen)	0.207

Ash Composition as determined by ashing in an oxygen atmosphere, shown in Table A1-1.

TABLE A1-1. RESIDUAL FUEL OIL ASH COMPOSITION.

OXIDE	% in ASH	ppm in FUEL	ELEMENT	% in ASH
V_2O_5	78.4	651	V	43.9
Na_2O	7.0	58	Na	5.3
NiO	7.2	60	Ni	5.7
SO_3	7.2	-	S	2.9
Fe_2O_3	1.7	14	Fe	1.2
Al_2O_3	1.25	10	Al	0.7
MgO	1.20	9	Mg	0.72
SiO_2	0.9	7	Si	0.42
CaO	0.85	7	Ca	0.61
ZnO	0.2	2	Zn	0.16

Trace amounts (less than 1 ppm in the fuel) of the following elements were present: Ba, Cu, Mn, Pb, Sn, Ti, Cr, Co.

The analysis exceeds 100% of the ash as some V_2O_5 (approx. 8%) dissociated to V_2O_4 but is reported as V_2O_5 .

Physical Properties.Relative Density.

at 289 K (15.5 C) = 0.9635

Gross Calorific Value.

42.9 MJ/kg (18400 BTU/lb)

Heat of Formation.

1.54 MJ/kg (664 BTU/lb)

Viscosity

	kinematic	absolute
at 294 K	$4.5 \times 10^{-4} \text{ m}^2/\text{s}$	0.432 Ns/m ²
at 311 K	$1.5 \times 10^{-4} \text{ m}^2/\text{s}$	0.145 Ns/m ²
at 373 K	$1.3 \times 10^{-5} \text{ m}^2/\text{s}$	0.012 Ns/m ²

Flash Point.

368 K

Distillation.

IBP	450 K
10% recovery	530 K
50% recovery	619 K

Vapour Pressure.

at 373 K	0.47 kN/m ²	(0.36 cm Hg)
at 673 K	322 kN/m ²	(245 cm Hg)

Latent Heat of Vaporisation. (64)

207.6 kJ/kg

Surface Tension. (64)

.020 J/m² (N/m)

Heat Capacity (liquid). (64)

3.64 kJ/kg K

APPENDIX 2.

DESIGN CALCULATIONS AND OPERATING PROCEDURE FOR
THE SPRAY COOLER PROBE.

Water Spray Cooling Requirement.

The following design calculations are based on operation with a fuel flow rate of 0.003 kg/s and a steam/fuel mass ratio of 0.9. It is assumed that spray cooling of the gas stream to not less than 1073 K (800 C) will cause blowout of the flame in the second stage. It is further assumed for the purpose of design calculations only, that the cooling tube does not cool the gas stream. Calculations were made for three levels of first stage equivalence ratio, $\phi_1 = 1.2, 1.7, 2.3$, and the results are tabulated in Table A2-1.

TABLE A2-1. SPRAY COOLING REQUIREMENTS.

ϕ_1	1.2	1.7	2.3
$\dot{T}$ K	2038	1770	1538
$\dot{M}$ kmole/s	.00150	.00117	.00096
x_{CH_4}	.004	.021	.043
x_{CO}	.040	.083	.107
x_{CO_2}	.104	.083	.078
x_{H_2}	.015	.047	.079
x_{H_2O}	.195	.188	.172
x_{N_2}	.642	.577	.520
x_{O_2}	-	-	-
$\dot{Q}$ kW	55.1	30.9	16.9
$\dot{m}_w$ kg/s	.0131	.0074	.0040

The heat removal required to reduce the gas stream temperature to 1073 K is

$$\dot{Q} = \dot{M} \int_{T}^{1073} C_{pm}(T) dT \quad \text{kW} \quad \text{A2.1}$$

where the mean molar heat capacity C_{pm} is

$$C_{pm} = \sum_{i=1}^7 x_i C_{pi}(T) \quad A2.2$$

The heat required to raise 1 kg of water from 293 K to 1073 K is

$$\begin{aligned} Q_w &= \int_{293}^{373} C_{pL}(T) dT + \lambda_w + \int_{373}^{1073} C_{pv}(T) dT \quad A2.3 \\ &= 4.20 \text{ kJ/kg} \end{aligned}$$

where C_{pL} is liquid heat capacity and C_{pv} is vapour heat capacity.

The mass flow rate of spray water is $\dot{m}_w$

$$\dot{m}_w = \frac{Q}{Q_w} \quad A2.4$$

Nozzle Specification.

The configuration of the furnace and low pressure (13.8 kN/m² gauge) utility water limited the range of suitable nozzles to those of the narrow spray angle low pressure type. The type of drop size distribution from a hollow cone spray nozzle appeared to be satisfactory.

A hollow cone spray nozzle with a throughput of .0226kg/s at a pressure drop across the nozzle of 13.8 kN/m² was used. The included spray angle was 40°, however it was found that this was reduced to 25° when the probe was placed in the furnace. The drop size distribution from this nozzle was not known; however, the initial D_{sm} for a spray from a nozzle of a similar type was known to be 280 μm , with a corresponding mean evaporative diameter* D_{med} of 250 μm .

* The mean evaporative diameter of a drop that evaporates at the same rate per unit volume as the bulk of the spray.

Length of Evaporation Section.

In order to calculate the cooling effects of the spray water, the spray must be completely evaporated before entering the second stage chamber. The length of the required evaporation section can be calculated from spray evaporation theory.

By mass balance on the evaporation section, of cross sectional area A_y

$$\dot{m} = v A_y \left(\rho_g + \frac{\rho_v}{m_r \epsilon_e} \right) \quad A2.5$$

where $\dot{m}$ is the mass flow rate of the combustion gases excluding the spray water, v is the gas velocity in the evaporation section, ϵ_e is the fractional evaporation efficiency of the spray, and m_r is the mass ratio of the spray water to the combustion gas

By enthalpy balance,

$$\dot{m} C_{pg} (T - T_{\epsilon}) = \dot{m} m_r \epsilon_e (\lambda_w + C_{pv}(T_{\epsilon} - 373)) \quad A2.6$$

where T_{ϵ} is the gas temperature at the corresponding fractional evaporation efficiency ϵ_e . Both combustion gas density ρ_g and water vapour density ρ_v are related to the evaporation efficiency viz.

$$\rho_g = 1.282 \left(\frac{273}{T_{\epsilon}} \right) \quad \text{kg/m}^3 \quad A2.7$$

$$\rho_v = 0.804 \left(\frac{273}{T_{\epsilon}} \right) \quad \text{kg/m}^3 \quad A2.8$$

The evaporation rate of a relatively stationary monosize spray of diameter D_{med} is

$$\dot{m}_w = \frac{2 \pi k \ln (1 + B_m)}{C_{pv}} D_{med} \quad A2.9$$

where k is the thermal conductivity of the gas and B_m is the

mean transfer number over the evaporation section. The equation applies where D_{med} is constant, which has been determined from experiment to be for $0 < \epsilon_e \leq 0.75$ (74).

In terms of fractional evaporation efficiency, A2.9 becomes

$$\frac{d\epsilon_e}{dt} = \frac{12 k \ln(1 + B_m) (1 - \epsilon_e)}{C_{pv} \rho_L D_{\text{med}}^2} \quad \text{A2.10}$$

The required evaporation section length L is related to fractional evaporation and gas velocity viz.

$$\frac{d\epsilon_e}{dt} = \frac{d\epsilon_e}{dL} \frac{dL}{dt} = v \frac{d\epsilon_e}{dL} \quad \text{A2.11}$$

or, for small increments

$$\Delta L = v \frac{dt}{d\epsilon_e} \Delta \epsilon_e \quad \text{A2.12}$$

By the use of equations A2.5, A2.10 and A2.12, the length of the evaporation section required for $\epsilon_e = 0.75$ is calculated as follows for operation at $\phi_1 = 1.2$, $F_2/F_1 = 0.9$, $F_1 = 3.00 \text{ g/s}$

T	$=$	2033 K	
T_1	$=$	1073 K	
$T_{0.75}$	$=$	1315 K	
C_{pg}	$=$	1.25 kJ/kg K	(1315 - 373 K)
C_{pv}	$=$	2.39 kJ/kg K	(2038 - 373 K)
C_{pv}	$=$	2.21 kJ/kg K	(373 - 1315K)
k	$=$	8.37 J/m s K	
λ_w	$=$	2260 kJ/kg	
ρ_L	$=$	1000 kg/m ³	
D_{med}	$=$	250 μm	
$\dot{m}_w$	$=$	.0131 kg/s	
$\dot{m}$	$=$	.0375 kg/s	
m_r	$=$	$\dot{m}_w/\dot{m} = 0.35$	
A_y	$=$	.0053 m ²	

Calculation of log mean transfer number B_m

$$B_{0.} = 2.39 (2038 - 373) / 2260 = 1.76$$

$$B_{0.75} = 2.21 (1315 - 373) / 2260 = 0.92$$

$$B_m = \frac{1.76 - 0.92}{\ln \left(\frac{1.76}{0.92} \right)} = 1.29$$

From equations A2-5, A2.7 and A2.8,

$$v = 0.084 T_{\epsilon} (0.22 + \epsilon_e)$$

From equation A2.6,

$$T_{\epsilon} = \frac{2545 - 491 \epsilon_e}{1.25 - .805 \epsilon_e}$$

From equation A2.10,

$$\frac{d\epsilon_e}{dt} = \frac{37.7 \times 10^{-6} (1 - \epsilon_e)}{D_{med}^2}$$

Using equation A2.12, the following results are obtained

$$\Delta L = 1.36 \times 10^6 D_{med}^2 \frac{(.22 + \epsilon_e)(5.18 - \epsilon_e) \Delta \epsilon_e}{(1.55 + \epsilon_e)(1 - \epsilon_e)}$$

The results are shown in Table A2-2 for ϵ_e up to 0.90 although this method is valid only for $\epsilon_e \leq 0.75$.

TABLE A2-2. LENGTH OF COOLING SECTION FOR SPRAY COOLING.

ϵ_e	$\Delta \epsilon_e$	ϵ_{em}	ΔL (m)	L (m)
0.	0.15	.		0.
0.15	0.15	0.075	.0128	.0128
0.30	0.15	0.225	.0206	.0334
0.45	0.15	0.375	.0305	.0639
0.60	0.15	0.525	.0450	.1089
0.75	0.15	0.675	.0712	.1801
0.90	0.15	0.825	.1400	.3201

It is seen that 75% of a spray having $D_{med} = 250\mu m$ will evaporate in .18 m of the cooling section. As ϵ_e approaches 1, it can be seen from the previous equation for ΔL that ΔL approaches infinity, hence the required evaporation length will be over-estimated. By extrapolation to $L = 1.22$ m, it is found that $\epsilon_e > 0.95$, hence the error in the assumption that the spray is completely evaporated is less than 5%. Calculations for the largest expected drop ($\approx 340 \mu m$) using the previous equations indicate that it is only 65% evaporated at the outlet of the cooling section, however the fractional mass of the spray represented by these large drops is small, and the errors introduced will not be significant.

Operating Procedure.

The spray probe consists of a water cooled arm fitted with a hollow cone spray nozzle, and mounted on a face plate which allows it to be withdrawn from the flame zone. This arrangement is shown in Fig.3-6. The probe is designed so that the flow through the nozzle can be controlled by an 'on-off' type plunger that seats on the nozzle inlet. This arrangement allows the spray to be started and stopped abruptly without interrupting the cooling water flow through the probe head. The nominal included spray angle is 40° , but this is reduced to $20-25^\circ$ by high gas flow rates. Cold tests, using first stage air only, indicated that the spray did not impinge on either the refractory lining or the cooling tube walls.

