GAS REBURNING SYSTEM DESIGN
For
CHEROKEE UNIT #3

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1.0 INTRODUCTION

Nitrogen oxides have been recognized as air pollutants due to their role in producing smog and their effects on human and animal health. Emissions of NO_x in the United States are produced by a wide range of sources with coal combustion in utility boilers representing a significant fraction. Many coal fired utility boilers are already required to control NO_x emissions to some extent: units designed after 1971 are required to meet New Source Performance Standards (NSPS). In addition, some older units are subject to controls in areas where ambient levels of NO_x exceed air quality standards.

Under this program Gas Reburning will be used to control NO_x emissions from a coal fired utility boiler. Gas Reburning involves cofiring 15 to 20 percent natural gas with coal. The gas is injected into the furnace above the main coal combustion zone to produce a slightly fuel rich zone where NO_x produced by the combustion of coal is "reburned" and reduced to atmospheric nitrogen (N_2). Additional over fire air is then added above this "reburning zone" to burn out the combustibles. Gas reburning alone can achieve about 60 percent NO_x reductions. This process also results in approximate reductions of SO_2 by 20 percent, particulate matter by 20 percent, and CO_2 by 8 percent.

The host boiler is Cherokee Station Unit 3 which is owned and operated by the Public Service Company of Colorado (PSCo). This is a 172 MW coal fired utility boiler which burns Colorado bituminous coal and has a fabric filter particulate control device. Low NO_x burners will be simultaneously installed in the boiler (not under this contract) in addition to the gas reburning process. The gas reburning and low NO_x burner technologies are fully compatible and synergistic. The installation of low NO_x burners by themselves will result in a 30 to 50 percent reduction in NO_x levels. Combining low NO_x burners with gas reburning can increase the total NO_x reduction to as much as 75 percent.

This project will be conducted in three phases. In the initial phase EER will design the Gas Reburning system. In Phase 2 the Gas Reburning system will be installed on the Host Unit and in the final phase a comprehensive assessment of the performance of the system will be conducted. The initial phase involves six tasks:

Task 1: Project Management
Task 2: Process Design
Task 3: Engineering Design
Task 4: Environmental Reports, Permitting, Plans, and Design
Task 5: Baseline Test and Phase 3 Test Plan
Task 6: Technology Transfer

This report describes the results of the work performed under Task 2: Process Design. The Process Design task involves the application of various experimental and analytical studies to determine the locations and configurations of the natural gas and overfire air injectors and the corresponding flow rates. Results of other tasks are described in separate volumes.

2.0 DESIGN METHODOLOGY

The application of gas reburning to specific boiler systems essentially requires the detailed specification of injectors (number, size, velocity, locations) for the injected natural gas and the reburn air. To achieve this, EER has developed a generalized design methodology as a result of a number of previous projects. This methodology involves the application of various experimental and analytical tools to adapt the reburning process requirements to the specific boiler geometry and operating parameters. The design of a gas reburning system consists of the following six steps:

1) An initial field test to obtain a limited amount of data for thermal performance characterization and use in the later analytical studies;
2) The construction of an isothermal flow model to fully characterize the flow field and test the injection systems;
3) Heat transfer modeling to provide temperature profiles for use in the design of the reburning system and to determine the impact of the system on boiler performance;
4) Preliminary design specifications are obtained based on the process requirements combined with heat transfer modeling results, flow model data and boiler structural constraints;
5) The preliminary injection system design is tested on the isothermal flow model and modified, if necessary, to obtain a final detailed injector design; and
6) Performance predictions based on the final design to determine NO_x reduction potential and of the potential impacts on boiler thermal performance. The boiler heat transfer model is exercised over the range of operational variables to determine the impact on parameters such as temperature distribution, steam generation rate, steam conditions, and overall unit performance. Model results are also used to evaluate secondary performance parameters, such as the impact on predicted wall deposit temperatures which would indicate changes in slagging propensity.

3.0 PROCESS DESIGN

The design process for the gas reburning injection system involves several steps combining computational and experimental modeling methods. Initially the baseline operating conditions of the boiler are used to design an isothermal model of the boiler and to determine the gas reburning injection elevations. The injection system for the reburning fuel and over fire air are designed based on the gas reburning conditions and then modeled on the isothermal model. The jet characteristics are analyzed on the model and the injector designs are refined to achieve rapid mixing and adequate coverage of the furnace.

3.1 Isothermal Model

The isothermal model of the Cherokee Unit 3 is illustrated in Figure 3-1. The burner zone of this furnace consists of a 4 x 4 matrix of burners (8 burners on each side of the furnace division wall). Three swirl configurations for the new low-NO_x burners were considered. The initial burner swirl configuration specified that all burners have a clockwise swirl when viewed facing the boiler front wall (Figure 3-2). Inspection of this configuration revealed that a large recirculation zone formed above the top burner row (Figure 3-3). A velocity profile taken at the nose (Figure 3-4) indicated the flow was strongly non-uniform. The second burner swirl configuration consisted of burners with opposing swirl directions (Figure 3-5). The opposing burner swirl configuration eliminated the recirculation zone. A comparison of the first and second burner swirl configuration velocity profiles (Figure 3-6) illustrates the change in flow behavior for the two configurations. The velocity profile of the first configuration shows reduced velocities in the front wall area while the second configuration provided a substantially more uniform velocity profile. The final burner swirl configuration (Figure 3-7) was used in the evaluation of the gas reburning system design. The velocity profile (Figure 3-8) taken using this final configuration was similar to the profile taken with the opposing burner swirl configuration with high velocities along the side and rear walls although the velocity gradients were not as severe and no recirculation was noticed.

3.2 Gas Reburning System

The gas reburning injection systems were initially designed using empirical relationships. The injector designs were then installed on the isothermal model and further injector modifications were

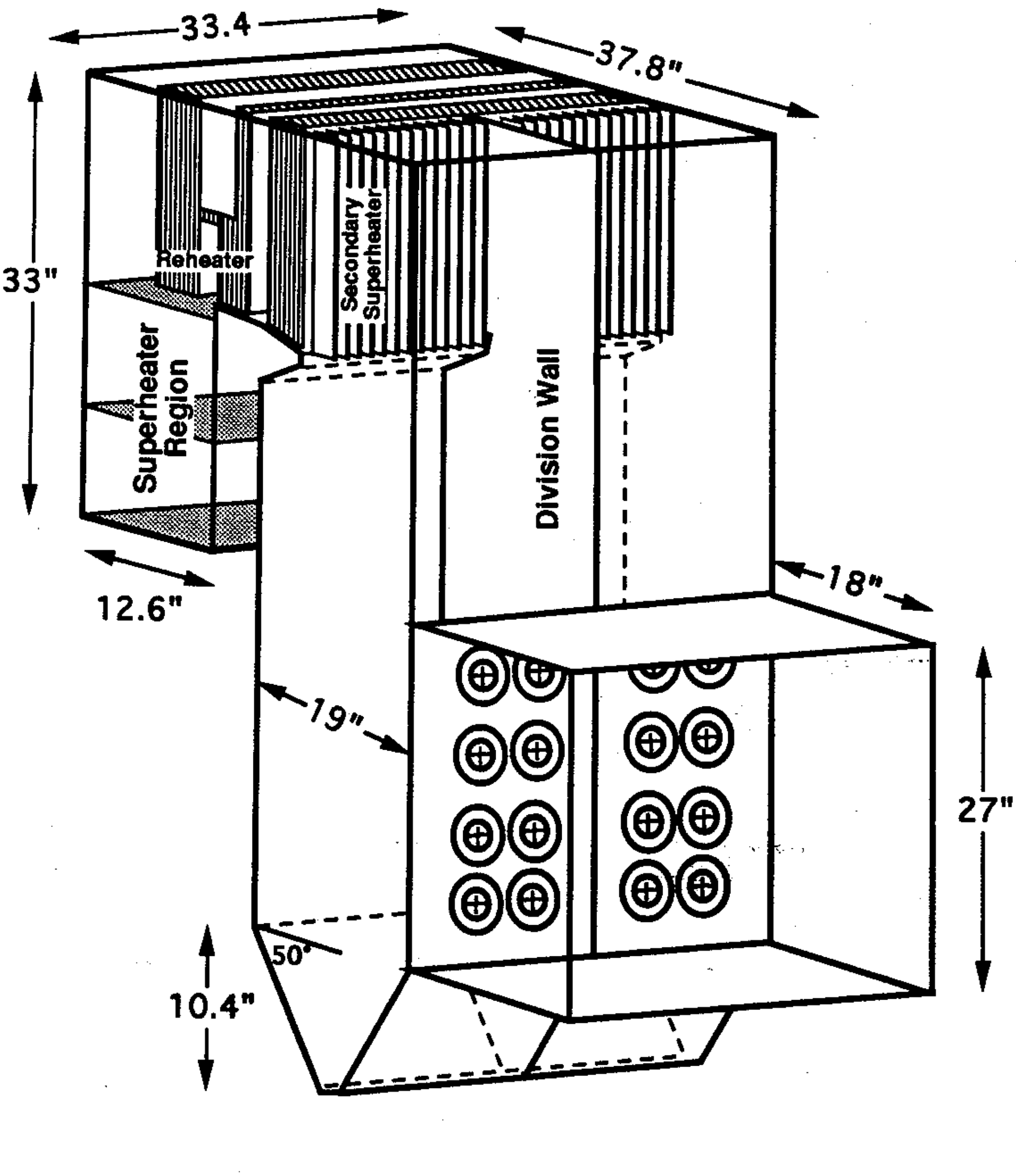


Figure 3-1. Isothermal model of Cherokee Unit 3.

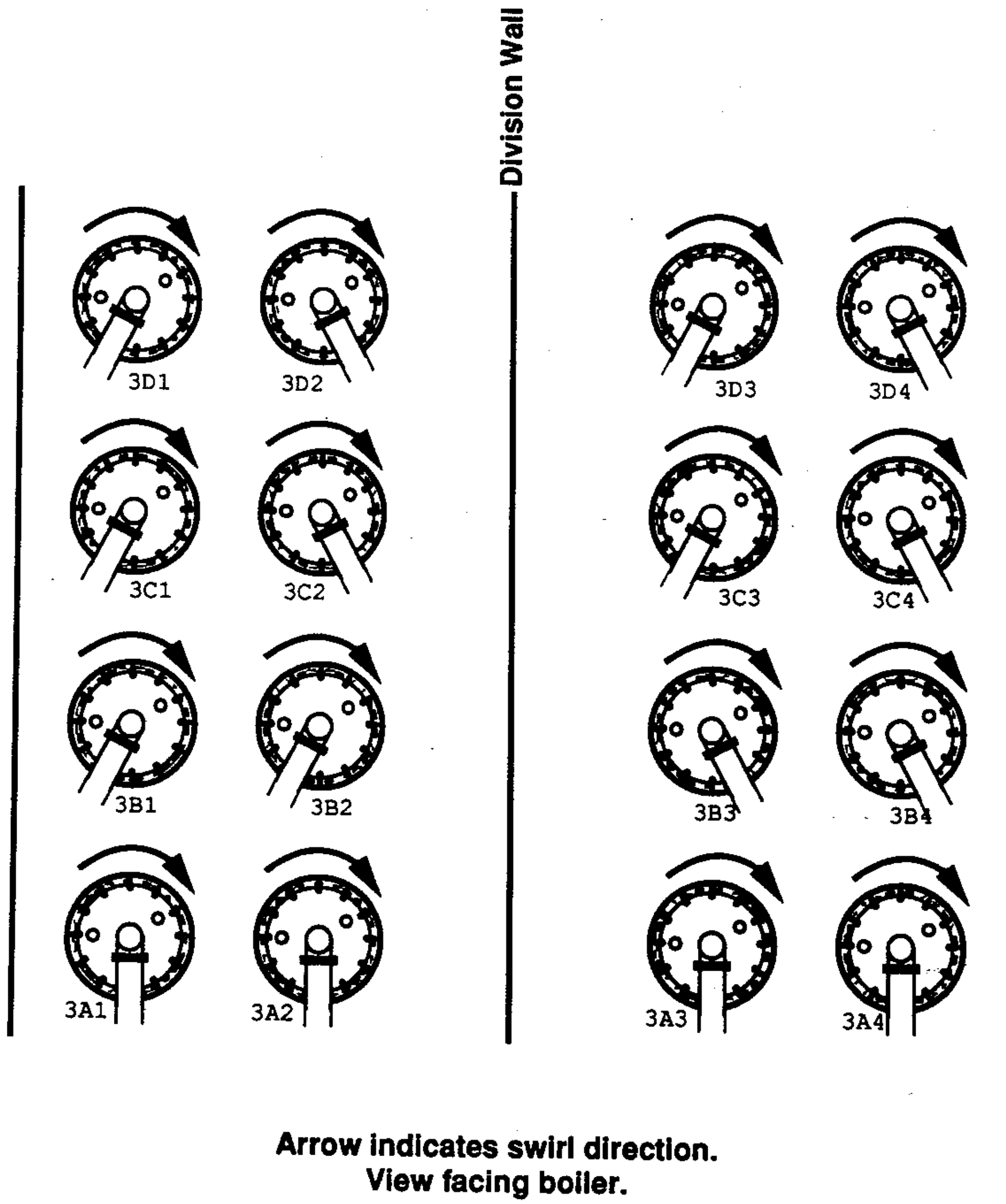


Figure 3-2. Clockwise burner swirl configuration.