A measurement of the blowout is obtained by varying the spray flow rate using valves V_1 and V_2 (Fig.3-6) and

observing the second stage flame through the observation port. The procedure is as follows.

The nozzle is closed and cooling water circulated through the probe at the maximum flow (.040 kg/s approx.) by opening V_1 and V_2 . The probe is inserted into the flame zone, and nozzle aligned with the downstream axis of the chamber. The nozzle is opened and the spray water flow rate increased by gradually closing V_2 until the flame is extinguished, i.e. no visible flame in the chamber. V_2 should never be closed completely as this will interrupt the flow of cooling water. Spray flow is continued until the walls of the combustion chamber have visibly cooled, after which the nozzle is closed. If the flame does not reignite immediately, withdrawal of the probe into the collar will aid reignition.

As soon as the flame ignites, the probe is re-inserted and the nozzle opened. If the flame extinguishes immediately, the nozzle is closed to allow the flame to re-ignite. The downstream valve V_2 is then opened slightly before the nozzle is opened again, and the procedure repeated. Thus a decreasing water spray flow rate will increase the inlet temperature to the second stage until at one setting, ignition is maintained. The corresponding chamber temperature is the ignition temperature, and the temperature at which the flame extinguishes is the blowout temperature. This procedure is repeated until consistent results are obtained. The flow rate of water into the gas stream is determined by comparing the readings on rotameters R_1 and R_2 .

It was found that a simpler technique gave the same results. This method involved the same initial procedure for cooling the chamber walls. After the flame has re-ignited, V_2 is opened and the probe re-inserted. The nozzle is opened, and the flow rate gradually increased by closing V_2 slowly. If blowout cannot be achieved by the use of V_2 , V_1 may be opened fully. After blowout occurs, the nozzle is closed and the test repeated until reproducible results are obtained. The method has also been used to determine blowout temperatures under hot wall conditions, i.e. when no cooling of the walls preceded the measurements.

At blowout conditions, some heat is removed by the cooling tube, hence water flow rate, and inlet and outlet temperatures are required. The measured cooling water outlet temperature is not indicative of the heat removal from the combustion gas stream at blowout, as the cooling tube has not reached thermal equilibrium. It is necessary, therefore, to continue the spray water flow corresponding to the blowout condition until the cooling tube reaches steady state. Representative tests indicated that in all cases the steady state cooling water outlet temperature was 9-11 K below that temperature measured at the moment of blowout.

APPENDIX 3.INTEGRATION OF DROP SIZE NUMBER DISTRIBUTION FUNCTION
AND CALCULATION OF D_{sm} , N_c , N_s and n .

Basis: 1 kg fuel.

Substitution of equation 2.1 into equation 4.33
where $m = n - 1$ yields

$$\int_0^{\infty} k_1 D^{n-1} \exp(-k_2 D^n) dD = 1 \quad A3.1$$

$$\text{Let} \quad k_2 = k_2^* n \quad A3.2$$

$$\text{and} \quad k_2^* D = t^{1/n} \quad A3.3$$

$$\text{Therefore} \quad D = \frac{t^{1/n}}{k_2^*} \quad A3.4$$

$$\text{and} \quad dD = \frac{1}{nk_2^*} t^{1/n-1} dt \quad A3.5$$

Substitution into A3.1 yields

$$k_1 \int_0^{\infty} \frac{t^{1-1/n}}{k_2^{*n-1}} \exp(-t) \frac{t^{1/n-1}}{nk_2^*} dt = 1 \quad A3.6$$

or in integrated form

$$\frac{k_1}{n k_2^{*n}} = 1 \quad A3.7$$

$$\text{hence} \quad k_1 = nk_2^n \quad A3.8$$

Surface mean diameter - Char.

Substitution of 2.1 into 2.3 for $m = n - 1$, leads to

$$D_{sm} = \frac{\int_0^{\infty} k_1 D^3 D^{n-1} \exp(-k_2 D^n) dD}{\int_0^{\infty} k_1 D^2 D^{n-1} \exp(-k_2 D^n) dD} \quad A3.9$$

Considering the numerator and denominator individually, and making the same substitutions as in A3.2 to A3.5,

$$\begin{aligned}
 \text{Numerator} &= \int_0^{\infty} \frac{k_1}{k_2^{*n+2}} t^{1+2/n} \exp(-t) t^{1/n} - 1 dt \\
 &= \int_0^{\infty} \frac{k_1}{n k_2^{*n+3}} t^{3/n} \exp(-t) dt \quad \text{A3.10}
 \end{aligned}$$

This can be expressed as a Gamma function

$$\text{Numerator} = \frac{k_1}{n k_2^{1+3/n}} \Gamma\left(\frac{3}{n}+1\right) = \frac{3k_1}{n^2 k_2^{1+3/n}} \Gamma\left(\frac{3}{n}\right) \quad \text{A3.11}$$

A similar treatment of the denominator yields

$$\text{Denominator} = \frac{2k_1}{n^2 k_2^{1+2/n}} \Gamma\left(\frac{2}{n}\right) \quad \text{A3.12}$$

hence

$$D_{sm} = \frac{\text{Numerator}}{\text{Denominator}} = \frac{1.5 \Gamma(3/n)}{k_2^{1/n} \Gamma(2/n)} \quad \text{A3.13}$$

Surface Mean Diameter - Soot.

A similar procedure based on equation 4.43 is used to determine the surface mean diameter of the soot particles.

Number of Particles - Char.

The number of particles N_t per kilogram of fuel is

$$N_t = \frac{1 - \epsilon_v}{\rho_L \frac{\pi}{6} \int_0^{\infty} G_n D^3 dD} \quad \text{A3.14}$$

Integrating as before

$$N_t = \frac{2(1 - \epsilon_v) n^2 k_2^{(1+3/n)}}{\rho_L \pi k_1 \Gamma(3/n)} = N_c \quad \text{A3.15}$$

Number of Particles - Soot.

The number of soot particles N_s per kilogram of fuel is determined from the knowledge of the mass of soot m_s

$$N_s = \frac{m_s}{\rho_s \frac{\pi}{6} \int_0^{\infty} G_s D^3 dD} \quad \text{A3.16}$$

Integration is achieved as before.

Determination of the Constant n from Evaporation Theory.

If the constant n can be determined from evaporation theory, the constants k_1 and k_2 in equation A3.1 can be found using equations A3.8 and A3.12.

The method of evaluating n requires the use of equation 2.12 viz.

$$\int_0^D \frac{D \, dD}{2 + k_3 N_{Re}^{0.5}} = k_4 D^n \quad A3.17$$

Equation A3.17 can be rewritten explicitly in D

$$\int_0^D \frac{D \, dD}{2 + k_3' D^{0.5}} = k_4 D^n \quad A3.18$$

where

$$k_3' = \frac{k_3 v}{v} \quad A3.19$$

Values of k_3 for various burning fuel drops have been determined by EISENKLAM et al (75). It can be seen that k_3' is dependent on particle/gas relative velocity. The dependence of velocity on drop diameter can be determined from equation 2.16, however if k_3' is constant for small changes in D, equation A3.18 can be integrated to yield

$$k_4 D^n = -\frac{2D}{k_3'^2} + \frac{2D^{3/2}}{3k_3'} + \frac{8D^{1/3}}{k_3'^3} - \frac{16}{k_3'^4} \ln \left(\frac{D^{1/2} + 2/k_3'}{2/k_3'} \right) \quad A3.20$$

By substituting values of D into the right hand side of A3.20 for constant k_3' , values of $k_4 D^n$ are found. If the ln (right hand side) is plotted against ln(D), straight lines of slope n and intercept ln(k_4) result, hence n can be determined.

APPENDIX 4.TABULATED RESULTS.

- Table A4-1. Comparison of measured and predicted first stage solids.
- A4-2. Comparison of measured and predicted second stage solids.
- A4-3. Experimental blowout settings.
- A4-4. Interstage heat removal at blowout.
- A4-5. Calculated blowout conditions.

TABLE A4-1. COMPARISON OF MEASURED AND PREDICTED
FIRST STAGE SOLIDS.

EXPERIMENT NUMBER ++	TOTAL MEASURED (kg/kg fuel)	CHAR (model) (kg/kg fuel)	SOOT. (eqn.5.1) (kg/kg fuel)	% DIFFERENCE
H 2	.0101	.0042	.0102	+ 42.5
H 4	.0216	.0124	.0077	- 6.9
H 5	.0278	.0225	.0071	- 6.5
H10	.0323	.0305	.0078	0.0
H13	.1826	.1087	.0720	- 1.0
H14	.0449	.0384	.0095	+ 6.7
H15	.0379	.0305	.0082	+ 2.1
H16	.0141	.0063	.0093	+ 10.6
H17	.0264	.0185	.0073	- 2.3
H18	.0235	.0145	.0073	- 7.2
H19	.0661	.0500	.0158	- 0.5
H21	.0290	.0205	.0068	- 5.8
H22	.0154	.0063	.0087	- 2.6
H25	.0187	.0083	.0122	+ 9.6
H26	.0215	.0063	.0112	- 18.5
H27	.0184	.0083	.0131	+ 16.3
* H28	.0076	.0205	.0147	+ 364.0*
* H29	.0209	.0225	.0163	+ 85.9*
* H30	.0111	.0205	.0124	+ 196.0*
H31	.0190	.0042	.0115	- 17.4
H32	.0129	.0022	.0111	+ 3.1
H33	.0120	.0011	.0104	- 4.2
H34	.0129	.0022	.0105	- 1.5
H35	.0109	.0011	.0104	+ 5.5
H36	.0175	.0011	.0171	+ 4.0
H37	.0165	.0011	.0173	+ 11.5
H38	.0232	.0205	.0105	+ 33.5
H39	.0230	.0205	.0104	+ 34.4
H40	.0144	.0083	.0107	+ 33.4
H41	.0136	.0022	.0109	- 3.7
H42	.0165	.0042	.0122	- 0.6
H43	.0200	.0042	.0112	- 23.0

* Experimental results are suspect.

++ Reference (2) for experimental conditions.

TABLE A4-2. COMPARISON OF MEASURED AND PREDICTED
SECOND STAGE SOLIDS.

EXPERIMENT NUMBER *	TOTAL MEASURED**	TOTAL PREDICTED (kg/kg fuel)	% DIFFERENCE
7	.0079	.0134	+ 69.7
9	.0105	.0108	+ 2.8
10	.0036	.0058	- 32.5
11	.0105	.0060	- 42.8
12	.0128	.0105	- 18.0
13	.0140	.0116	- 17.1
14	.0039	.0069	+ 76.9
15	.0041	.0087	+ 112.0
17	.0124	.0200	+ 61.3
18	.0136	.0316	+ 132.0
20	.0089	.0105	+ 18.0
21	.0090	.0176	+ 95.5
23	.0070	.0185	+ 164.0
24	.0058	.0022	- 62.1
25	.0066	.0094	+ 42.4
26	.0064	.0057	- 10.9
27	.0047	.0064	+ 36.2
28	.0028	.0092	+ 229.0
29	.0026	.0010	- 61.5
30	.0099	.0043	- 56.6
32	.0050	.0048	- 4.0
33	.0015	.0018	+ 20.0
34	.0030	.0050	+ 66.7
37	.0013	.0010	- 23.1
46	.0013	.0014	+ 7.7
47	.0019	.0164	+ 762.0
51	.0126	.0010	- 92.1
53	.0017	.0010	- 41.1

* Reference (3) for experimental conditions

** includes estimated 0.0007 kg solid/kg fuel lost in stack gas.

TABLE A4-3. EXPERIMENTAL BLOWOUT SETTINGS.