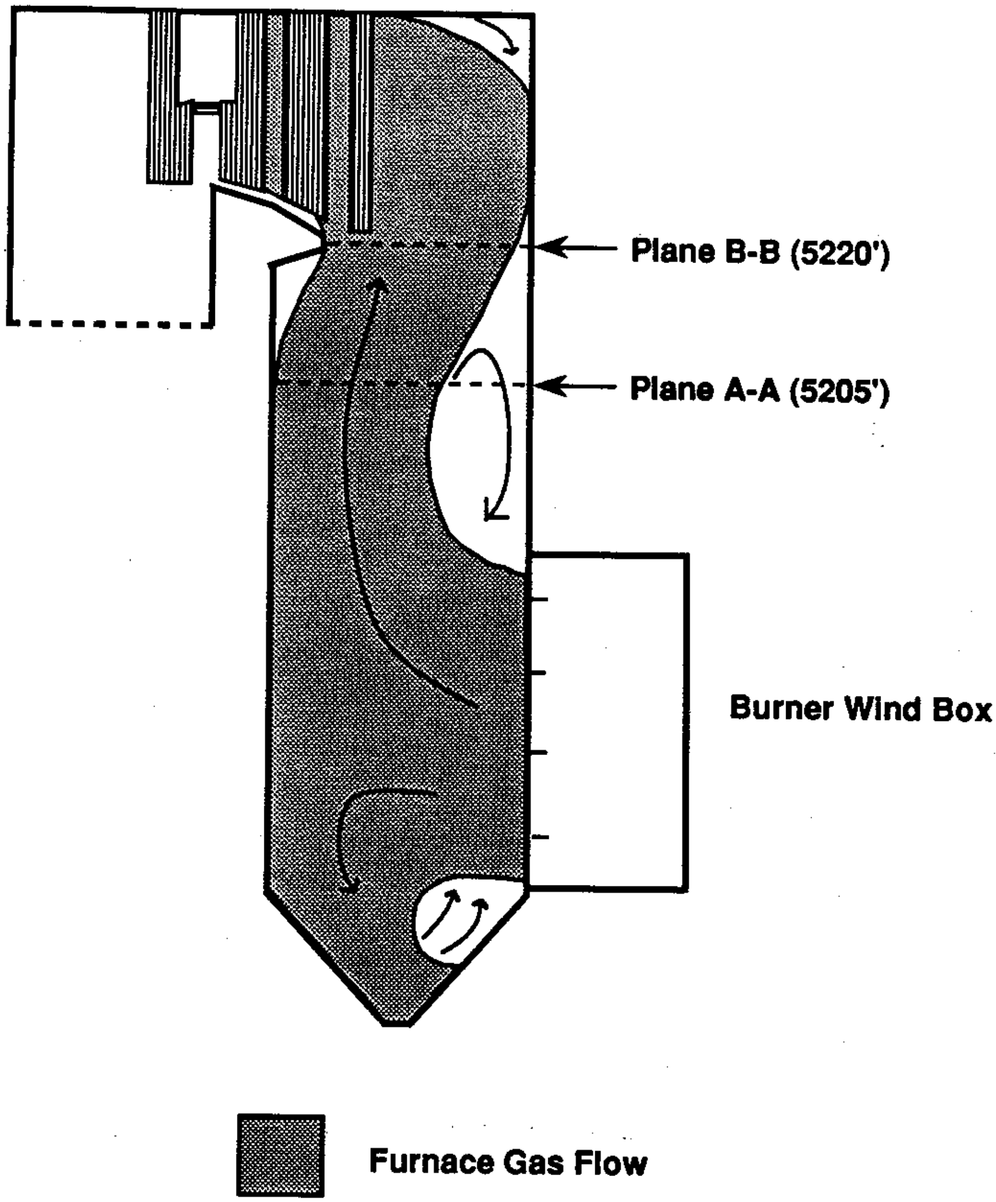
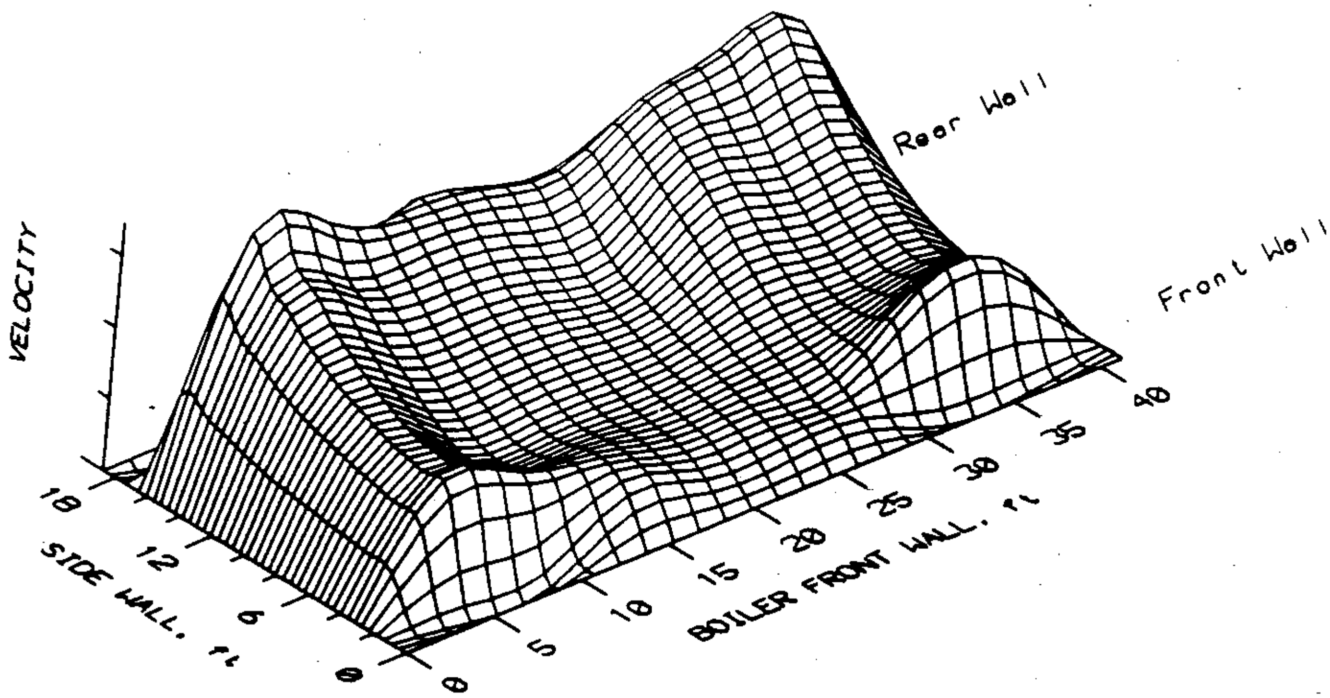


Figure 3-3. Baseline (full load, without reburning) furnace gas flow pattern for clockwise burner swirl configuration.



Nose plane B-B
Full load

Figure 3-4. Baseline (full load, without reburning) furnace gas velocity profile for clockwise burner swirl configuration.

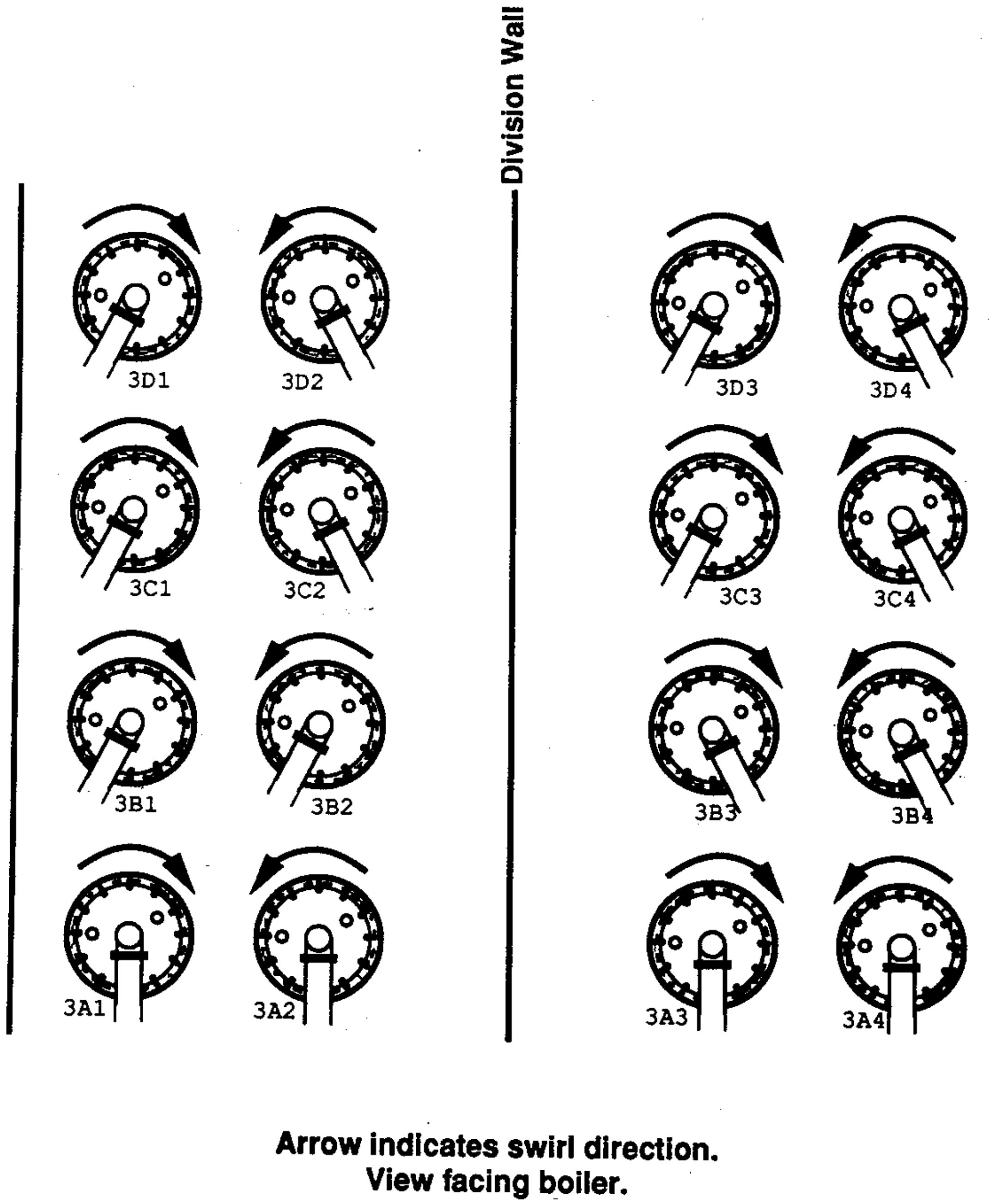
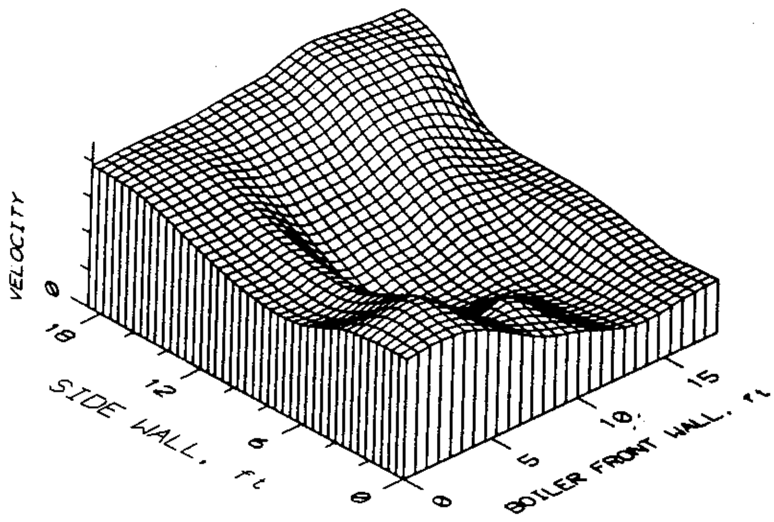
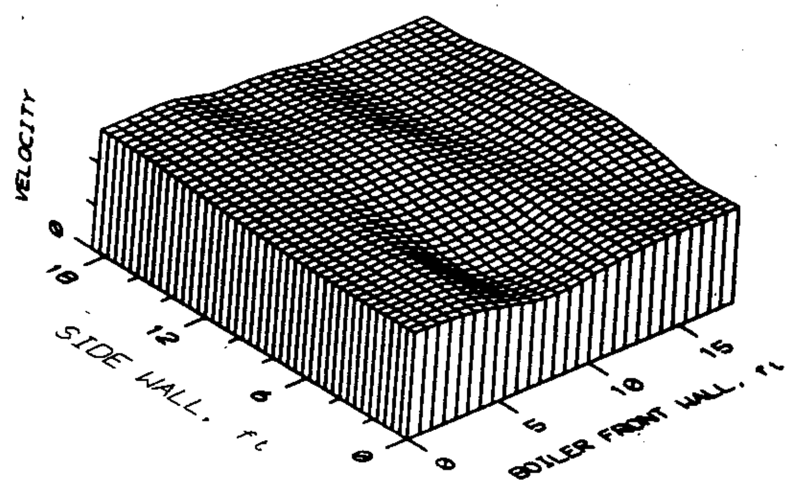


Figure 3-5. Opposing burner swirl configuration.



Full load clockwise burner swirl configuration



Full load opposed burner swirl configuration

Figure 3-6. Baseline (full load, without reburning) velocity profile at mid-furnace plane A-A.

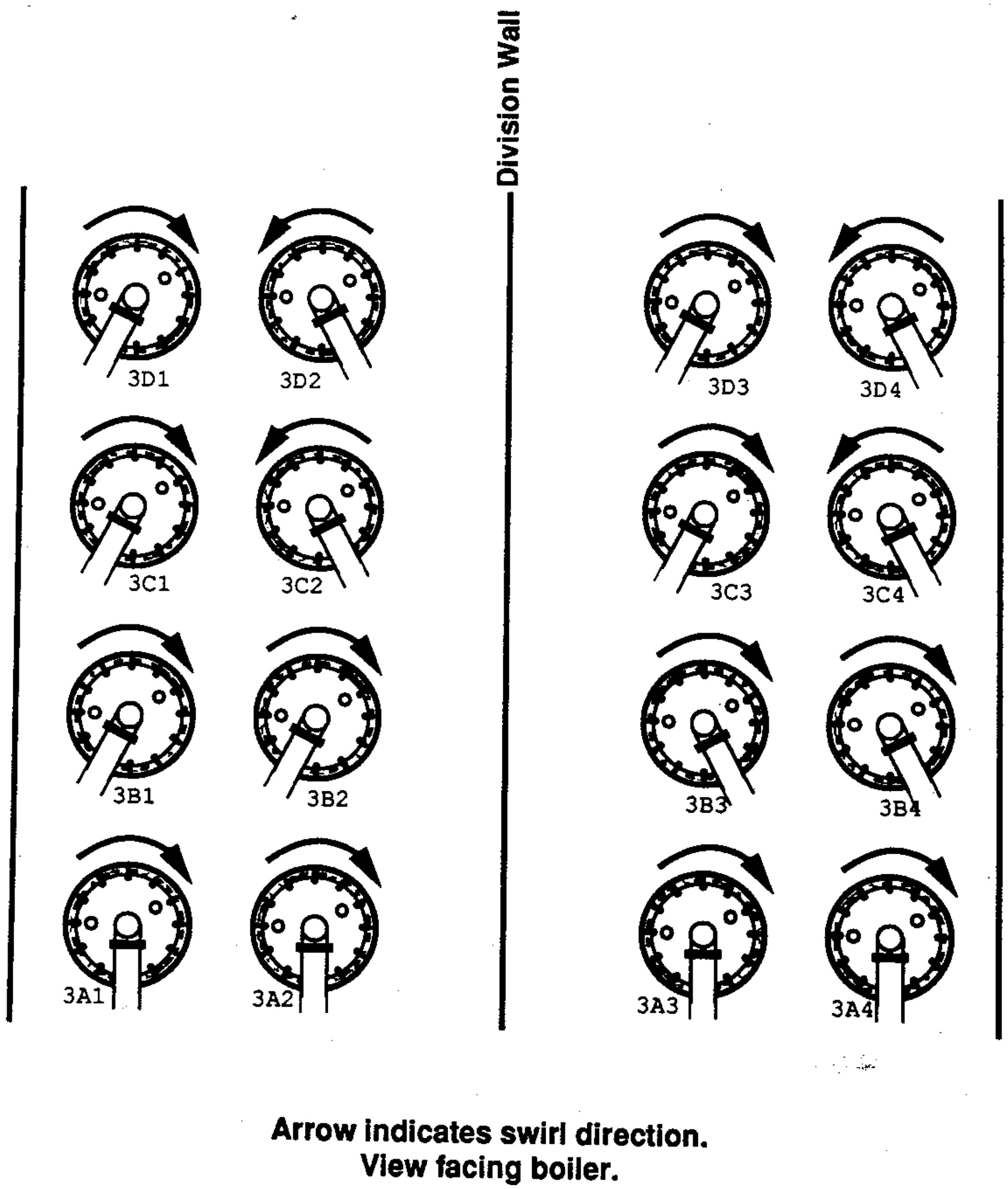
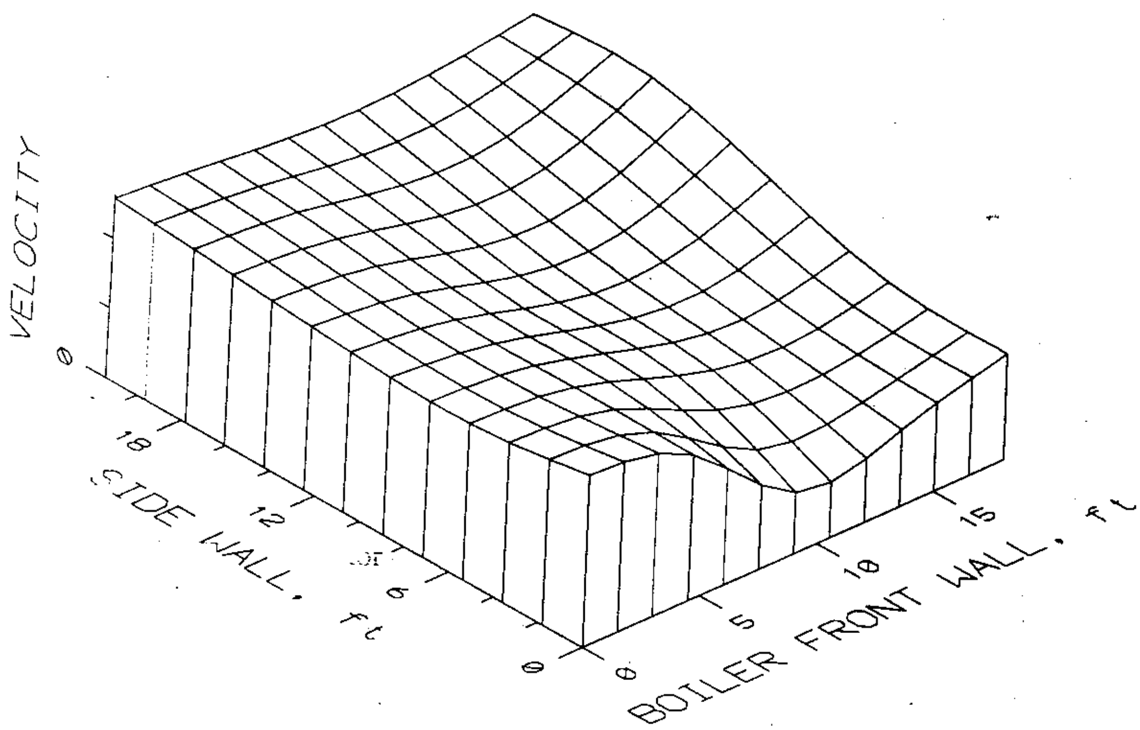


Figure 3-7. Mixed burner swirl configuration.



Full load mixed burner swirl configuration

Figure 3-8. Baseline velocity profile at mid-furnace plane A-A.

made based on the isothermal flow modeling studies. Process parameters used in the design of the Cherokee gas reburning system are listed in Table 3-1.

TABLE 3-1. PRELIMINARY DESIGN BASIS FOR CHEROKEE

Full Load Gas Reburning Conditions: Stoichiometries; Primary burner zone Reburning fuel zone Burn-out zone Natural Gas Flow MW Generation	 1.1 0.9 1.16 18% of heat input 172 MWe
Low Load Gas Reburning Conditions: Stoichiometries; Primary burner zone Reburning fuel zone Burn-out zone Natural Gas Flow MW Generation	 1.2 0.9 1.4 24% of heat input 86 MWe

3.2.1 Injector Elevations

Elevations for the reburning fuel and over fire air injectors were selected to provide the necessary residence times and temperatures for the reburning process to operate efficiently. Revisions to these elevations were made when access (due to buckstays, etc.) through the furnace walls was limited.

The elevation for the reburning fuel-injection system is 5193' 1.3" (shown in Figure 3-9). Both front and rear walls can be accessed at this elevation. The reburning fuel elevation was calculated using full load baseline conditions.

The initial design elevation for the over fire air injectors was 5209' 1.3" based on full load baseline operating conditions. The final elevation was raised to 5210' 10.3" to avoid a buckstay. At this elevation there was only room for a front wall injection system. Later investigation of the over fire air coverage resulted in tilting the injectors downward 10 degrees. This caused the burnout zone to begin below the 5210' 10.3' elevation and in effect lowered the elevation to approximately the initially specified elevation of 5209' 1.3".

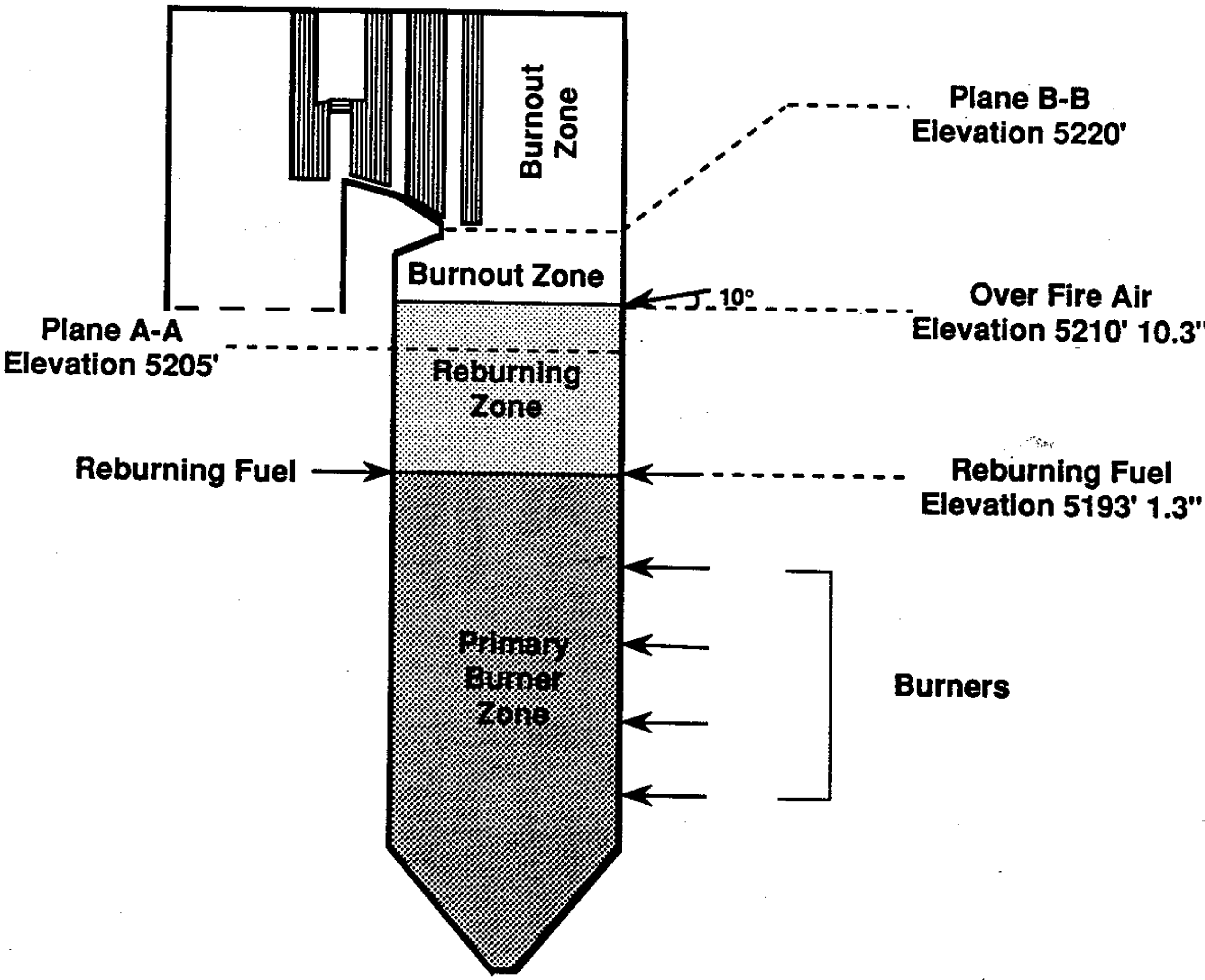


Figure 3-9. Injection elevations of gas reburning injection systems.

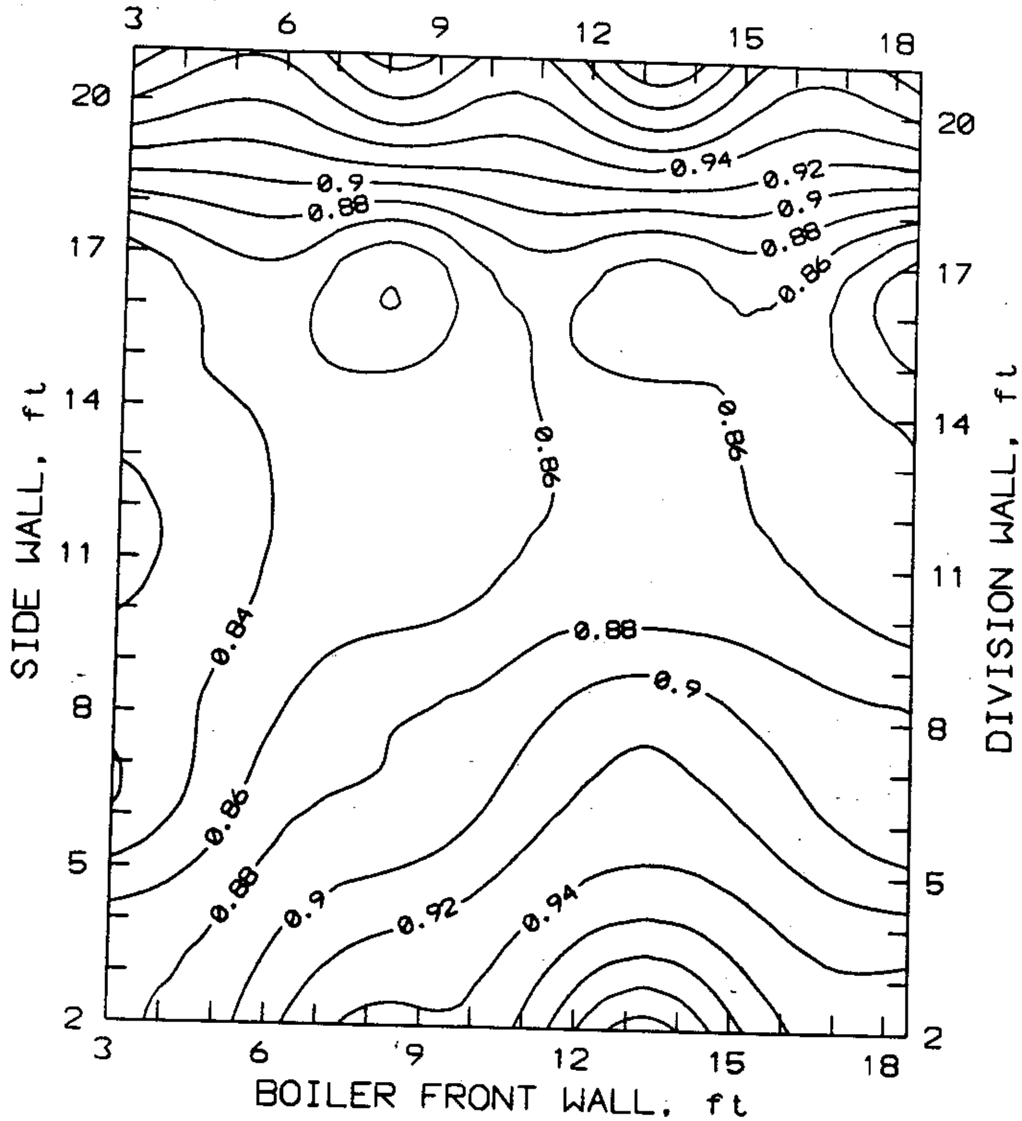
3.2.2 Injector Configurations

The initial injectors were designed to ensure adequate mixing of the natural gas and over fire air with the surrounding flow of furnace gases. The reburning fuel system was initially designed with 3% flue gas recycled (FGR) through the injectors to ensure adequate mixing of the natural gas. The FGR was subsequently increased to 3.4% to improve coverage. The initial designs specified the least number of injectors to minimize retrofit costs. The number of injectors was also subsequently increased to improve coverage. The reburning fuel and over fire air injectors were then installed on the model. The injector designs were initially inspected at full load conditions and then at low load conditions to ensure adequate performance over the full range of boiler loads.

3.2.2.1 Reburning Fuel Injection

The preliminary reburning fuel jet design consisted of six rear wall fired 6.5 inch diameter injectors. The flow from these rear wall injectors did not penetrate across the boiler adequately to provide coverage to the front wall. The jet coverage was also poor in the lateral direction (the direction perpendicular to the furnace flow and jet flow) leaving unmixed flow between the jets. A second jet design consisted of eight rear wall fired jets with 4.6 inch diameter nozzles designed to produce higher jet velocities. The higher velocity produced by these smaller jets improved coverage to the front wall and reduced coverage near the rear wall while the increased number of injectors improved the lateral jet coverage. A dispersion profile normalized to the reburning zone stoichiometry (0.9) is illustrated in Figure 3-10 for this configuration. The eight injector design produced low coverage at the rear wall where the local concentrations reached a maximum stoichiometric ratio of 1.2, well above the desired fuel-rich (0.9) stoichiometry. Weak coverage also apparent on the front wall is a function of the local flow pattern from the burner zone.

A front and rear wall injection system was designed to improve the wall to wall jet coverage. The injector design consisted of twelve 4.5 inch diameter nozzles with six injectors on the front wall and six injectors on the rear wall in an opposed fire configuration. The FGR was also increased from 3% to 3.4% to improve the mass flow per injector and aid mixing. This injector design over penetrated leaving low coverage along the front and rear walls and did not provide adequate lateral coverage.



Mid-furnace plane A-A
Full load conditions
Rear wall injection

Figure 3-10. Eight 4.6 inch diameter reburning fuel injector design.

The final reburning injection system consisted of sixteen 5.5 inch diameter front and rear wall jets (Figure 3-11). This configuration reduced the jet velocities and therefore penetration and provided adequate wall to wall and lateral coverage. Dispersion measurements for full and low load conditions are illustrated in Figure 3-12 and 3-13. The dispersion profiles, normalized to the reburning zone stoichiometry, indicate that fuel-rich conditions will be achieved at the mid-furnace plane A-A (Figure 3-9) at both loads. Standard deviations of 0.05 and 0.06 from the mean (0.9) for the high and low load conditions, respectively, indicate that this configuration will provide a highly uniform dispersion of the reburning fuel. The final injector design is summarized in Table 3-2.

TABLE 3-2. INJECTOR DESIGNS FOR CHEROKEE

Injector	# Nozzles	Location	Diam. (in.)	Elevation (ft.)	Max. Vel. (ft/sec)	Nozzle Pressure ("WC)	%FGR
Reburning	8 8	Front Wall Rear Wall	5.5	5193' 1.3"	186	3.83	3.4
Over Fire	6	Front Wall	20.5	5210' 10.3"	190	3.95	--

3.2.2.2 Over Fire Air Injection

The initial over fire air injector design consisted of four 29 inch diameter front wall nozzles. Inspection of this design revealed that the jet flow did not penetrate sufficiently to cover the flow around the boiler nose as illustrated in Figure 3-14. Smaller diameter, high velocity injectors (four on the front wall) were installed to improve jet coverage at the rear wall. These injectors improved penetration but decreased the lateral jet coverage.

To enhance the lateral jet coverage six 18 inch front wall injectors were installed. A dispersion profile (normalized to a stoichiometry of 1.156) measured using this configuration is illustrated in Figure 3-15. The increased number of injectors improved the lateral coverage but resulted in low coverage along the front wall and in the side/rear wall corner. Low coverage along the side wall and in the corner are due to high velocities in these regions. Coverage along the nose, away from the side wall, was adequate but the jets over penetrated and impinged on the rear wall causing recirculation into the lower furnace. This recirculation effectively reduces the residence time of the reburning fuel and can have a detrimental effect on NO_x reduction. The over penetrating jet

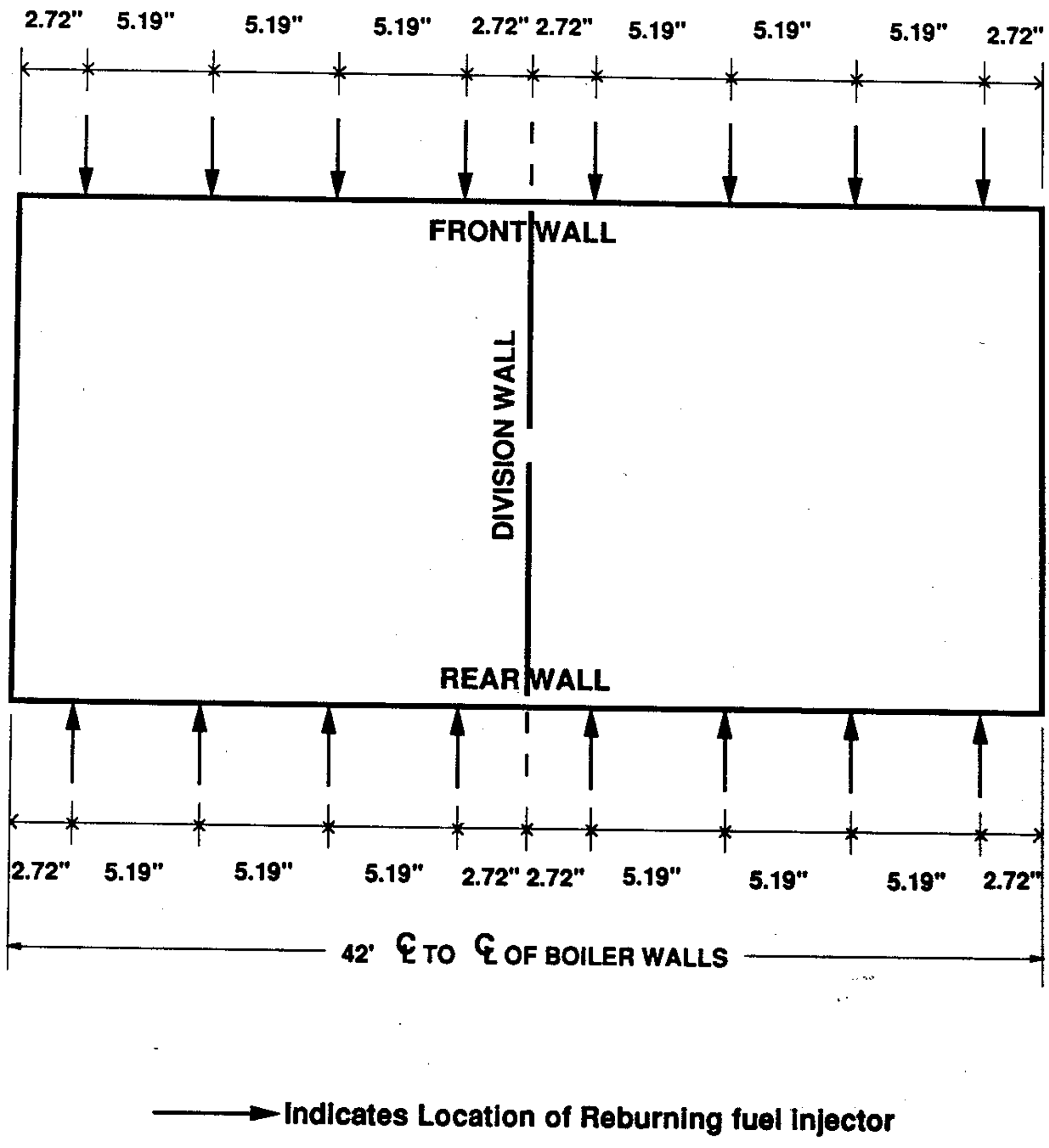


Figure 3-11. Plan view of reburning fuel injection nozzles.