EXPERIMENT NUMBER	ϕ_1	ϕ_2	FUEL FLOW RATE (g/s)	SPRAY WATER FLOW RATE (g/s)	COOLING WATER	
					FLOW RATE (g/s)	ΔT K
20	1.82	1.03	3.500	16.7	189.6	12.0
21	1.73	1.01	3.639	19.7	192.1	7.0
23	2.03	1.14	3.773	17.1	176.9	9.5
24	1.43	0.98	3.000	16.9	180.7	12.0
28	1.74	1.08	3.250	13.9	183.3	10.0
29	1.71	0.99	3.056	16.5	179.5	9.0
30	1.63	1.01	3.194	15.6	183.3	10.0
31	1.53	0.95	3.194	16.0	183.3	7.5
33	1.86	1.04	3.306	14.3	182.0	9.0
34	1.93	1.13	3.250	11.1	164.3	9.0
36	2.00	0.96	3.056	11.9	171.9	7.0
37	1.14	0.81	2.500	12.7	176.9	8.5
40	1.29	0.93	2.778	14.0	180.7	20.0
41	1.28	1.00	3.000	16.9	183.3	20.0
43	1.45	0.86	2.403	11.7	185.8	11.0
44	1.63	1.02	2.708	14.8	185.8	11.0
47	2.00	1.21	3.750	17.5	176.9	12.0
48	2.00	0.99	3.139	12.6	176.9	6.5
49	2.16	1.03	3.389	12.5	176.9	6.5
51	1.85	0.93	3.014	16.4	189.6	12.5
52	1.34	0.94	2.750	15.4	188.3	13.0
53	1.42	0.95	2.750	15.6	198.5	14.0
54	2.20	0.91	3.194	13.0	183.3	8.3
55	1.41	0.96	2.500	16.2	185.0	10.0
57	1.85	0.57	3.000	16.8	228.0	11.0

TABLE A4-4. INTERSTAGE HEAT REMOVAL AT BLOWOUT.

EXPERIMENT NUMBER	HEAT REMOVAL				
	SPRAY (kW)	COOLING TUBE (kW)	AIR RETURN (kW)	DIRECT TOTAL (kJ/kg fuel)	INDIRECT TOTAL (kJ/kg fuel)
20	48.11	9.53	.06	16485	19559
21	56.98	5.63	.23	17271	20199
23	47.51	7.04	.02	14445	16969
24	52.58	9.08	.16	20605	25642
28	43.83	7.88	.23	15984	19754
29	48.16	6.76	.18	17033	21299
30	48.78	7.67	.24	17749	21924
31	52.67	5.60	.22	18309	22079
33	43.69	6.86	.06	15312	18315
34	36.46	6.19	.10	13153	16778
36	36.64	5.04	.28	13533	18363
37	45.83	6.30	.16	24915	23510
40	47.49	15.13	.25	22636	25227
41	55.35	15.35	.28	23653	26455
43	38.81	8.53	.22	19797	23521
44	43.40	8.53	.25	19265	22701
47	48.48	8.89	.22	15357	17924
48	38.65	4.82	.15	13896	16742
49	37.74	4.82	.21	12621	16107
51	43.97	9.90	.13	17950	19858
52	50.96	10.22	.11	22309	25357
53	48.63	11.82	.40	22153	26315
54	36.36	6.37	.22	13411	15420
55	46.59	7.78	.51	21951	25730
57	43.73	10.49	.14	18122	20113

TABLE A4-5. CALCULATED BLOWOUT CONDITIONS.

EXPT.	INDIRECT + DIRECT COOLING			INDIRECT COOLING		
	INLET TEMP. K	% OF STAGE 1 HEAT RELEASE	F _{PM}	INLET TEMP. K	% OF STAGE 1 HEAT RELEASE	F _{PM}
20	515	77.5	.131	373	91.9	.125
21	531	76.6	.242	373	89.9	.158
23	465	79.4	.148	373	93.0	.100
24	625	73.3	.346	373	91.1	.255
28	648	70.2	.110	373	86.9	.076
29	531	72.6	.236	373	90.6	.163
30	639	71.4	.187	373	88.5	.128
31	720	68.0	.165	449	82.2	.140
33	592	72.4	.110	373	86.6	.078
34	703	65.5	.055	460	83.5	.041
36	606	71.2	.083	373	95.3	.062
37	859	64.1	.275	930	60.5	.163
40	758	67.7	.275	602	75.3	.190
41	703	70.6	.522	528	78.9	.341
43	721	68.4	.165	473	81.3	.116
44	531	77.4	.330	373	91.2	.218
47	443	81.2	.258	373	94.7	.152
48	592	72.3	.083	373	87.1	.061
49	573	73.1	.066	373	93.5	.049
51	391	84.4	.357	373	93.4	.238
52	711	69.6	.357	515	79.2	.237
53	617	74.5	.439	373	90.0	.290
54	471	79.0	.121	373	90.8	.090
55	504	79.4	.605	373	93.4	.384
57	372	85.5	.412	373	95.0	.281

APPENDIX 5.

ANALYSIS OF ERROR.

There are three sources of error that must be considered.

1. Random errors arising from uncertainty in experimental settings.
2. Bias errors arising from inaccuracy in experimental technique.
3. Prediction errors arising from inadequacy of theoretical models.

Random errors in experimental measurements arising by chance are not considered.

The random errors can be minimised by making repeated measurements at each experimental setting. These errors can be expressed as the standard deviation about the mean. The standard deviation and 95% confidence limits for each experimental setting are shown in Table A5-1. For variables such as ϕ_1 which are not measured directly, the coefficient of variation is shown as a percent of the mean.

The method of calculation of these errors is illustrated for the spray water flow rate. The net spray water flow rate was determined by difference from flow measurements on the inlet and outlet streams, as well as by collection and weighing of the spray water. These measurements were made over ten, one minute intervals for constant experimental settings. These measured values are given in Table A5-2 for nominal inlet and outlet flow rates of 30 g/s and 15 g/s respectively. At the 95% confidence

TABLE A5-1. ACCURACY OF EXPERIMENTAL SETTINGS.

	MEAN	VARIANCE	STANDARD DEVIATION	DEGREES OF FREEDOM	CONFIDENCE LIMIT(95%)
Fuel flow rate(g/s)	6.03	.000605	$\pm .0246$	11	$\pm .054$
Steam flow rate(g/s)	2.79	.000097	$\pm .0098$	5	$\pm .025$
Forward air flow rate(g/s)	14.8	.0098	$\pm .100$	5	$\pm .257$
Reverse air flow rate(g/s)	14.3	.0087	$\pm .093$	5	$\pm .239$
Interstage cooling water flow rate(g/s)	176.6	18.66	± 4.32	5	± 11.1
Stage 2 inlet air flow rate (g/s)	28.1	.1317	$\pm .362$	5	$\pm .930$
Stage 2 return air flow rate(g/s)	11.8	.0018	$\pm .042$	5	$\pm .108$
Inlet spray water flow rate(g/s)	29.6	.0765	$\pm .277$	9	$\pm .63$
Outlet spray water flow rate(g/s)	14.5	.0693	$\pm .263$	9	$\pm .60$
				COEFFICIENT OF VARIATION	
ϕ_1	2.90	.00053	$\pm .023$	0.79%	
F_2/F_1	.463	.0012	$\pm .011$	2.38%	
F_4/F_3	.967	.00009	$\pm .010$	1.03%	
ϕ_2	1.86	.00032	$\pm .018$	0.97%	
Spray water (g/s)	15.1	.1458	$\pm .382$	2.53%	

TABLE A5-2

SPRAY WATER FLOW RATES.

FLOW RATES FROM ROTAMETER READINGS			NET FLOW RATE (GRAVIMETRIC) g/s
INLET g/s	OUTLET g/s	NET g/s	
29.5	14.4	15.1	15.2
29.9	14.4	15.5	15.4
29.5	14.2	15.3	15.0
29.3	14.0	15.3	15.1
29.2	14.2	15.0	14.8
29.5	14.6	14.9	15.0
30.1	14.8	15.3	14.9
29.9	14.5	15.4	14.9
29.9	14.5	15.1	14.7
29.3	14.6	14.7	14.8
mean= 29.61	mean=14.45	mean=15.16	mean=14.98

TABLE A5-3. ERROR IN SECOND STAGE TEMPERATURE
DUE TO UNCERTAINTY IN EXPERIMENTAL
SETTINGS.

UNCERTAINTY IN ϕ_2 ONLY	NOMINAL ϕ_2		
	0.95	1.00	1.05
+ 0.03	+ 60K	- 30K	+ 30K
- 0.03	- 70K	- 60K	- 30K

level, the mean water spray flow rate calculated from rotameter readings is 15.16 ± 0.86 g/s and thus agrees with the true mean of 14.98 g/s determined gravimetrically. The coefficient of variation in this case is $\pm 2.84\%$.

The effect of these errors on system performance has been determined by perturbing each of the variables by its standard deviation and analysing the response using the system models. These effects as well as errors arising from experimental technique and model inadequacy are discussed in the following sections. In general, the most significant sources of error are random errors and bias in the experimental measurements. Errors arising from model inadequacy are less important.

Gas Composition.

Differences between measured and predicted gas compositions shown in Section 5-1. can be explained by errors in the reported experimental measurements. Analysis of the effects of uncertainty in experimental settings has indicated that the maximum absolute fluctuation in the mole fraction of any component is ± 0.004 . This fluctuation occurs in second stage outlet compositions when $\phi_2 = 1.00 \pm 0.02$. At all other conditions, the error is smaller. Errors arising from inadequacies in the model are negligible for all conditions where the equivalence ratio is greater than 0.9. At lower values, the heat balance iteration may fail to converge, resulting in large errors in predicted gas compositions.

The only significant errors in gas composition arise from the measurement technique. All gas compositions

reported by HTUN (2) indicate more carbon in the gas phase as CO_2 and CO than is present in the fuel. As these results have been recorded on a dry basis, the source of the error cannot be positively determined. The likely cause is the improper drying of the gas sample as noted in Section 7.2. Similar errors occur with respect to hydrogen in some of the gas compositions reported by ARCHER (3), particularly those results for the gasifier. In all gasifier outlet analyses, the hydrogen as H_2 , H_2O and CH_4 exceeds that in the fuel by about 25%.

The random experimental errors in the results determined by gas chromatography have not been determined as duplicate analyses for the same conditions were not reported. It is known that the composition of the calibration mixture is accurate to $\pm 5\%$ of the stated mole fraction of any component, hence the reported results may be in error by at least this much.

The measured gas compositions from the second stage do not indicate mass balance errors. These measured compositions correspond with those predicted on the basis of water gas shift equilibrium. Where $\phi_2 < 1.0$, no oxygen was detected in the exhaust gases. This indicates that the method of analysis may have been insensitive to oxygen concentrations less than 1%.

As the total hydrogen measured in the gasifier is consistently higher than that fed, the calibration of the chromatography unit for H_2 and H_2O is suspect. Condensation of water from the combustion gas sample would lead to under-estimation of the water vapour, whereas condensation from a water vapour calibration sample consisting of steam, would

lead to an over-estimation of the water vapour present. It is also possible that the chromatograph does not exhibit the same sensitivity to water vapour concentrations approaching 100% (steam sample) as concentrations of 20% in the combustion gas. This may explain the better agreement between experiment and theory when gas compositions are compared on a dry basis.

Gas Temperature.

The accuracy of gas temperatures measured by suction pyrometer is ± 24 K (1). The error arising from uncertainty in experimental settings has been determined by perturbing these settings within the 95% confidence limits for a set of ten experimental runs. The mean temperature fluctuations in the first stage were $+ 24.2$ K and $- 25.2$ K. Uncertainty in the steam/fuel or reverse/forward air ratios resulted in only ± 5 K changes. Uncertainty in the equivalence ratio setting resulted in changes of the order of ± 20 K.