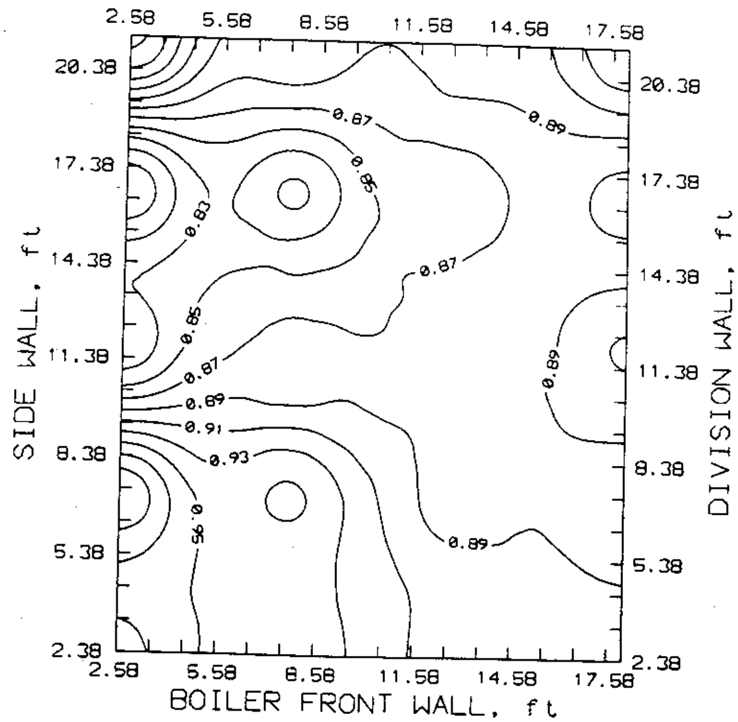


Figure 3-12. Sixteen reburning injectors operating at full load (172 MW) condition.

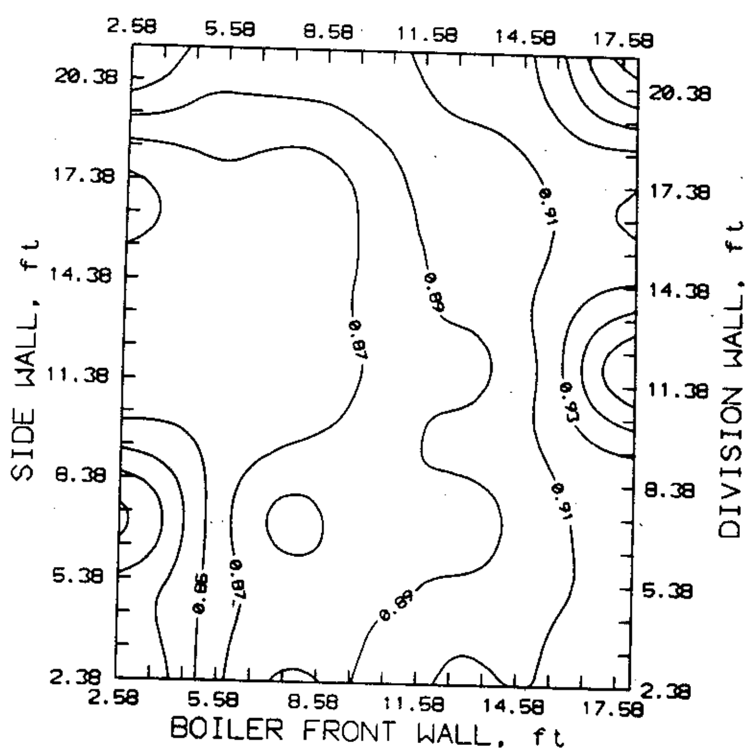


Figure 3-13. Sixteen reburning injectors operating at low load (86 MW) condition.

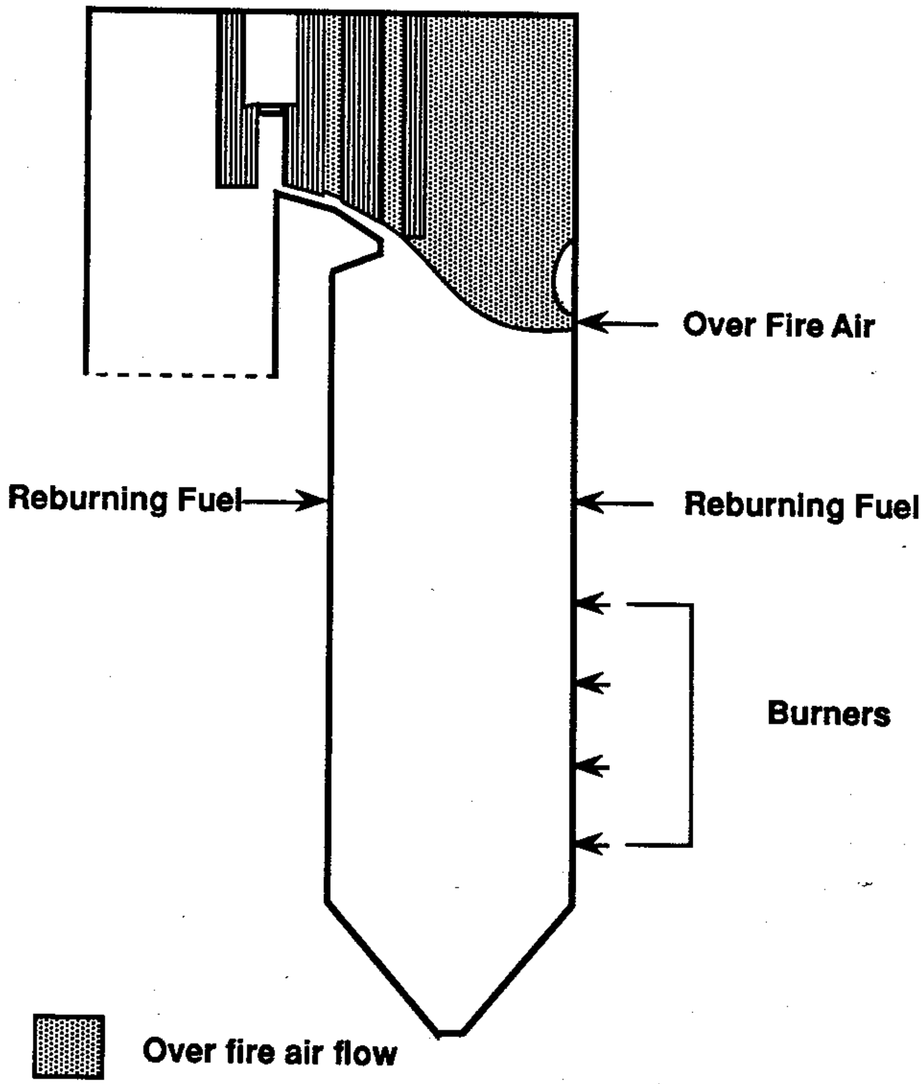
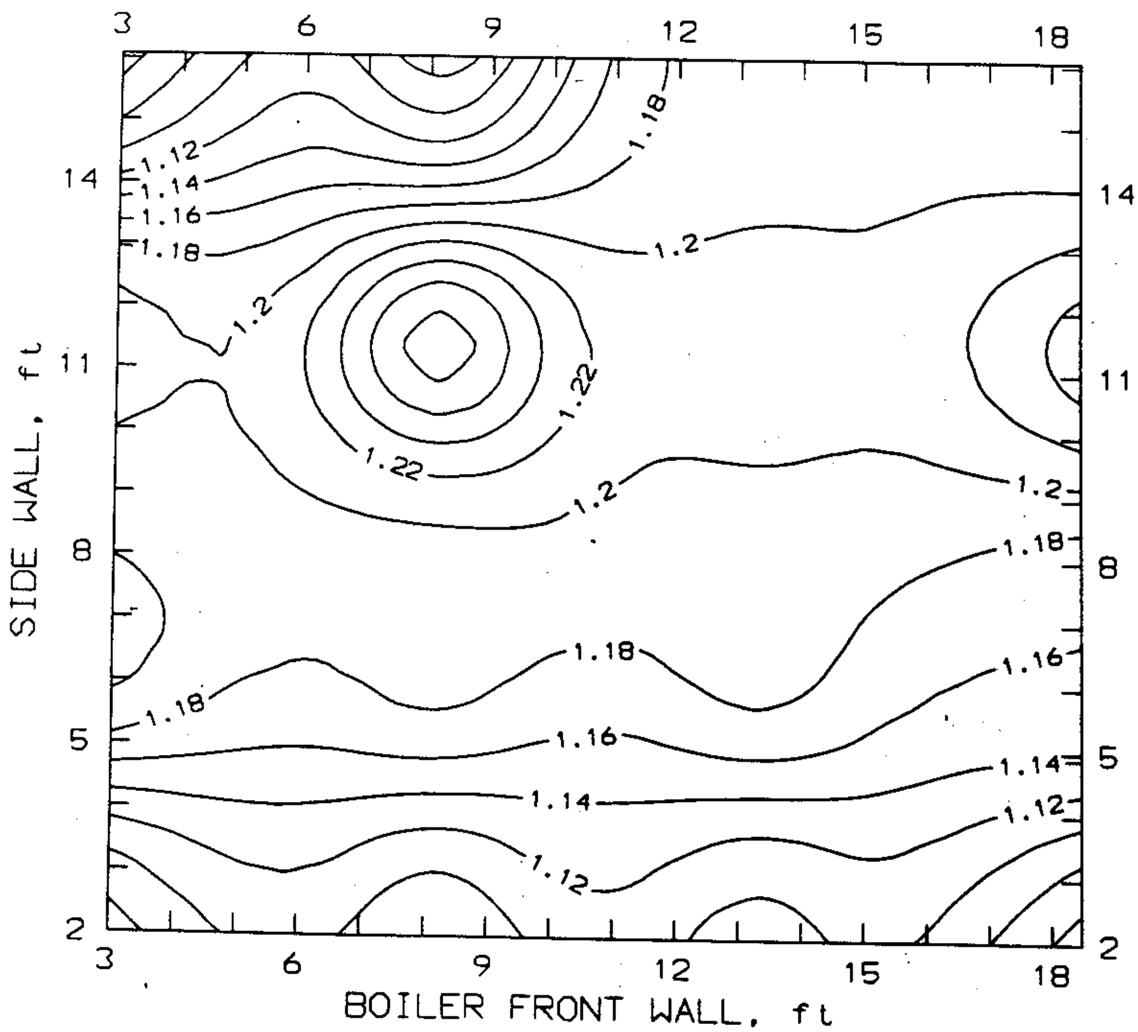


Figure 3-14. Dispersion pattern for initial over fire air injector design.



Nose plane B-B
Full load conditions

Figure 3-15. Six 18 inch diameter over fire air injector design.

trajectories provided very little residence time prior to the secondary superheater. Short residence times in the burnout zone reduce the time available for carbon burnout and can result in ash with a high carbon content. Several lower velocity jet configurations using six front wall injectors were considered to eliminate the recirculation and improve residence time. The lower velocity jets eliminated the recirculation but also reduced coverage around the nose.

The injectors of a six jet configuration were tilted downward to improve coverage, reduce recirculations and increase residence time. Tilting the injectors downward increased the trajectory length prior to the super heaters and increased the over fire air residence time. Various injector diameters were inspected to achieve the necessary penetration for uniform coverage at the nose. The jets tended to disperse into a wider area than horizontal fired jets due to the opposing flow of the furnace. The front wall coverage was quite uniform because of the improved dispersion of the jet flow. The jet momentum carried flow towards the rear wall where the flow became predominantly vertical and eliminated the recirculation. The flow pattern for the tilted injectors is compared to the horizontal injectors in Figure 3-16.

The final over fire air injector design, shown in Figure 3-17, is comprised of six front wall 20.5 inch diameter injectors. The dispersion profiles for the over fire air jet flow were obtained at an elevation of 5220' (Figure 3-9). The dispersion profiles (Figure 3-18 and 3-19) are normalized to 1.156 and 1.4 stoichiometry for the high and low load cases, respectively. Both dispersion profiles show that the fuel-lean conditions required in the upper furnace exist at the nose plane. The full load dispersion profile has a standard deviation of 0.077 from the over fire air stoichiometry of 1.156 while the low load dispersion profile has a standard deviation of 0.15 from the over fire air stoichiometry of 1.4. In all cases the standard deviation is less than 10% of the mean which is considered adequate. The over fire air design is summarized in Table 3-2.

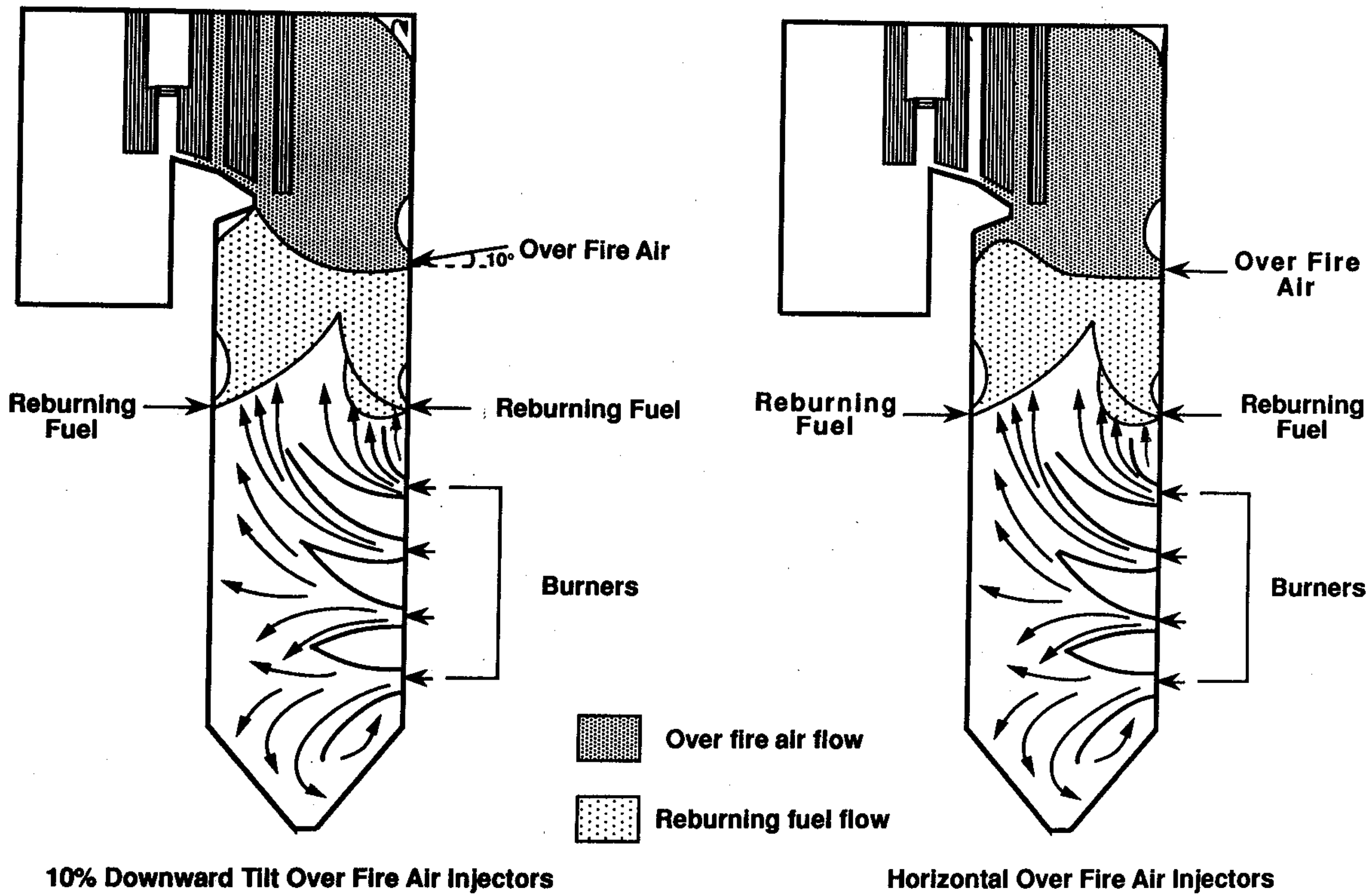


Figure 3-16. Dispersion pattern for Cherokee Unit 3 gas reburning.

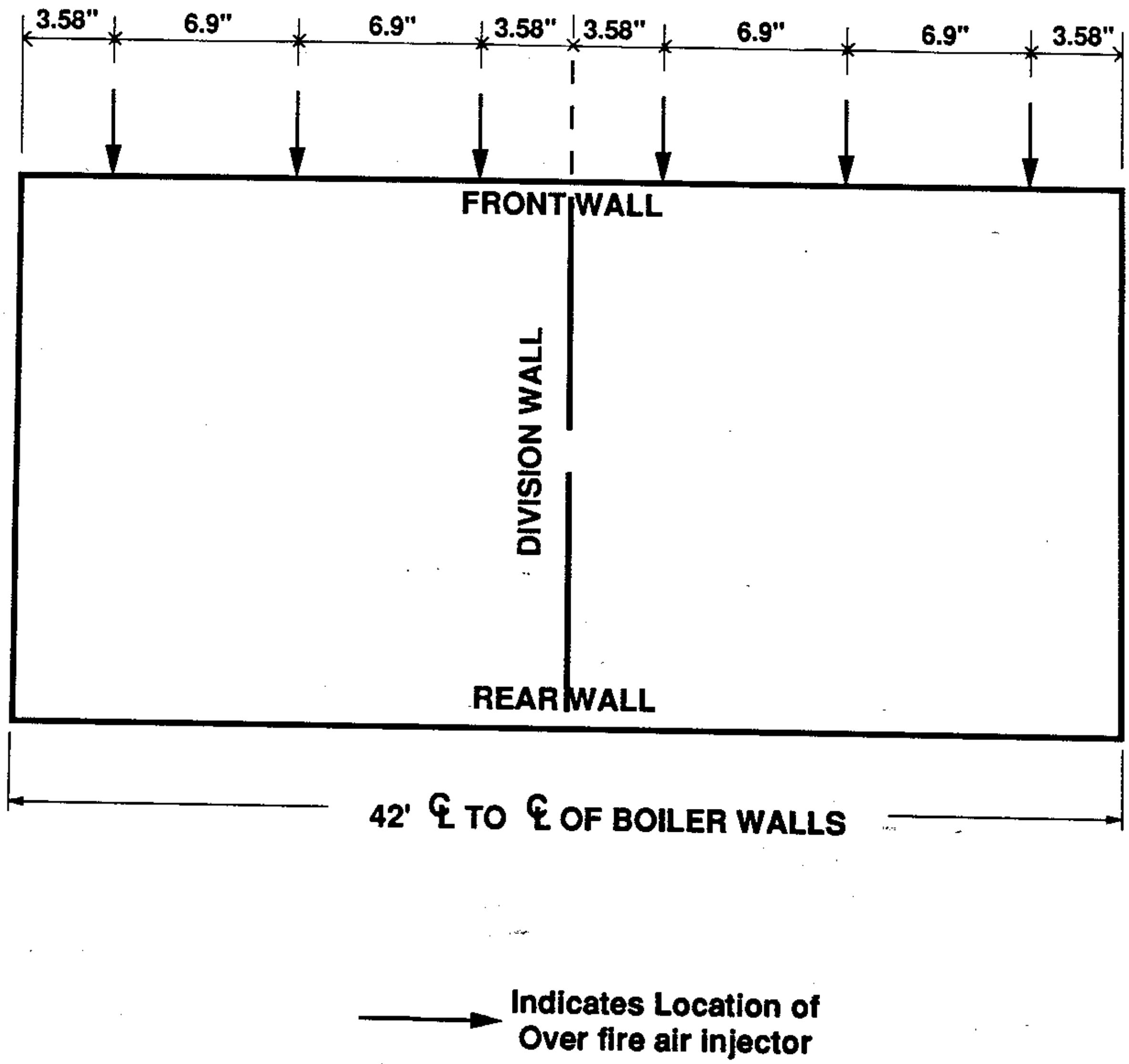


Figure 3-17. Plan view of the over fire air system.

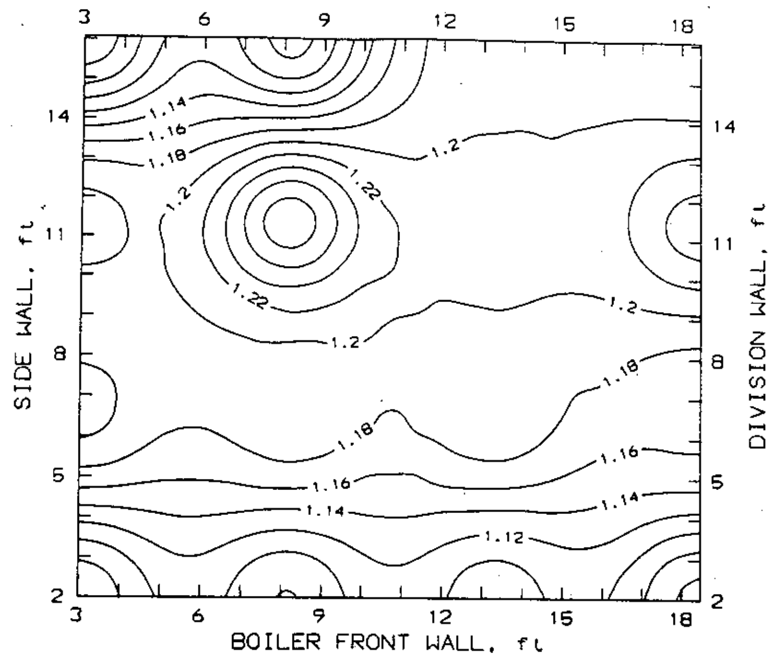


Figure 3-18. Six over fire air injectors operating at full load (172 MW) condition.

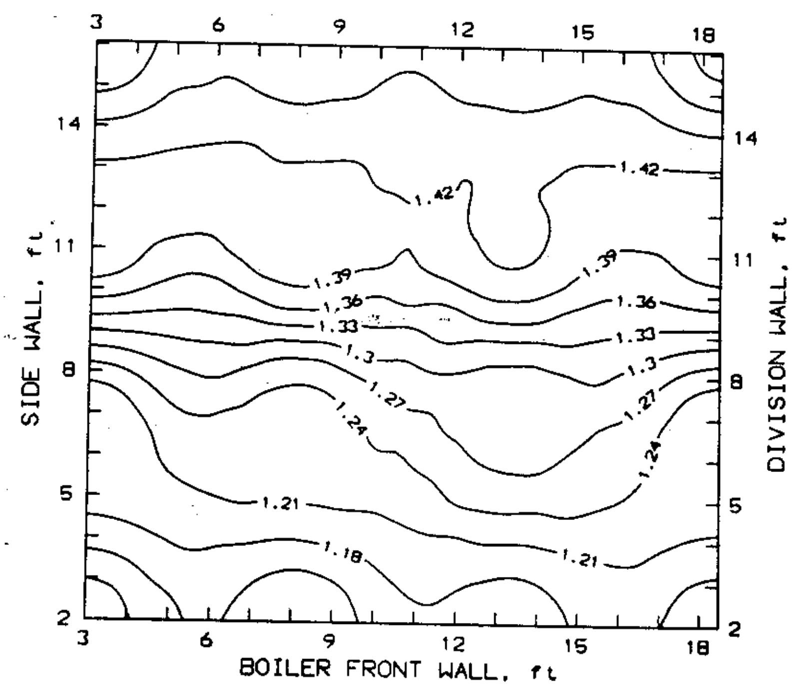


Figure 3-19. Six over fire air injectors operating at low load (86 MW) condition.

4.0 GAS REBURNING SYSTEM OPERATION

The gas reburning system is designed to operate over the typical load range of the Cherokee Unit 3 boiler. The load typically varies from 86 to 170 MW and is operated at 90 to 100% load from 7 a.m. to 8 p.m. and at 60 to 65% load during the night. The lower burner row is put out of service and flue gas is recycled through the hopper as the boiler load is dropped below 70% of maximum. Flue gas is recycled through the hopper at low load to increase the mass flow and maintain the steam quality of the boiler. The maximum flue gas recycled is 29% at 86 MW. For gas reburning conditions the primary burner zone stoichiometry varies from 1.1 to 1.2 as the boiler load decreases from 172 MW to 86 MW. Typical operating conditions utilized in the gas reburning design are presented in Table 4-1.

TABLE 4-1. GAS REBURNING OPERATING CONDITIONS

Load (MW)	Burner Zone Stoichiometry	Reburning Fuel Stoichiometry	Over Fire Air Stoichiometry	% Flue Gas Recycled to the Hopper
86	1.20	0.9	1.4	29
110	1.172	0.9	1.32	10
130	1.15	0.9	1.28	0
160	1.114	0.9	1.20	0
172	1.10	0.9	1.16	0

Natural gas flow rates are given in Figure 4-1 as a function of load. The increasing natural gas flow rate reflects the increasing overall heat input and increasing primary burner zone stoichiometry. The recycled flue gas to the reburning fuel jets increases with increasing MW generation, shown in Figure 4-2. The maximum nozzle pressure for the reburning fuel natural gas and FGR mixture is 3.83 "WC at 172 MW. The reburning fuel jet velocities are shown in Figure 4-3.

The mass flow of over fire air and percent excess oxygen exiting the furnace are plotted as functions of load in Figure 4-4. The maximum over fire air mass flow occurs at about 80% load which corresponds to a maximum nozzle pressure of 3.95 "WC. The maximum flow occurs at this load due to the relationship between burnout zone stoichiometry and the heat input. Figure 4-5 illustrates how jet velocity varies for the over fire air injectors. The design of the over fire air injectors allows for positioning of the nozzles between 0° and 20°. The isothermal model studies

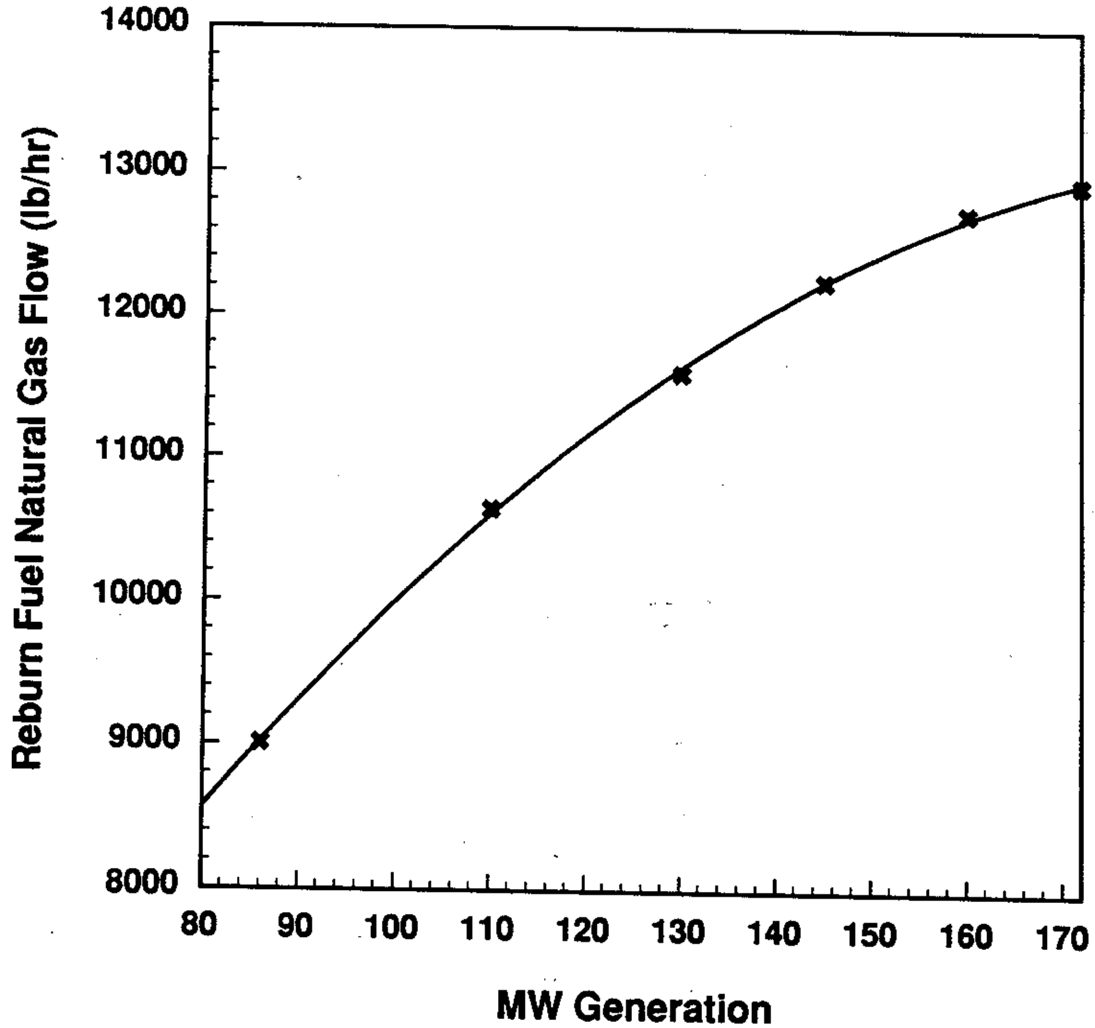


Figure 4-1. Natural gas mass flow to reburning fuel injectors.