Where ϕ_1 and interstage heat removal are accurate, the uncertainty in second stage temperature depends on the value of ϕ_2 as shown in Table A5-3. It can be seen that the fluctuation in temperature for a given uncertainty in ϕ_2 increases as ϕ_2 decreases.

The second stage temperature predicted from theory depends on interstage heat removal. The inaccuracy in the interstage heat removal arises from uncertainty in both the cooling water flow rate and measured temperature rise of the cooling water (± 1 K). These combined errors may affect the second stage temperature by as much as ± 35 K depending on the absolute value interstage heat removal.

First Stage Solids.

It has been suggested (3) that the first stage solids measured by HTUN (2) may be low by as much as 20% due to sampling error. However, as this work could not be repeated, these results were generally assumed to be without experimental error. Three results obtained at $\phi_1 = 1.62$ and $F_2/F_1 \approx 0.7$ by HTUN (2) were considered suspect as they indicated a threefold reduction in solid formation with a 30 K increase in steam temperature, whilst similar results at different equivalence ratios showed little dependence on steam temperature.

The percentage fluctuations at the maximum absolute deviation in predicted solid loadings corresponding to uncertainties in experimental settings are shown in Table A5-4 for both soot and char. It can be seen that uncertainty in ϕ_1 and steam flow rate F_2 have little effect on the amount of soot or char formed. Soot is affected only by the uncertainty in fuel flow rate which is as expected from Equation 5.1. The maximum error of $\pm .0020$ occurs at $\phi_1 = 2.3$. This error decreases to $\pm .0012$ as ϕ_1 approaches 1.1.

Char formation is unaffected by errors in fuel and steam flow rates as long as the steam/fuel ratio is accurate. When $F_2/F_1 > 0.9$, uncertainties in the true setting result in errors in the amount of char formed of the order ± 0.0005 kg solid/kg fuel. As F_2/F_1 approaches 0.5, the error increases to ± 0.0020 kg solid/kg fuel. Uncertainty in the reverse/forward air ratio has little effect on char formation for $F_2/F_1 > 0.9$. Where $F_2/F_1 < 0.7$, the maximum error associated with uncertainties in F_4/F_3 is ± 0.0030 kg solid/kg fuel.

TABLE A5-4. ERROR IN FIRST STAGE SOLID LOADING DUE
TO UNCERTAINTY IN EXPERIMENTAL SETTINGS.

VARIABLE	UNCERTAINTY RANGE	DEVIATION % OF TOTAL SOLIDS	
		CHAR	SOOT
equivalence ratio ϕ_1	± 0.02	± 1.5	± 1.0
fuel flow rate F_1 g/s	± 0.05	< 1.0	± 3.0
steam flow rate F_2 g/s	± 0.03	< 1.0	< 1.0
steam/fuel mass ratio F_2/F_1	± 0.06	± 3.0	< 1.0
reverse/forward air ratio F_4/F_3	± 0.03	± 3.2	< 1.0
transfer function $f(s)$ eqn. 3.4	± 0.036	± 9.0	< 1.0

Model inadequacy is limited to the error associated with the first stage transfer function $f(s)$ calculated from equation 3.4. An error of $\pm .036$ in $f(s)$ will result in an error of $\pm .0020$ kg solid/kg fuel where fuel evaporation efficiency is 99%, and $\pm .0070$ kg solid/kg fuel at 95% efficiency.

Particle mass distribution is affected by inadequacy of the model and uncertainty in experimental settings. The maximum deviation in mass distribution arising from experimental uncertainty is shown in Fig.A5-1. The nominal settings and applied uncertainties are

$$F_1 = 3.00 \pm 0.05 \text{ g/s}$$

$$F_2 = 2.92 \pm 0.03 \text{ g/s}$$

$$F_3 = 14.89 \pm 0.26 \text{ g/s}$$

$$F_4 = 17.78 \pm 0.24 \text{ g/s}$$

The solid curve indicates the distribution corresponding to $\phi_1 (\text{true}) > \phi_1 (\text{measured})$ and $F_2/F_1 (\text{true}) < F_2/F_1 (\text{measured})$. The broken curves indicate the opposite case.

The main inadequacy in the model is the assumption that mean soot particle diameter is constant at $0.04 \mu\text{m}$ for all operating conditions. This assumption however, does not appear to influence system performance significantly.

Heat Transfer.

Measured heat transfer rates are subject to errors arising from uncertainties in temperature and flow measurement. Uncertainties in cooling water flow rate (± 11.1 g/s) and temperature measurement (± 0.5 K at each reading), result in errors of between $\pm 7.5\%$ and $\pm 11.6\%$ in the measured heat removal.

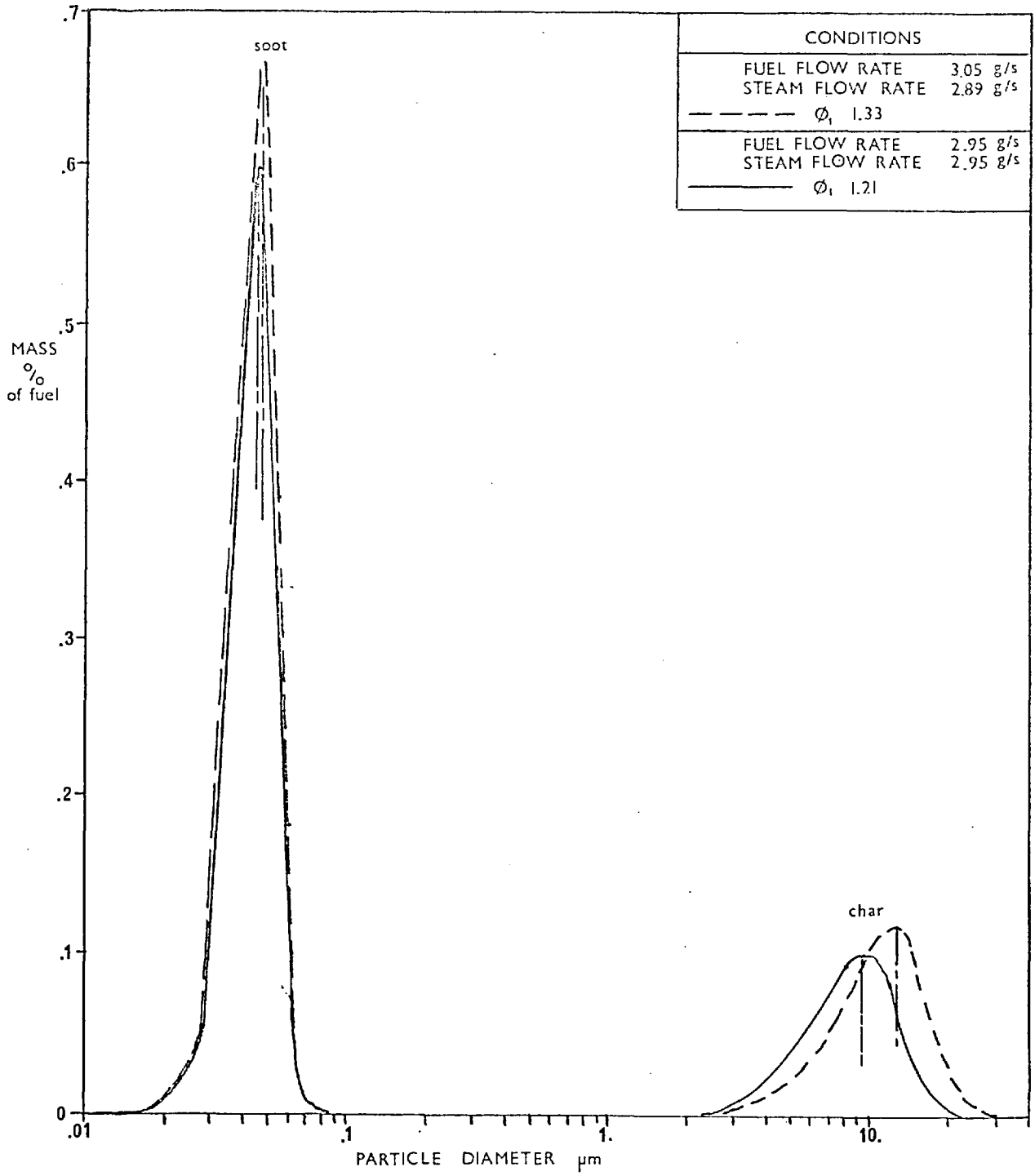


FIG.A5-1 THE EFFECT OF UNCERTAINTY IN EXPERIMENTAL SETTINGS ON PARTICLE MASS DISTRIBUTION FROM THE FIRST STAGE

Uncertainty in first stage settings results in errors of the order of $\pm .0015 \text{ kW/m}^2/\text{K}$ in interstage heat transfer coefficients. This corresponds to a percentage error of $\pm 2\%$.

The consistently high predicted values of heat removal of the order of 70% indicate either experimental measurement error or model inadequacy. The various convective heat transfer expressions yielded similar results, whilst variation of the parameters in the radiant heat transfer expressions did not noticeably increase the heat transfer. The predicted heat transfer rate depends on the fireside wall temperature in the cooling tube.

In practice, the fireside wall temperature in the cooling tube was calculated from a knowledge of the cooling water temperature and the thermal conductivity of the 18-8 stainless steel ($16 \text{ W m/m}^2/\text{K}$). In most cases the wall temperature calculated by this method was 320-330 K. No provision was made for either waterside or fireside fouling, as examination of the equipment indicated that only slight deposition occurred. If it is assumed that a typical fouling coefficient for both sides combined is $6 \text{ kW/m}^2/\text{K}$, the inlet wall temperature increases to 335-345 K, reducing the mean difference between the measured and predicted results from $\pm 70\%$ to $+ 65\%$. No improvement could be made in the scatter of the results by this method.

The most reasonable explanation of this error is that the fireside wall temperature of the cooling tube is considerably higher than 335 K, thus giving a lower temperature driving force and lower predicted heat transfer rates. The presence of high temperature heat sources at each

end of the cooling tube would encourage heat conduction from each end, and cause higher wall temperatures.

Second Stage Solids and Solids Burning.

Measured second stage solids are subject to two possible errors. Most analyses were made gravimetrically assuming a mean filter paper weight. The use of such a mean weight requires the inclusion of the error in the mean; in this case ± 0.406 g. at the 95% confidence level. When the mass of solid collected is small, this uncertainty in filter paper weight can result in large percentage errors in the measured solids.

Collection inefficiency is the other source of experimental error. This inefficiency arises from solids not removed from the combustion gases which were vented to atmosphere, and solids not removed from the collection system during sampling. The solids in the stack gases comprise only 0.07% of the fuel fired, indicating that scrubber efficiency is high (See Appendix 7).

Mass balances on the overall system as described in Appendix 7 have indicated that some sampling was carried out while the system, in particular the cyclone, was not at steady state. At high scrubber water flow rates, when cyclone liquid sampling was begun, the cyclone collection vessel began to fill with liquor. Before sampling, cyclone liquor was pumped to drain through 3 m of 25 mm $\varnothing$ pipe, but during sampling, liquor was pumped through 10 m of 19 mm $\varnothing$ pipe to the sample point. The increase in back pressure resulted in liquor accumulation in the cyclone. As the volume of cyclone liquor collected was not recorded, no definitive estimate of the

error was possible. It was found however, that when scrubber water flow rate exceeded 65 g/s, liquid accumulation was likely to occur, and experimental readings taken at scrubber water flow rates above this value were placed in doubt. The error was eliminated in later tests by taking the sample directly from the cyclone collection vessel and ensuring that the vessel was empty while sampling.