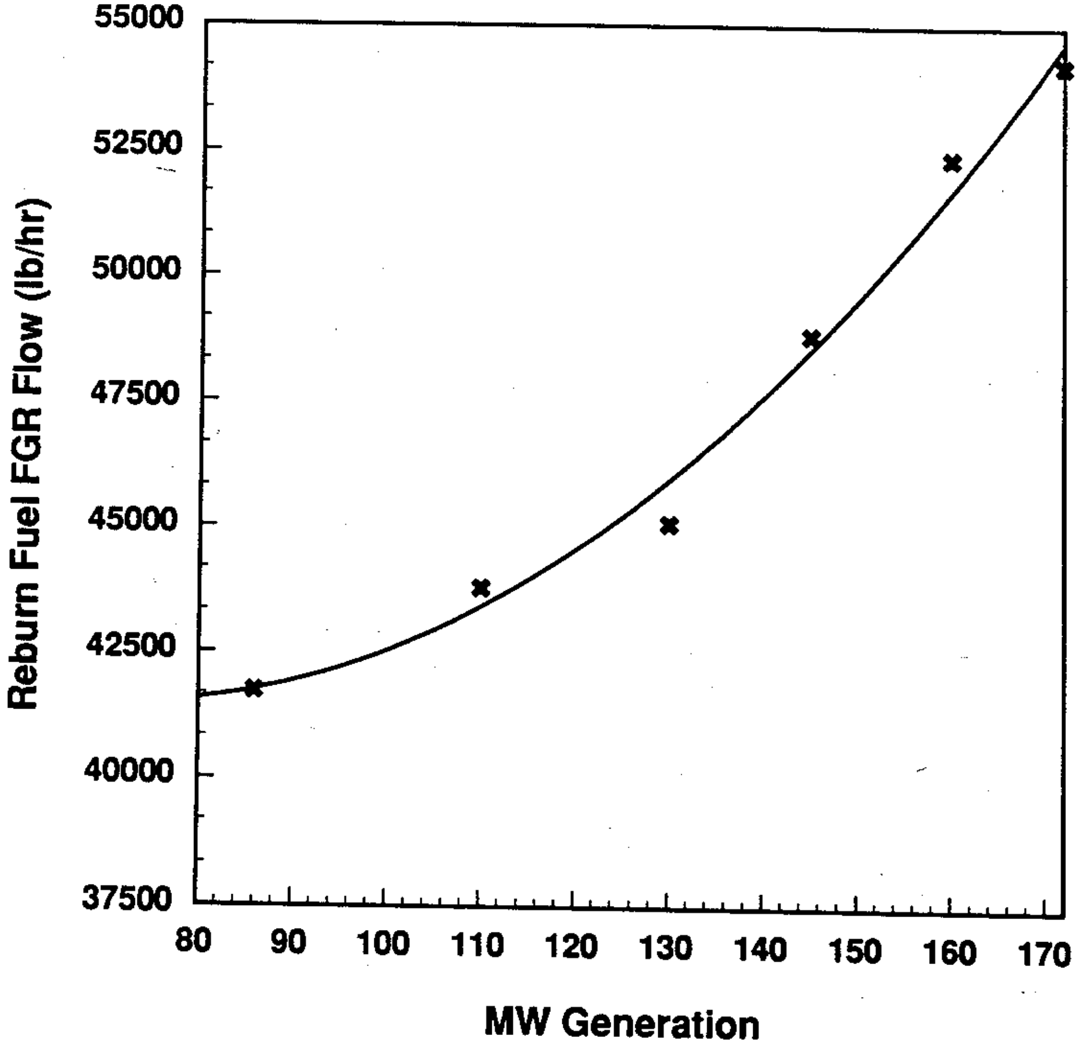


Figure 4-2. Recycled flue gas mass flow to reburning fuel injectors.

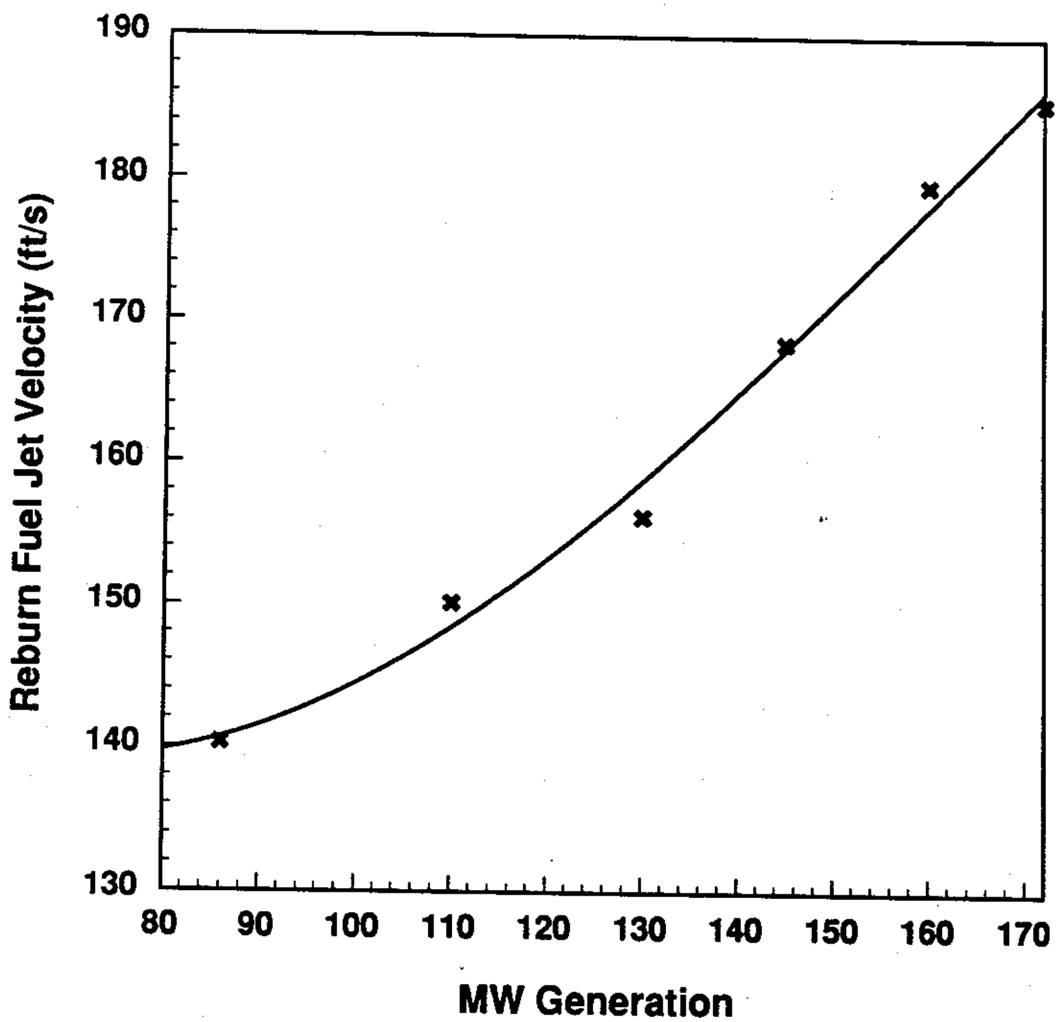


Figure 4-3. Reburning fuel injector velocity.

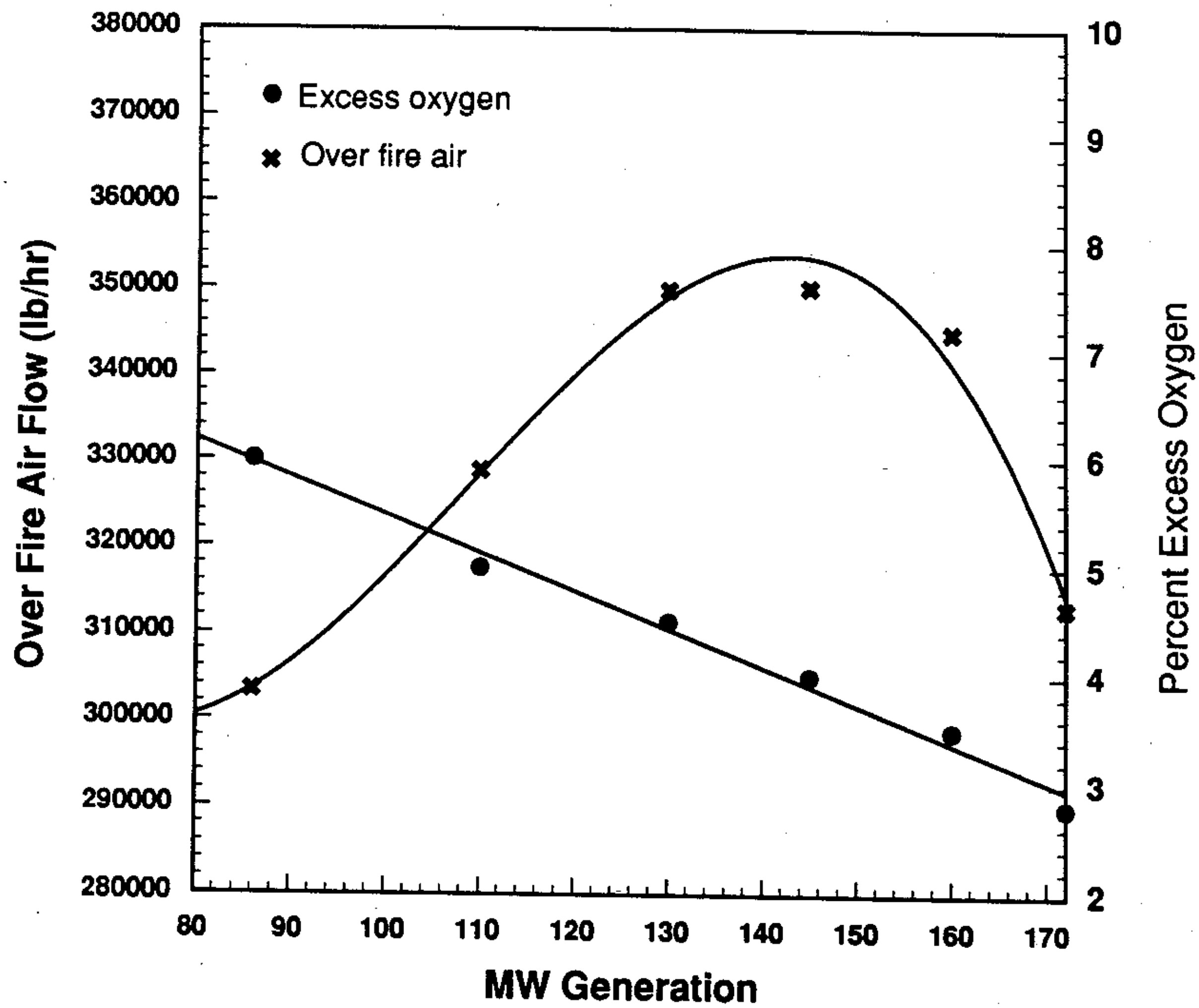


Figure 4-4. Over fire air mass flow and percent excess oxygen.

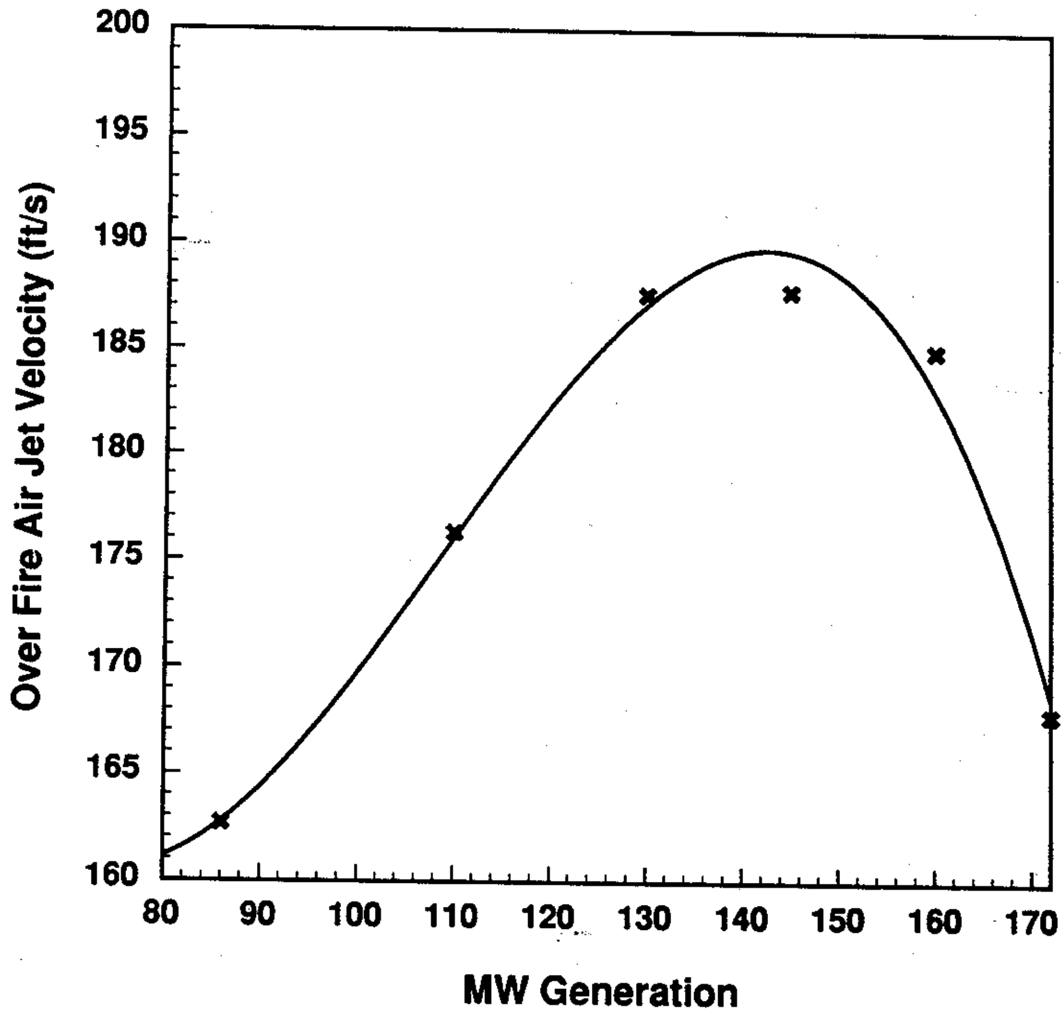


Figure 4-5. Over fire air injector velocity.

discussed above suggest that a 10° downward nozzle deflection will be adequate for good dispersion and coverage over the entire range of boiler loads.

5.0 GAS REBURNING PERFORMANCE

The reduction of NO_x by the gas reburning system was modeled using the One-Dimensional-Flame (ODF) code. The ODF code models the rate at which 43 species react in 201 reactions. The model was first run in a stirred tank reactor möde to obtain the initial gas phase concentrations (of 42 of the 43 species considered by the model) in the flame zone. These concentrations were then used as input for subsequent model runs of the gas reburning process. The concentration of NO_x exiting the flame zone was based on the specifications provided by the manufacturer of the low NO_x burners. The addition rates of natural gas and over fire air and concentrations of these streams were calculated by a spreadsheet which considered the carbon to hydrogen ratio of the coal, flue gas recirculation (FGR) into the bottom of the furnace, and FGR in the gas reburning jets. Temperature profiles were obtained from EER's heat transfer model.

Predicted levels of NO_x concentration are plotted in Figure 5-1 as a function of stoichiometric ratio in the reburning zone (SR_2) for five different loads ranging from a low load of 86 MW to a full load of 172 MW. (The initial concentrations were calculated based on the advertised performance of the low-NO_x burners being installed.) Levels of NO_x remain constant until SR_2 is increased to approximately one. At higher values of SR_2, the chemistry driving the gas reburning process slows and NO_x reductions cease to occur. The predictions illustrated in Figure 5-1 were made by assuming that the reburning and over fire jets evenly and uniformly mix with the furnaces gases. In reality, mixing will not occur evenly across a furnace. Pockets of gas both richer and leaner than the overall stoichiometry will pass along the length of the furnace. Model predictions which consider this are illustrated in Figure 5-2 for the full load case. Below SR_2's of 0.9 predicted NO_x levels remain the same. In the region of 0.95, NO_x levels are slightly higher than for the uniform mixing cases while at higher values of SR_2, uneven mixing results in reduced NO_x concentrations. Predicted concentrations of NO_x assuming uneven mixing are shown in Figure 5-3 as a function of load.

The emission rates of NO_x in terms of pounds of NO_x per hour are plotted in Figure 5-4. The relationship between the emission rate and load would be approximately linear if stoichiometric ratios were kept constant. As the low load (50%) is approached, however, the emission rate appears to level off. This results from a combination of the slight increase in NO_x concentration at low loads and an increase in the stoichiometric ratio of the burnout region (SR_3) from 1.16 at full load to 1.4 at low load. At low loads a proportionally higher volume of gas is produced at higher NO_x

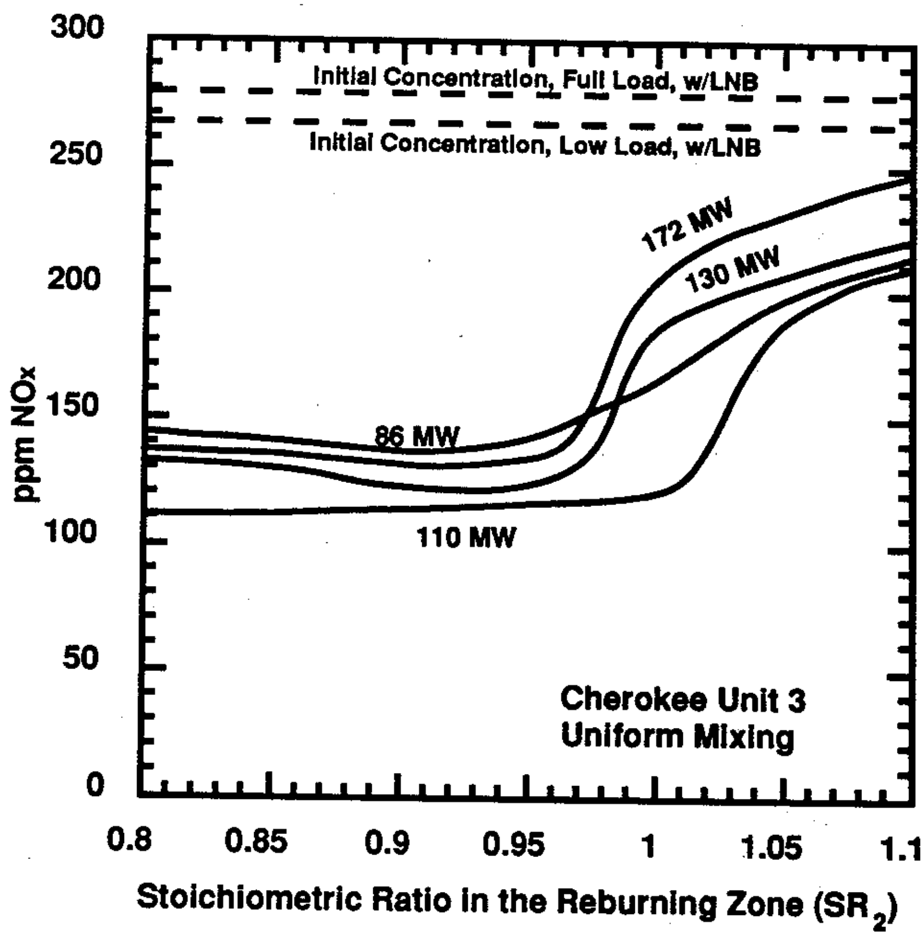


Figure 5-1. Predicted NOx concentrations as a function of stoichiometric ratio in the reburning zone.

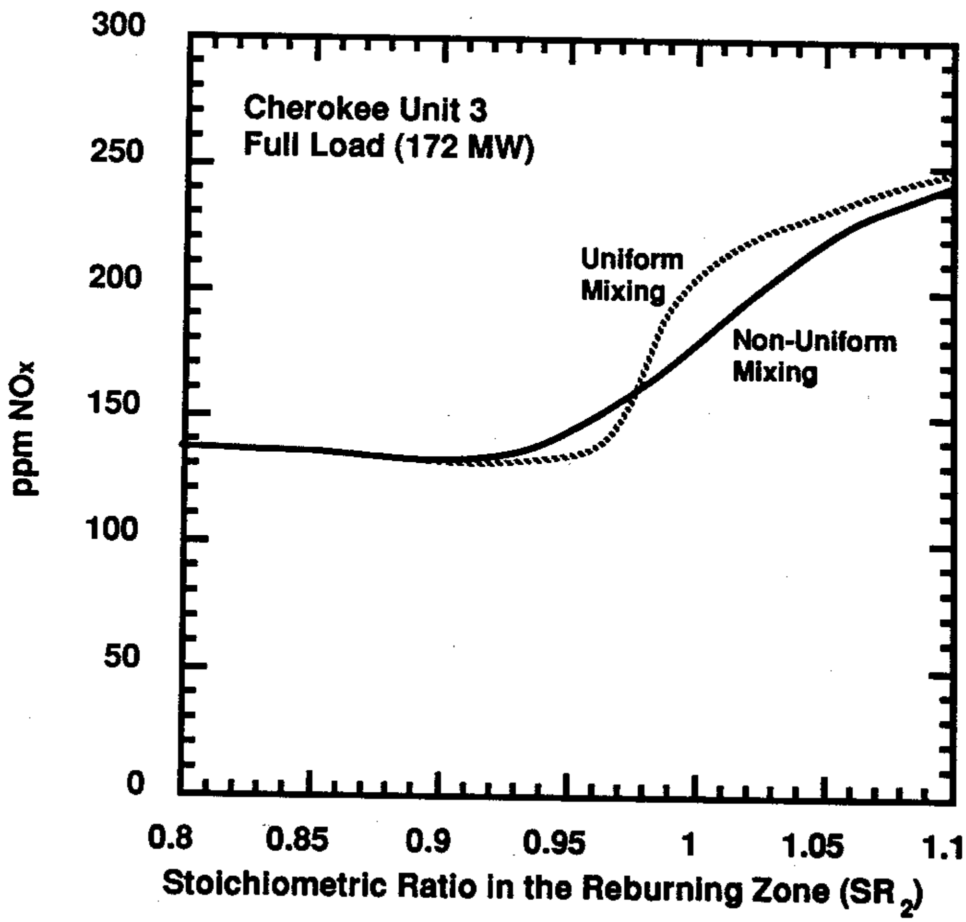


Figure 5-2. Predicted NOx concentrations when non-uniform mixing is considered.

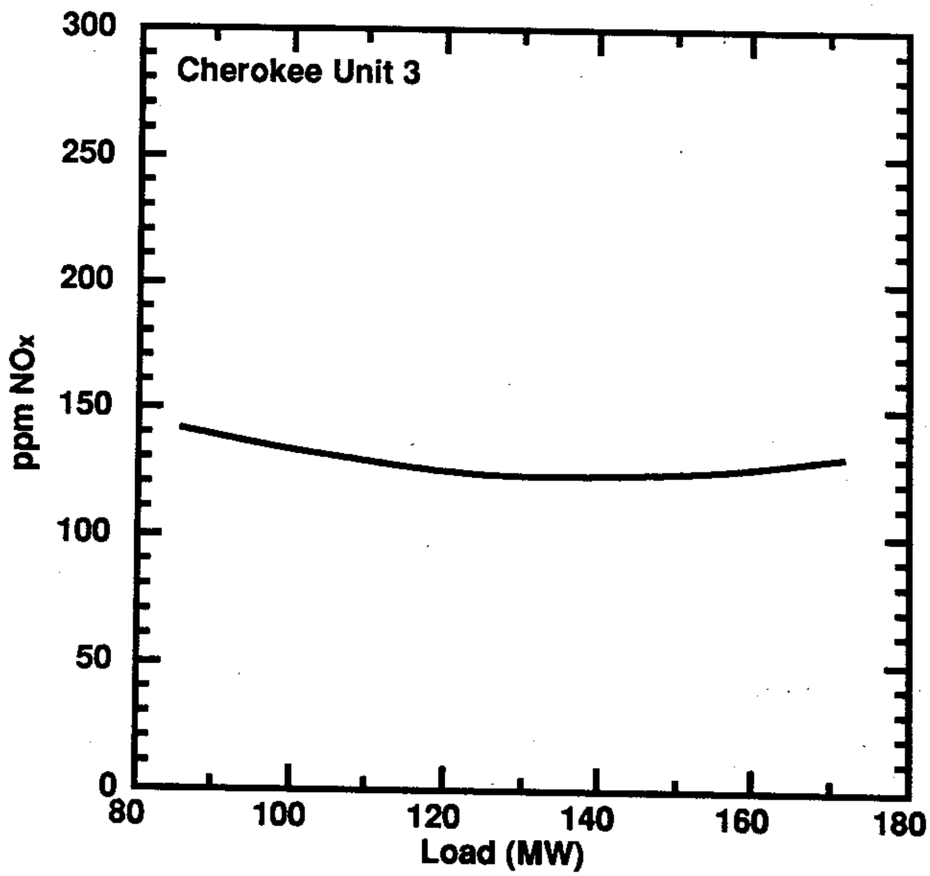


Figure 5-3. Predicted NOx concentration as a function of load when non-uniform mixing is considered.

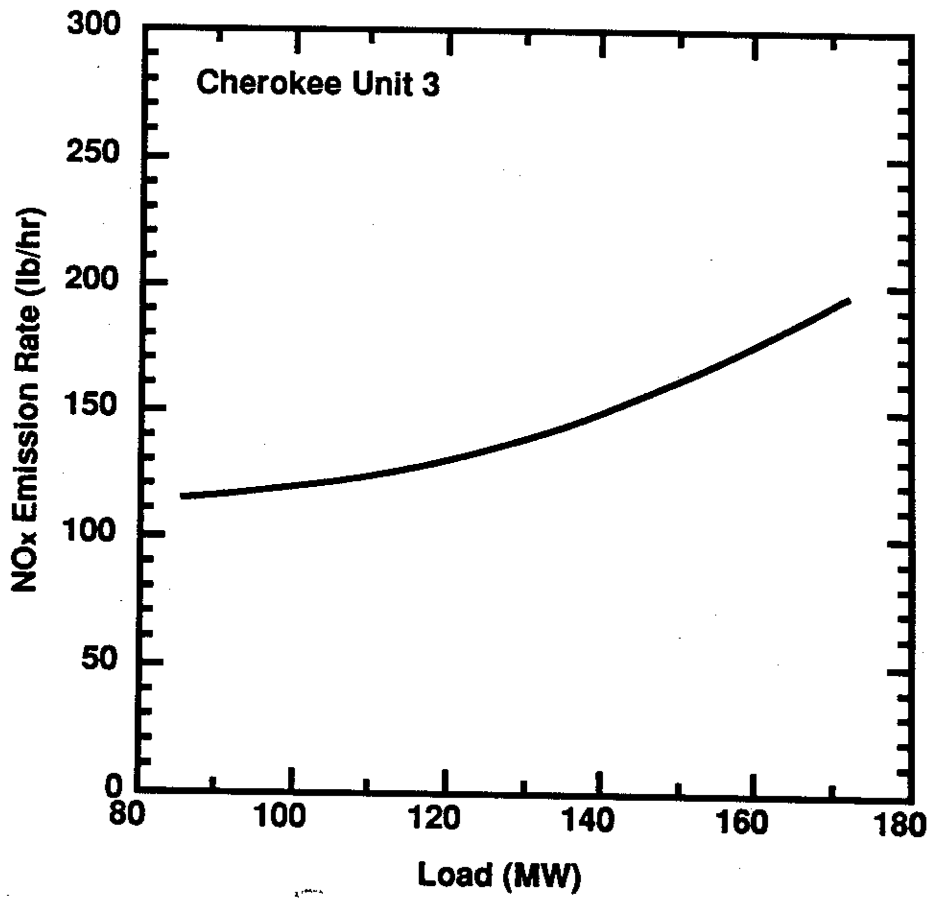


Figure 5-4. Predicted rate of NOx emissions.

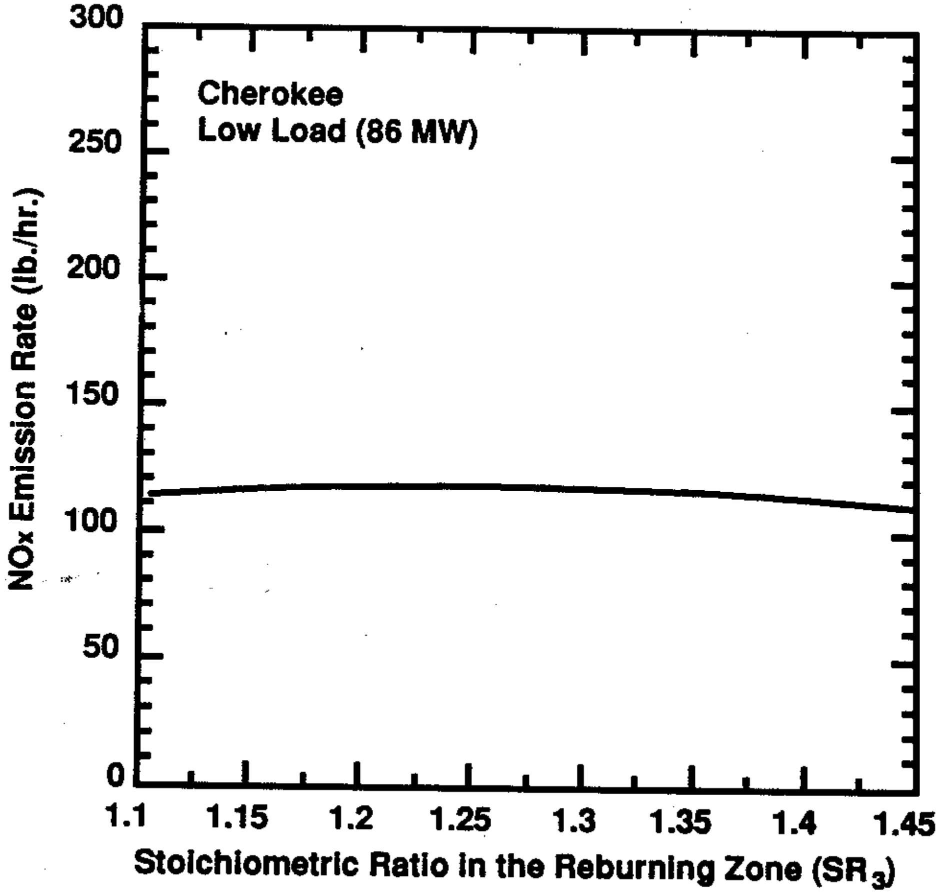


Figure 5-5. Predicted rate of NOx emissions at low load.

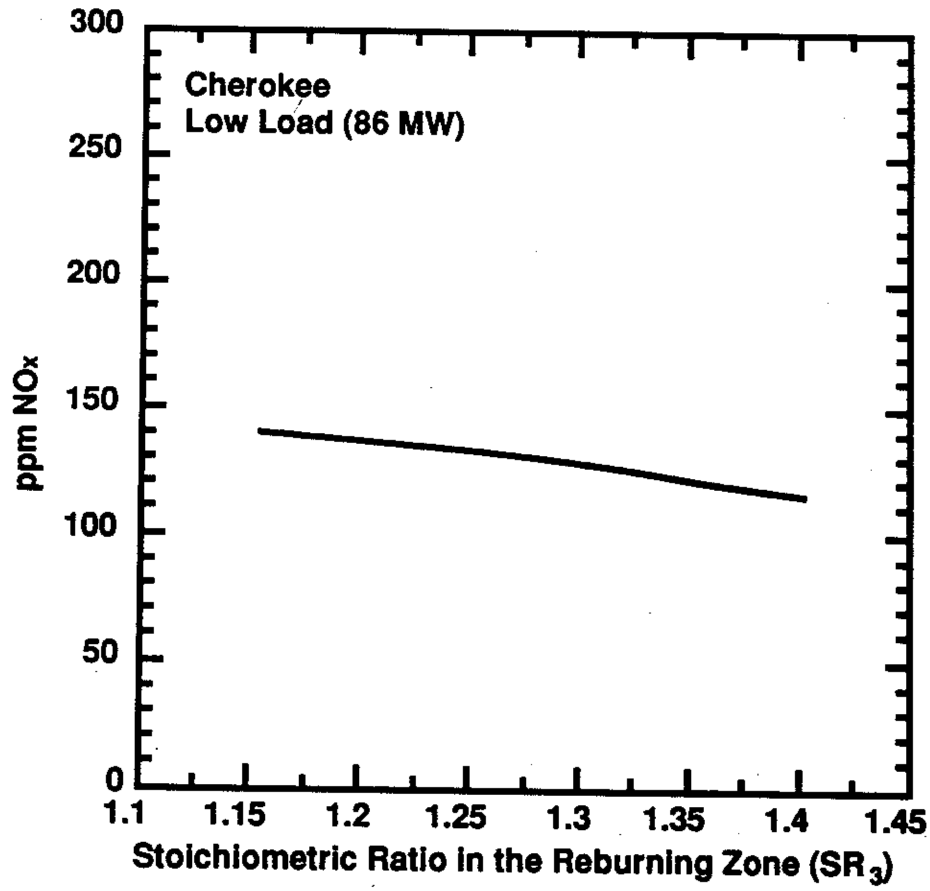


Figure 5-6. Predicted NOx concentrations at low load.

concentrations. Reducing the burnout zone stoichiometry will not reduce the NO_x emissions rate (Figure 5-5). The NO_x concentration, plotted as a function of SR_3 in Figure 5-6, increases as reducing the burnout zone stoichiometry concentrates the NO_x gases into a smaller volume of gas.