The errors associated with the first and second stage solids are shown in Table A5-5. The range of solid combustion efficiencies is also indicated.

The large errors are due almost exclusively to the error in filter paper weight when small amounts of solids were collected.

The effect of uncertainties in experimental settings on the mass distribution of particles from the second stage is shown in Fig.A5-2. The first stage operating conditions are the same as in Fig.A5-1. The nominal second stage equivalence ratio is 1.00 ± 0.03 . It can be seen that the char distribution is only slightly affected, whereas the soot is completely burned at $\phi_2 = 0.97$, but is only partially burned at $\phi_2 = 1.03$.

Errors in the solid burning model may arise in the treatment of the distribution of particle sizes. For the purpose of calculation, the particle size distribution is considered for a finite number of size increments. The increments were not equally spaced with respect to diameter because of the wide range of sizes encountered. Previously (37,40), the diameter chosen to represent each size increment was the arithmetic mean diameter. This diameter does not represent a particle of mean surface or mean volume

TABLE A5-5. ERRORS ASSOCIATED WITH FIRST AND SECOND STAGE
SOLIDS AND SOLID COMBUSTION EFFICIENCY.

EXPERIMENT NUMBER**	SOLIDS (kg solid/kg fuel)		% BURNED	
	PREDICTED	MEASURED	MEASURED	PREDICTED
	FIRST STAGE ++	SECOND STAGE *		
7	.0181 $\pm$.0061	.0079	34.2 - 67.4	26.0
9	.0284 $\pm$.0061	.0105 $\pm$.0012	47.6 - 73.0	62.0
10	.0261 $\pm$.0061	.0086 $\pm$.0024	45.0 - 80.7	77.8
11	.0247 $\pm$.0061	.0105 $\pm$.0015	35.5 - 71.0	75.3
12	.0247 $\pm$.0061	.0128 $\pm$.0018	21.5 - 64.3	57.5
13	.0273 $\pm$.0061	.0140 $\pm$.0020	24.5 - 64.1	57.5
14	.0351 $\pm$.0061	.0039 $\pm$.0003	35.3 - 91.3	80.4
15	.0390 $\pm$.0061	.0041 $\pm$.0006	35.7 - 92.3	77.7
17	.0281 $\pm$.0061	.0124 $\pm$.0031	29.5 - 72.9	40.5
18	.0389 $\pm$.0061	.0136 $\pm$.0022	51.3 - 74.8	18.7
20	.0328 $\pm$.0061	.0089 $\pm$.0005	65.6 - 78.3	63.0
21	.0417 $\pm$.0061	.0090 $\pm$.0006	73.0 - 32.3	57.8
23	.0466 $\pm$.0061	.0070 $\pm$.0015	79.0 - 89.5	60.3
24	.0183 $\pm$.0061	.0058 $\pm$.0013	44.0 - 82.0	87.9
25	.0326 $\pm$.0061	.0066 $\pm$.0011	70.9 - 85.9	71.2
26	.0280 $\pm$.0061	.0064 $\pm$.0011	65.8 - 84.3	79.7
27	.0264 $\pm$.0061	.0047 $\pm$.0007	73.8 - 87.7	75.8
28	.0278 $\pm$.0061	.0028 $\pm$.0008	33.5 - 94.1	66.9
29	.0209 $\pm$.0061	.0026 $\pm$.0005	79.2 - 92.2	95.3
30	.0229 $\pm$.0061	.0099 $\pm$.0027	53.1 - 75.2	80.4
32	.0293 $\pm$.0061	.0050 $\pm$.0005	76.2 - 87.4	83.7
33	.0258 $\pm$.0061	.0015 $\pm$.0003	90.9 - 96.3	93.3
34	.0262 $\pm$.0061	.0030 $\pm$.0008	81.0 - 93.1	80.3
37	.0103 $\pm$.0061	.0013 $\pm$.0009	47.7 - 97.4	90.2
46	.0179 $\pm$.0061	.0013	89.0 - 94.5	92.1
47	.0377 $\pm$.0061	.0019	93.4 - 95.5	56.5
51	.0216 $\pm$.0061	.0126	13.7 - 54.5	95.4
53	.0135 $\pm$.0061	.0017	77.0 - 91.3	92.6
++ error associated with equation 5.1 ($\pm$.0061 kg solid/kg fuel).				
* error associated with filter paper weight.				
** Reference 3 for experimental conditions.				

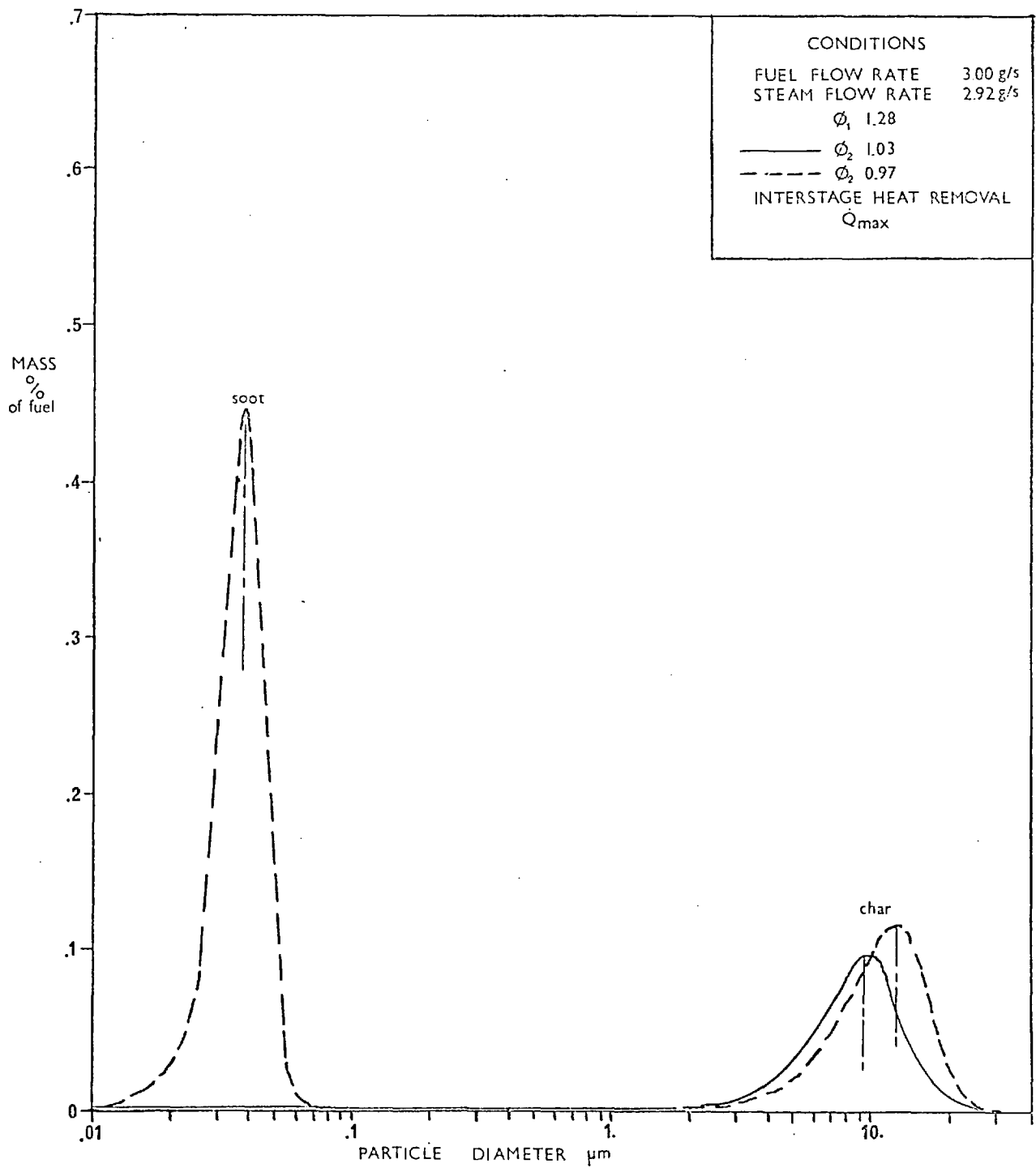


FIG. A5-2 THE EFFECT OF UNCERTAINTY IN EXPERIMENTAL SETTINGS ON PARTICLE MASS DISTRIBUTION FROM THE SECOND STAGE

in the increment, hence reaction rate calculations based on surface area and particle burnout based on volume will be in error. The error which results depends on the increment size as well as the particle diameter and is shown in Fig. A5-3. To reduce this error, the size distribution function G_n is approximated by the linear function

$$G_n = k_D D \quad A5.1$$

where k_D is constant over the increment D_1 to D_2 .

The mean particle volume is then V_{pm}

$$V_{pm} = \frac{\pi}{6} D_{vm}^3 = \frac{\pi}{6} \frac{\int_{D_1}^{D_2} k_D D D^3 dD}{\int_{D_1}^{D_2} k_D D dD} \quad A5.2$$

Equation A5.2 reduces to

$$D_{vm} = \sqrt[3]{0.4 \left(\frac{D_2^5 - D_1^5}{D_2^2 - D_1^2} \right)} \quad A5.3$$

where D_{vm} is volume mean diameter.

Similarly,

$$D_{sm} = \sqrt{0.5 \left(\frac{D_2^4 - D_1^4}{D_2^2 - D_1^2} \right)} \quad A5.4$$

Thermal Stability.

Tests for reproducibility of the experimentally determined blowout points indicated that where $\phi_1 < 1.8$ the results were reproducible to $\pm 3\%$ corresponding to a heat removal of ± 1.3 kW. Above $\phi_1 = 1.8$, the reproducibility was $\pm 5\%$ or ± 2.1 kW.

The main source of error is uncertainty in the spray water and cooling water flow rates. The errors associated with these uncertainties at the 95% confidence

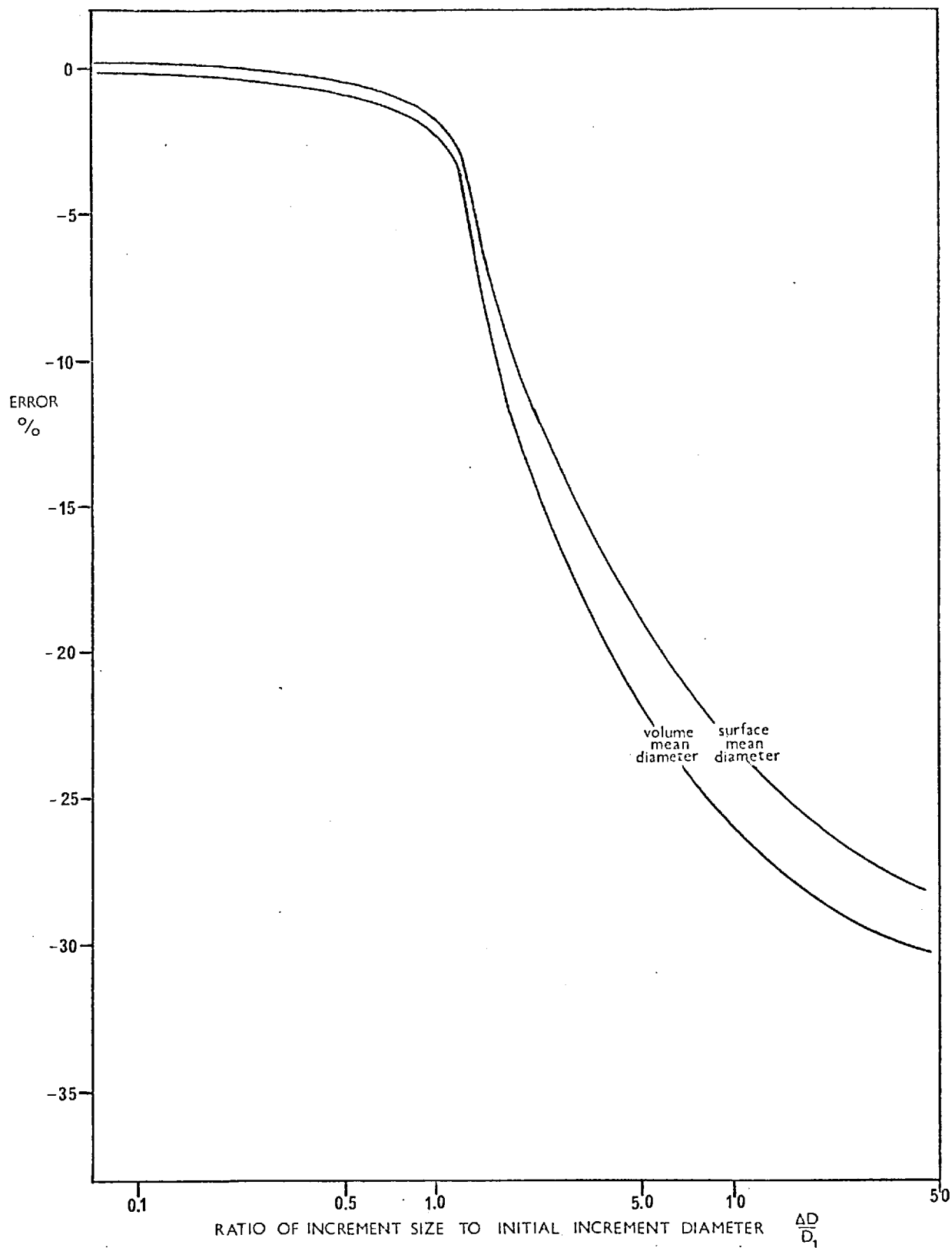


FIG.A5-3 ERROR ASSOCIATED WITH ARITHMETIC MEAN DIAMETER ASSUMPTION

level are tabulated in Table A5-6. The maximum percent error in heat removal by spray water is 9.2% corresponding to 3.84 kW.

The effect of errors on the stability curves is shown in Fig.A5-4. The stability curve corresponds to experiment number 29 where the error in heat removal is $\pm 7.1\%$. The change in gas composition is small, and does not affect the sigmoid curve. The heat balance line AB is shifted by ± 85 K as indicated by the broken lines CD and EF. The slope of the heat balance line increases slightly with positive errors and decreases with negative ones. The effect of this error on the value of F_{pm} (fraction perfectly mixed) is large. For example, an error of $\pm 5\%$ in heat removal corresponding to a ± 50 K shift in the heat balance line, requires a $\pm 50\%$ change in F_{pm} . Hence, large absolute errors occur in F_{pm} as F_{pm} approaches 1. The same errors apply to the stability analysis for indirect heat transfer.

Model inadequacy influences the heat removal predicted for indirect heat transfer conditions. The main error is associated with the choice of F_{pm} at blowout conditions without spray water injection. To preserve system similarity, F_{pm} has been scaled to give the same mean residence time in the perfectly mixed section for both direct and indirect heat removal. Fortunately, the heat removal is relatively insensitive to F_{pm} as mentioned, since a $\pm 50\%$ error in F_{pm} will yield only a $\pm 5\%$ error in the heat removal.

TABLE A5-6. ERRORS ASSOCIATED WITH INTERSTAGE HEAT REMOVAL
AT BLOWOUT CONDITIONS.

EXPERIMENT NUMBER	ERROR IN HEAT REMOVAL RATE			INLET TEMPERATURE ERROR K
	SPRAY (kW)	COOLING TUBE (kW)	% TOTAL	
20	± 2.48	± 1.35	± 6.65	± 82
21	± 2.48	± 1.13	± 5.75	± 71
23	± 2.39	± 1.18	± 6.53	± 77
24	± 2.68	± 1.31	± 6.45	± 82
28	± 2.71	± 1.25	± 7.64	± 83
29	± 2.51	± 1.17	± 7.14	± 86
30	± 2.69	± 1.24	± 6.93	± 80
31	± 2.83	± 1.08	± 6.70	± 75
33	± 2.61	± 1.18	± 7.48	± 82
34	± 2.83	± 1.11	± 9.21	± 87
36	± 2.64	± 1.05	± 8.78	± 89
37	± 3.10	± 1.14	± 6.80	± 80
40	± 2.92	± 1.68	± 7.31	± 88
41	± 2.82	± 1.69	± 6.38	± 81
43	± 2.86	± 1.29	± 8.71	± 97
44	± 2.52	± 1.29	± 7.29	± 90
47	± 2.38	± 1.30	± 6.39	± 77
48	± 2.64	± 1.04	± 8.43	± 88
49	± 2.60	± 1.04	± 8.50	± 86
51	± 2.31	± 1.37	± 6.80	± 87
52	± 2.35	± 1.39	± 6.90	± 93
53	± 2.88	± 1.51	± 7.17	± 93
54	± 2.40	± 1.15	± 8.29	± 89
55	± 2.47	± 1.24	± 6.65	± 87
57	± 2.24	± 1.47	± 6.82	± 91

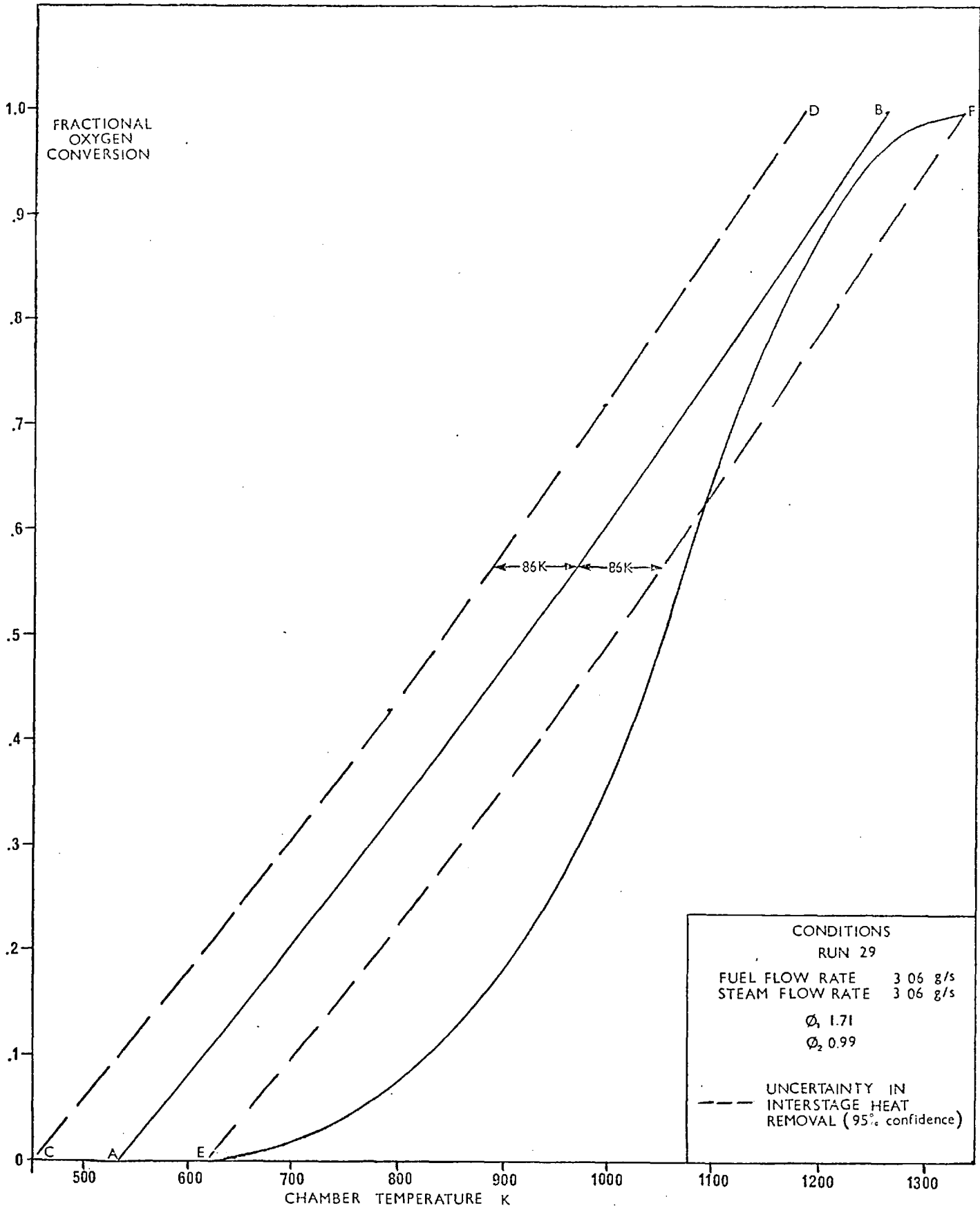


FIG.A5-4 THE EFFECT OF UNCERTAINTY IN INTERSTAGE HEAT REMOVAL ON BLOWOUT CONDITIONS

Optimisation.

Errors in experimental measurements and uncertainties in experimental settings are considered in the optimisation of the system. These errors are particularly important where optimal on-line control is required. For example, the fuel flow rate can be controlled to ± 0.05 g/s which is the accuracy of the fuel setting, hence the optimum can be maintained to this accuracy only.

Determination of the optimal conditions or control set point is influenced by inadequacies of the model as well as limitations in the optimisation program. The heat balance iterations in the model are accurate to $\pm 1\%$. Small changes in some process variables will not alter the heat balances within this accuracy, hence no apparent change in system performance will occur. Thus, the objective function is not continuous, but rather a series of small steps. Where the region near the minimum is flat, the width of these steps increases, and larger changes in each variable are required to obtain a function decrease.

Numerical methods are used to evaluate objective function gradients at each point by perturbing each variable in turn. If the current point lies on one of the wide steps, the size of a variable perturbation may not be large enough to achieve a function change, hence the gradient becomes zero and a false minimum may be reported.

In general, these errors can be avoided by setting the required convergence criteria to approximately the same order of magnitude as the uncertainty associated with the objective function.

APPENDIX 6.CALCULATION OF EQUILIBRIUM CONSTANTS FROM THERMODYNAMIC PROPERTIES.

The equilibrium constant K_e for any reversible reaction is related to the change in the standard free energy ΔF_T viz.

$$K_e = \exp \left(- \frac{\Delta F_T}{RT} \right) \quad A6.1$$

The equilibrium constant can be calculated if ΔF_T is known at the reaction temperature T . The standard free energy can be expressed in terms of enthalpy H_T and entropy S .

$$\Delta F_T = \Delta H_T + T \Delta S \quad A6.2$$

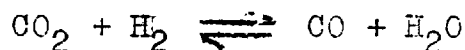
$$\Delta S = \int \frac{\Delta C_p}{T} dT \quad A6.3$$

$$\Delta H_T = \Delta H_0 + \int \Delta C_p dT \quad A6.4$$

Thus the change in standard free energy can be expressed in terms of known quantities C_p and H_0 .

$$\Delta F_T = \Delta H_0 + \int \Delta C_p dT - T \int \frac{\Delta C_p}{T} dT \quad A6.5$$

For example, the water gas shift equilibrium constant at 1500 K is calculated as follows



hence

$$K_e = \frac{x_{CO} x_{H_2O}}{x_{CO_2} x_{H_2}}$$

From Tables (64)

 $\Delta H_{298} \frac{\text{kJ}}{\text{k mole}}$

CO	-110,576
H ₂ O	-241,942
CO ₂	-393,701
H ₂	0

Reaction

- 41,183

From Ref. (62)

$$C_{pCO} = 23.12 + 0.1672 \times 10^{-2} T + 0.537 \times 10^{-6} T^2 - 2.22 \times 10^{-9} T^3 \quad \frac{\text{kJ}}{\text{kmole}}$$

$$C_{pH_2O} = 32.20 + 0.1920 \times 10^{-2} T + 1.068 \times 10^{-6} T^2 - 3.59 \times 10^{-9} T^3$$

$$C_{pCO_2} = 22.26 + 5.975 \times 10^{-2} T - 3.500 \times 10^{-6} T^2 + 7.46 \times 10^{-9} T^3$$

$$C_{pH_2} = 29.10 - 0.1915 \times 10^{-2} T + 0.400 \times 10^{-6} T^2 - 0.87 \times 10^{-9} T^3$$

$$\Delta C_p = 8.96 - 5.424 \times 10^{-2} T + 4.705 \times 10^{-6} T^2 - 12.40 \times 10^{-9} T^3$$

At 1500 K

$$\Delta H_{1500} = -99,427 \text{ kJ/k mole}$$

$$T \Delta S = -87,659 \text{ kJ/k mole}$$

$$\Delta F_{1500} = -11,768 \text{ kJ/k mole}$$

$$K_e = \exp \left(\frac{11768}{8.32 \times 1500} \right) = 2.57$$

This corresponds to a value of $K_e = 3.00$ as predicted using the polynomial equation of reference (1) which is advocated for use up to 1500 K.

APPENDIX 7.DETERMINATION OF STACK GAS SOLIDS AND COLLECTION ERROR.

Basis 1 hr. of operation.

Experiment No. 51.

Settings:
 $F_1 = 3.014 \text{ g/s}$ (fuel)
 $F_2 = 2.917 \text{ g/s}$ (steam)
 Scrubber water = $70.4 \pm 2.8 \text{ g/s}$
 $\phi_1 = 1.85$
 $\phi_2 = 0.98$

Stack Condensate Collection.

Time interval = 12 min.
 Condensate volume = 585 cm^3
 Solids = 1.1244 g.
 Gas temperature = $356 \pm 0.5 \text{ K}$
 Humidity = $0.7026 \pm 0.0281 \frac{\text{kg H}_2\text{O}}{\text{kg dry gas}}$

Cyclone Liquid Collection.

Time interval = 1.25 min.
 Liquid collection = 2400 cm^3
 Solids = 2.6940 g.

Water Balance Calculation.

Water in :

from steam = $2.917 \times 3.6 = 10.50 \text{ kg/hr.}$
 from scrubber = $70.4 \times 3.6 = 253.4 \pm 10.0 \text{ kg/hr.}$
 from combustion = $3.014 \times 0.110 / 2.016 \times 18.0 \times 3.6$
 = 10.7 kg/hr.
 total = $274.6 \pm 10.0 \text{ kg/hr.}$

Water out :

$$\begin{aligned} \text{as water vapour} &= 0.7026 (3.014(1 + \frac{14}{0.98})3.6 - 10.7) \\ &= 109.0 \pm 4.4 \text{ kg/hr.} \end{aligned}$$

$$\text{as condensate} = \frac{585}{1000} \times \frac{60}{12} = 2.9 \text{ kg/hr.}$$

$$\text{as cyclone liquid} = \frac{2400}{1000} \times \frac{60}{1.25} = 116.0 \text{ kg/hr.}$$

$$\text{total} = 227.9 \pm 4.4 \text{ kg/hr.}$$

$$\text{Unaccounted water} = 274.6 - 227.9 = 46.7 \pm 14.4 \text{ kg/hr.}$$

Similar results are tabuled in Table A7-1. It can be seen that when the cyclone sampling line was modified, the unaccounted water was zero within experimental error. This finding led to the conclusion that the unaccounted water was accumulating in the cyclone collection vessel. At a rate of accumulation of 46.7 kg/hr, the cyclone would overflow in approximately 20 min. after the start of sampling if the cyclone were initially empty. During Experiment No.7, the cyclone overflowed. after about 15 min. As a result, sampling times were always kept to less than 10 min.

The influence of scrubber water flow on accumulation rate is shown in Fig.A7-1. If the dependence is linear, no accumulation should occur at scrubber flow rates less than 65 g/s.

TABLE A7-1. SYSTEM WATER BALANCE ERRORS.

Experiment Number *	Scrubber Water Flow Rate g/s	Total Water In g/s	Total Water Out g/s	Unaccounted Water g/s
47	80.9 $\pm$ 2.8	86.9 $\pm$ 2.8	53.4 $\pm$ 1.2	33.5 $\pm$ 4.0
51a	70.4 $\pm$ 2.8	76.4 $\pm$ 2.8	63.5 $\pm$ 1.2	12.9 $\pm$ 4.0
51b	78.3 $\pm$ 2.8	84.2 $\pm$ 2.8	58.6 $\pm$ 1.2	25.6 $\pm$ 4.0
53 ++	84.5 $\pm$ 2.8	89.7 $\pm$ 2.8	89.1 $\pm$ 0.9	0.6 $\pm$ 3.7
54 ++	70.5 $\pm$ 2.8	74.2 $\pm$ 2.8	78.3 $\pm$ 1.2	-4.1 $\pm$ 4.0
* See Reference(3)for experimental conditions.				
++ modified sampling technique.				

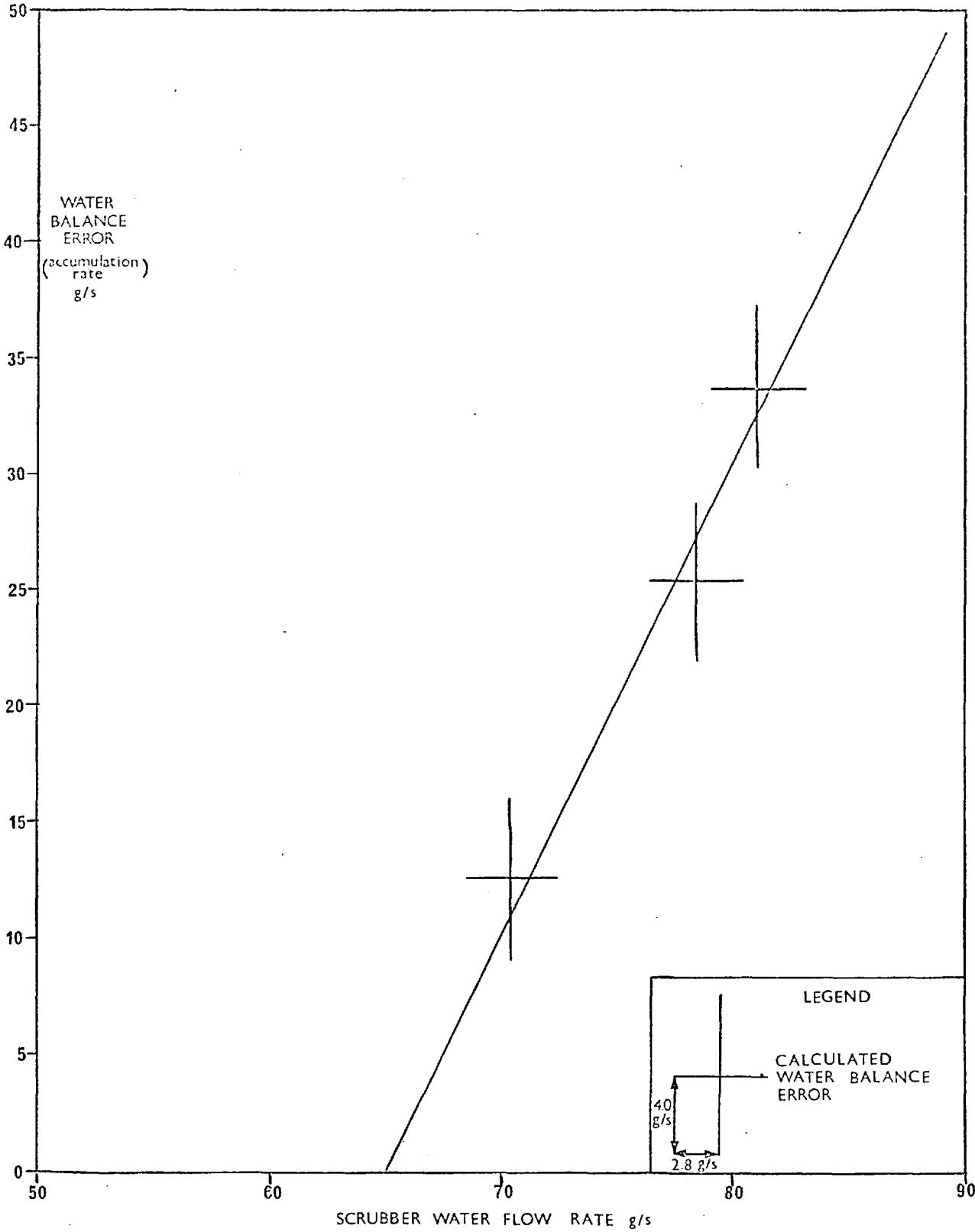


FIG.A7-1 THE EFFECT OF SCRUBBER WATER FLOW RATE ON SAMPLING ERROR

REFERENCES.

1. ARCHER, J.S., EISENKIAM, P.
Multistage combustion of residual fuel oil.
Part I: The effect of using air or steam for the
atomisation of residual fuel oil in a high intensity
first stage.
J. Inst. Fuel (To be published).
2. HTUN, M.N.
Residual fuel oil combustion: The effect of air/fuel
ratio on combustion products formed.
Ph.D. Thesis, University of London (1966).
3. ARCHER, J.S.
An experimental investigation of spray combustion in a
two stage system.
Ph.D. Thesis, University of London (1969).
4. Discussion on 'High temperature corrosion investigations
on an oil-fired boiler at the Marchwood power station'
J. Inst. Fuel 43 (1970) 97.
5. DRAKE, P.F., HARNETT, C.G.
The effects of gas borne carbon, excess oxygen and fuel
sodium content on fuel oil ash corrosion of superheaters.
J. Inst. Fuel 42 (1969) 339.
6. Ministry of Housing and Local Government,
Grit and Dust: The measurement of emissions from boiler
and furnace chimneys.
H.M.S.O. London (1967).
7. IRELAND, F.E., PRICE, D.H.A.
Incineration: Statutory requirements and environmental
aspects.
J. Inst. Fuel 43 (1970) 115.
8. MANNY, E.H., SKOPP, A.
Pollution control of nitrogen oxide emissions from
stationary sources.
A.P.C.A. Paper No. 69-46
62 nd Annual Meeting (New York) 1969.
9. WASSER, J.H.
Effects of combustion gas residence time on air
pollutant emissions from an oil-fired test furnace.
NOPI Technical publication 106ED, Paper 68-WS 110.
10. HOTTEL, H.C., SAROFIM, A.F.
Radiative transfer.
Pub. McGraw-Hill New York (1967).

11. DOMBROWSKI, N. JOHNS, W.R.
Thermal properties of combustion gases from a residual fuel oil-fired high intensity combustion chamber.
J. Inst. Fuel 42 (1969) 194.
12. LOCKWOOD, F.C.
Optimum design of the package boiler.
J. Inst. Fuel 42 (1969) 150.
13. BEER, J.M., LEE, K.B.
The effect of residence time distribution on the performance and efficiency of combustors.
Tenth Symp. (Int.) on Combustion (1964) 1187
Pub. Comb. Inst. (Penn.).
14. LONGWELL, J.P., WEISS, M.A.
High temperature reaction rates in high temperature combustion.
Ind. Eng. Chem. 47 (1955) 1634.
15. HERBERT, M.V.
A theoretical analysis of reaction rate controlled systems.
N.G.T.E. Report R.213 (1957).
16. NUKIYAMA, S., TANASAWA, Y.
Atomisation by means of an air stream.
Trans. Soc. Mech. Eng. (Japan) 14 (1938) 86.
17. WIGG, D.L.
Drop size prediction for twin fluid atomisers.
J. Inst. Fuel 37 (1964) 500.
18. FRASER, R.P., EISENKLAM, P.
Liquid atomisation and the drop size of sprays.
Trans. Instn. Chem. Engrs. 34 (1956) 294.
19. HOPKINS, M.J., EISENKLAM, P.
On the theory of spray reactors.
Paper to be presented.
CHEMECA, 1970 (Australia).
20. EISENKLAM, P., ARUNACHALM, S.A., WESTON, D.A.
Evaporation rates and drag resistance of burning drops,
Eleventh Symp. (Int.) on Combustion (1967) 715.
Pub. Comb. Inst. (Penn.)
21. SHAPIRO, A.H., ERICKSON, A.J.
On the changing size spectrum of particle clouds undergoing evaporation, combustion and acceleration.
Trans. ASME. 79 (1957) 775.

22. MICHAEL, M.I., EL-WAKIL, M.M.
On the self ignition of hydrocarbon mixtures.
Eleventh Symp. (Int.) on Combustion (1966) 1027
Pub. Comb. Inst. (Penn.).
23. BRAGG, S.L.
Application of reaction rate theory to combustion
chamber analysis.
A.R.C. 16170, Report CF 270.
24. HARDCASTLE, J.W.
A theoretical analysis of reaction rate controlled
systems.
Seventh Symp. (Int.) on Combustion (1958) 649
Pub. Butterworths (London).
25. SCHOLTE, T.G., VAGGS, P.B.
Burning velocities of mixtures of hydrogen, carbon
monoxide and methane with air.
Comb. and Flame 3 (1959) 511.
26. KOZLOV, G.I.
On high temperature oxidation of methane.
Seventh Symp. (Int.) on Combustion (1958) 142
Pub. Butterworths (London).
27. FRIEDMAN, R., CYPHERS, J.A.
On the burning rate of carbon monoxide.
J. Chem. Phys. 25 (1956) 448.
28. LEWIS, B., von ELBE, G.
Combustion, flames and explosions of gases.
Pub. Academic Press Inc. (1961) London.
29. HOTTEL, H.C., WILLIAMS, G.C., NERHEIM, N.M.,
SCHNEIDER, G.R.
Kinetic studies in stirred reactors: Combustion of
carbon monoxide and propane.
Tenth Symp. (Int.) on Combustion (1964) 111
Pub. Comb. Inst. (Penn.).
30. SCHOLTE, T.G., VAGGS, P.B.
Burning velocity of hydrogen-air mixtures and mixtures
of some hydrocarbons and air.
Comb. and Flame 3 (1959) 495.
31. BATCHELDER, H.R., BUSCHE, R.M., ARMSTRONG, W.P.
Kinetics of coal gasification.
Ind. Eng. Chem. 2 (1953) 1856.

32. NARASIMHAN, K.S., FOSTER, P.J.
The rate of growth of soot in turbulent flow with
combustion products and methane.
Tenth Symp. (Int.) on Combustion (1964) 253.
Pub. Comb. Inst. (Penn.).
33. GADSBY, J., HINSHELWOOD, C.N.
Mechanism of the steam-carbon reaction.
Proc. Roy. Soc. A187 (1946) 129.
34. HOPKINS, M.J.
Multistage combustion of fuel oil: Reaction rates in
high intensity furnaces.
Ph. D. Thesis, University of London (1970).
35. HEDLEY, A., GULDENPFENNING, F.
Theoretical considerations of the effect of pressure on
the combustion rate of a cloud of spherical particles
of non-uniform size distribution.
Sheffield University FTCE 52/9/1965.
36. MARSDEN, C.
Anthracite dust cloud combustion.
Ph.D. Thesis Univ. of Sheffield (1964).
37. FIELD, M.A., GILL, D.W., MORGAN, E.B., HAWKSLEY, P.W.
Combustion of pulverised coal.
Pub. B.C.U.R.A. (1967).
38. LEE, K.B., THRING, M., BEER, J.M.
On the rate of combustion of soot in a laminar soot
flame.
Comb. and Flame 6 (1962) 137.
39. MAYERS, M.A.
The rate of reduction of carbon dioxide by graphite.
J. Am. Chem. Soc. 56 (1934) 70.
40. DAVIES, T.W., BEER, J.M., SIDDAL, R.G.
The use of a mathematical model for the prediction of
burnout of char suspensions.
C.E.S. 24 (1969) 1153.
41. DEZULAY, E.A.
A comparative investigation of a homogeneous combustion
chamber with a two stage combustion chamber.
Jet Propulsion 2 (1959) 77.
42. EVANGELISTA, J.J., SHINNAR, R., KATZ, S.
The effect of imperfect mixing on stirred combustion
reactors.
Twelfth Symp. (Int.) on Combustion (1968) 901
Pub. Comb. Inst. (Penn.).

43. DRAKE, P.F., HUBBARD, E.H.
Combustion system aerodynamics and their effect on the
burning of heavy fuel oil.
J. Inst. Fuel 39 (1966) 98.
44. HEDLEY, A.B., JACKSON, E.W.
Effect of recirculation on reaction rates in a
homogeneous combustion system.
Trans. Instr. Chem. Engrs. 44 (1966) T 85.
45. HEDLEY, A.B., JACKSON, E.W.
A simplified mathematical model of a pulverised coal
flame showing the effect of recirculation on
combustion rates.
J. Inst. Fuel 5 (1966) 208.
46. HOW, M.E.
Influence of turbulence on the burning rates of
fuel suspensions.
J. Inst. Fuel 4 (1966) 150.
47. WILDER, M.A., WOODWARD, E.C.
An improved analysis of reaction rate controlled systems.
Comb. and Flame 8 (1964) 1.
48. THRING, M., MASDIN, E.G.
The addition of air in stages to a perfectly stirred
reactor.
Comb. and Flame 3 (1959) 125.
49. JAIN, V.K., SPALDING, D.B.
The effects of finite recirculation in a stirred
reactor.
Comb. and Flame 9 (1965) 4.
50. ARCHER, J.S., GROUT, P., EISENKIAM, P.
Multistage combustion of residual fuel oil.
J. Inst. Fuel (to be published).
51. WILDE, D.J., BEIGHTLER, C.S.
Foundations of optimisation.
Prentice-Hall, N.J. (1967).
52. ROSENBROCK, H., STOREY, C.
Computational techniques for chemical engineers.
Pub. Pergamon, Oxford (1966).
53. FLETCHER, R., POWELL, M.J.D.
A rapidly convergent descent method for minimisation.
Comp. J. 6 (1963) 67.
54. FLETCHER, R. (editor).
Keele Conference on Optimisation and Non-linear
Programming.
Academic Press (London) 1969.

55. BIAU, G.E., WILDE, D.J.
Generalised polynomial programming.
Odn. J. Chem. Eng. 47 (1969) 317.
56. PETERS, D.A., COTTON, P.S.
The operation of fired heaters in the petroleum industry.
Symp. Adv. in design and operation of process furnaces
and fired heaters. London (1970).
57. STANDLEY, C.A., NUTCHER, P.B.
Recent advances in high intensity combustion applied
to refinery furnaces and air heaters.
Symp. Adv. in design and operation of process furnaces
and air heaters. London (1970).
58. KUZNETSOV, Y., MAKAROV, A., MELENT'EV, L.
A system of mathematical models for optimising long
term fuel and power balances.
Thermal Eng. 13 (1966) 6 (February).
59. LOOSEMORE, M.J., BRAY, J.W.
Optimisation model for a lead-zinc blast furnace.
Meeting of the Instn. of Chem. Engrs. (Midlands Branch)
Univ. of Nottingham (April) 1967.
60. MAAS, R.B.
Automatic control of furnace off take temperature
in an Imperial Smelting Furnace.
M.A. Thesis, University of Lancaster (1968).
61. ROOTS, W.K., MEEKER, L.D.
Note on the rapid evaluation of thermal process
stability.
J. Inst. Fuel 43 (1970) 120.
62. HOUGEN, O.A., WATSON, M.K., RAGATZ, R.A.
Chemical Process Principles, Part I.
J. Wiley & Sons (New York) 1962.
63. PERRY, R.H., CHILTON, C.H., KIRKPATRICK, S.D.
Chemical Engineers' Handbook.
Mc Graw-Hill (New York) Fifth Edition.
64. SPIERS, H.M.
Technical Data on Fuel.
Pub. British National Committee Fifth Edition.
65. TESNER, P.A.
Formation of dispersed carbon by thermal
decomposition of hydrocarbons.
Seventh Symp. (Int.) on Combustion (1958) 546.
Pub. Butterworths (London).

66. BONNE, U., HOMANN, K.H., WAGNER, H.G.
Carbon formation in premixed flames.
Tenth Symp. (Int.) on Combustion (1964)
Pub. Comb. Inst. (Penn.).
67. MACFARLANE, J.J., HOLDERNESS, F.H., WHITCHER, F.S.
Soot formation rates in premixed C₅ and C₆ hydrocarbon
-air flames at pressures up to 20 atm.
Comb. and Flame 8 (1964) 215.
68. OSHIMA, Y., FUKUDA, Y.
Effect of ash on the combustion characteristics
of carbons.
Ind. Eng. Chem. 27 (1935) 212.
69. STREET, P.J., TWAMLEY, C.S., WEBSTER, J.M.
Studies of the rapid combustion of single fuel
particles.
J. Photo. Science 3 (1970) 113.
70. ARTHUR, J.S.
Reactions between carbon and oxygen.
Trans. Faraday Soc. 47 (1951) 164.
71. GADSBY, J., LONG, F.J.
Mechanism of the carbon dioxide-carbon reaction.
Proc. Roy. Soc. A193 (1948) 357.
72. EFROYMSON, M.A.
Step-wise multiple regression.
'SHARE' General Programme Library E.R.MPR 3(PA) 3143.
73. Composition of 'Vanilla' residual fuel oil,
Admiralty- NGTE Communication (1955).
74. FRASER, R.P., DOMBROWSKI, N., JOHNS, W.R.
Cooling hot gas with evaporating spray.
Brit. Chem. Eng. 8 (1963) 390.
75. EISENKLAM, P., ARUNACHALAM, S.A., WESTON, J.A.
Evaporation rates and drag resistance of burning drops.
Eleventh Symp. (Int.) on Combustion (1967) 715
Pub. Comb. Inst. (Penn.).
76. REID, R.C., SHERWOOD, T.K.
The Properties of Gases and Liquids.
McGraw-Hill (New York) Second Edition Ch.11.
77. Private Communication.
Urquharts (1926) Ltd.
Cost of high intensity combustion equipment (1970).

78. PETERS, M.S., TIMMERHAUS, K.D.
Plant Design and Economics for Chemical Engineers.
McGraw-Hill (New York) Second Edition.
79. Private Communication.
Sturtevant Engineering Co. Ltd.
Cost of turbo-exhausters and blowers (1970).
80. BAUMAN, H.C.
Fundamentals of Cost Engineering in the Chemical
Industry.
Reinhold (New York) 1964.