6.0 THERMAL PERFORMANCE ANALYSIS

Application of gas reburning (GR) and low NO_x burner (LNB) technologies may influence the thermal performance and operation of the Cherokee Unit 3 boiler. The use of numerical models allows prediction of boiler thermal characteristics as functions of various input and operating variables, and consequently provides necessary information for the selection of GR strategies. The primary objective of the computer modeling work in this program was to study the impacts of gas reburning and low NO_x burners and allow identification of possible remedies for any negative impacts which could prevent Cherokee Unit 3 from operating at full capacity and within the manufacturer's design limits (attemperation flow, tube metal temperatures,etc.).

A two-dimensional furnace heat transfer and combustion model (2D Code) was used in conjunction with a boiler performance model (Boiler Code) to predict the thermal performance impacts of GR and LNB application on Cherokee Unit 3. The following sections describe the modeling approach and then discuss the results of the model studies.

6.1 Approach

The geometric representation of Cherokee Unit 3 in the 2D code is illustrated in Figure 6-1. Figure 6-1(a) illustrates how the unit was divided into 22 layers in the direction of the gas flow while Figure 6-1(b) illustrates how the three-dimensional boiler is represented in the 2D code as an axisymmetric cylindrical grid. The length and radius of each section of the grid are chosen to approximately match both the volume and wall surface area of the corresponding section of the full-scale unit. The dimensions necessary to match volume and surface area result in a grid which is taller and narrower than the actual boiler. It is, however, considered more important to match volumes and surface areas between the model and the actual boiler in order to maintain heat transfer similarity.

The models were initially calibrated using field measurements to verify that they were properly simulating the performance of Cherokee Unit 3 for baseline coal-firing operating conditions at 170 MWe (full load). A series of simulations were then run to investigate the effects of LNB and the GR process. All low load cases were modeled by reducing total heat input to half of that at full load and taking the bottom burner row out of service. The effects of increasing burner swirl, introducing Flue Gas Recirculation (FGR) into hopper bottom, changing the primary zone stoichiometry (SR_1),

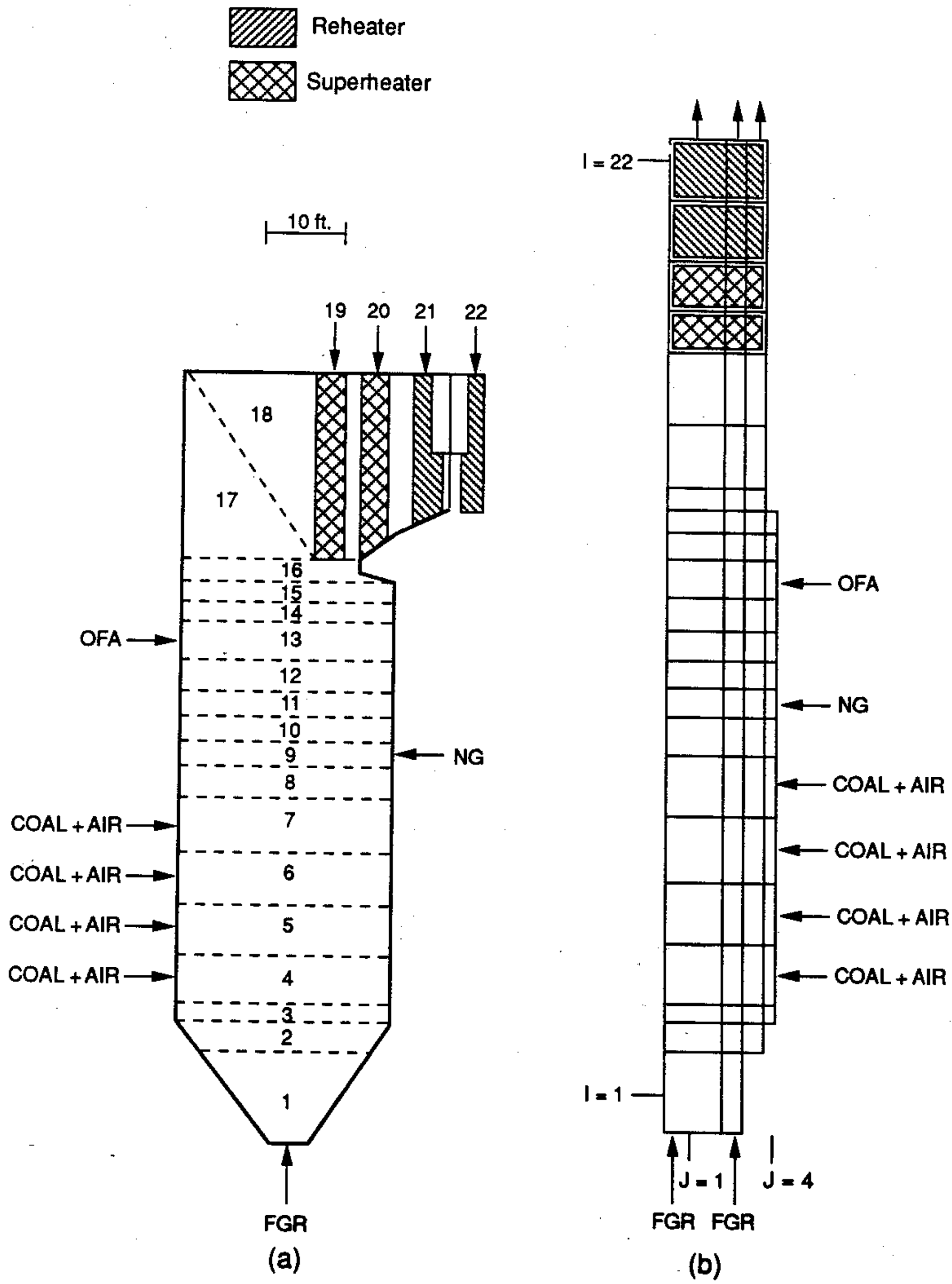


Figure 6-1. (a) Schematic of actual geometry, showing division into layers
(b) 2D cylindric grid for computation.

and varying overall exit stoichiometry (SR_3) on furnace thermal characteristics, boiler performance and boiler efficiency were also studied with or without the GR system.

Tables 6-1 and 6-2 summarize key parameters of all thermal modeling cases for Cherokee Unit 3. Burner swirl was simulated by reducing the recirculation strength above the top burner row upwards to the nose and increasing the recirculation strength below the bottom burner row downward to the hopper bottom. Delayed mixing was used to model LNB burnout characteristics of volatile matter. Natural gas was injected along with 3.4% FGR in the reburning zone. Mass flow rates of coal and natural gas for the GR cases were calculated, based on the Lower Heating Values(LHV) and total fuel heat inputs remaining at 210.7 and 105.4 MW for 100% and 50% loads respectively.

6.2 Full Load Cases

Verification of the model against the baseline case (Case 1) is discussed first followed by a discussion of the impacts of burner swirl and LNB with no GR. The impacts of the GR system on Cherokee Unit 3 are described in detail.

6.2.1 Model Verification at Full Load

Predicted mean gas temperatures in the furnace up to the entrance of the primary superheater section at full load are shown in Figure 6-2. The model verification was performed by comparing model predictions of boiler performance with measurements taken by EER's field crew. Table 6-3 summarizes a comparison of predicted boiler performance with measured data for the baseline calibration case. The predicted data basically agrees with the available measurements except that the Boiler Code predicts a slightly greater attemperation flow for the reheat steam cycle than measured. This is due to the increased amount of heat absorption in the reheater section predicted by the 2D code. The calibration of model parameters was considered adequate since the difference in the attemperation flow rate is considered insignificant.

6.2.2 Impacts of Burner Swirl and LNB at Full Load

Figures 6-3, 6-4, 6-5, and 6-6 show the impacts of the burner swirl and LNB on mean gas temperature distributions, net total heat flux densities, surface temperatures of ash deposits and

TABLE 6-1. SUMMARY OF KEY PARAMETERS OF THERMAL MODELING STUDIES FOR CHEROKEE UNIT 3.

Case No.	Thermal load (%)	Adjusting burner swirl	Low NOx burner	FGR from hopper bottom(%)	GR + 3.4% FGR	% of Heat Input from Coal/Natural Gas	Stoichiometry Ratio Burner	NG	Exit
1	100 (Baseline)	No	No	0	No	100.0/0.0	1.1661	N/A	1.1661
1a	100	Yes	No	0	No	100.0/0.0	1.1661	N/A	1.1661
1b	100	Yes	Yes	0	No	100.0/0.0	1.1661	N/A	1.1661
1c	100	Yes	Yes	0	Yes	82.20/17.80	1.1	0.9	1.1661
2	50 (Baseline)	Yes	Yes	0	No	100.0/0.0	1.3909	N/A	1.3909
2a	50	Yes	Yes	10	No	100.0/0.0	1.3909	N/A	1.3909
2b	50	Yes	Yes	20	No	100.0/0.0	1.3909	N/A	1.3909
2c	50	Yes	Yes	30	No	100.0/0.0	1.3909	N/A	1.3909
2d	50	Yes	Yes	30	Yes	90.23/9.77	1.0	0.9	1.3909
2e	50	Yes	Yes	30	Yes	82.20/17.80	1.1	0.9	1.3909
2f	50	Yes	Yes	30	Yes	75.49/24.51	1.2	0.9	1.3909
2g	50	Yes	Yes	30	Yes	75.49/24.51	1.2	0.9	1.2200

TABLE 6-2. SUMMARY OF MASS FLOW RATES OF THERMAL MODELING STUDIES FOR CHEROKEE UNIT 3.

Case No.	Case Definition	Burner rows in service	Fuel flow(kg/s)		FGR(kg/s)		Air flow(kg/s)		Total mass input (kg/s)
			Coal	Natural Gas	Hopper bottom	Reburning zone	Coal burner	OFA	
1	100%, Baseline	All 4 rows	18.61	0	0	0	165.66	0	184.27
1a	100%, Burner Swirl	All 4 rows	18.61	0	0	0	165.66	0	184.27
1b	100%, Low NO_x Burner	All 4 rows	18.61	0	0	0	165.66	0	184.27
1c	100%, GR	All 4 rows	15.30	1.81	0	6.24	128.46	37.98	189.78
2	50%, Baseline	Top 3 rows	9.30	0	0	0	98.80	0	108.11
2a	50%, 10% FGR	Top 3 rows	9.30	0	10.81	0	98.80	0	118.92
2b	50%, 20% FGR	Top 3 rows	9.30	0	21.62	0	98.80	0	129.73
2c	50%, 30% FGR	Top 3 rows	9.30	0	32.43	0	98.80	0	140.54
2d	50%, SR_1=1.0, SR_3=1.3909	Top 3 rows	8.40	0.50	32.38	3.67	64.09	34.96	144.00
2e	50%, SR_1=1.1, SR_3=1.3909	Top 3 rows	7.65	0.90	32.34	3.67	64.23	35.03	143.82
2f	50%, SR_1=1.2, SR_3=1.3909	Top 3 rows	7.02	1.24	32.31	3.66	64.34	35.10	143.67
2g	50%, SR_1=1.2, SR_3=1.2200	Top 3 rows	7.02	1.24	28.65	3.25	64.34	22.88	127.38

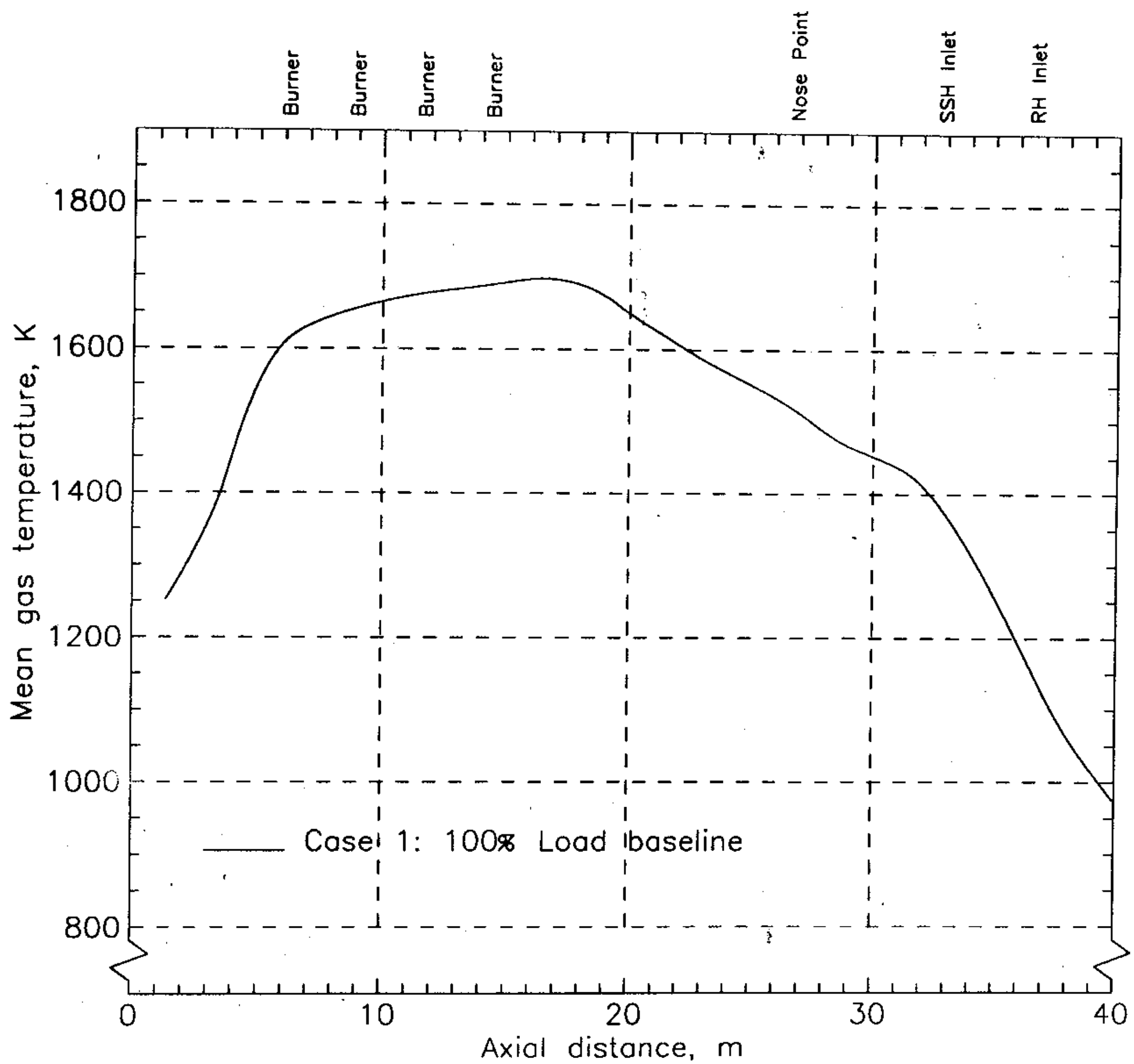


Figure 6–2. Predicted mean gas temperatures at 100% load baseline.

TABLE 6-3. COMPARISON OF PREDICTED BOILER PERFORMANCE WITH MEASURED ONE AT 100% LOAD BASELINE.

Case Number	#	Case 1
Case Definition	Measured Data	Baseline Prediction
Exit Gas Temperatures(K) of		
RH(rear)	-	974
PSH	-	672
Economizer	-	649
Air Preheater	-	429
Steam Flow(kg/s)		
Main Steam	142.76	142.33
RH Steam	116.72	116.71
Attemperation flow(kg/s)		
SH	1.36	1.354
RH	0.28	0.341
Water/steam temperatures(K)		
Economizer Inlet	526	526
Economizer Outlet	-	533
PSH Inlet	-	609
PSH Outlet	-	696
SSH Attemp. Outlet	690	690
SSH Outlet	803	803
RH Attemp. Outlet	615	613
RH Outlet	812	812
Heat absorptions(kW)		
Furnace	-	212756
Economizer	-	4846
PSH	67106	66684
SSH	45850	45925
RH	51399	52080
Unburned Fixed Carbon (% of Total Fixed Carbon Input)	-	0.73

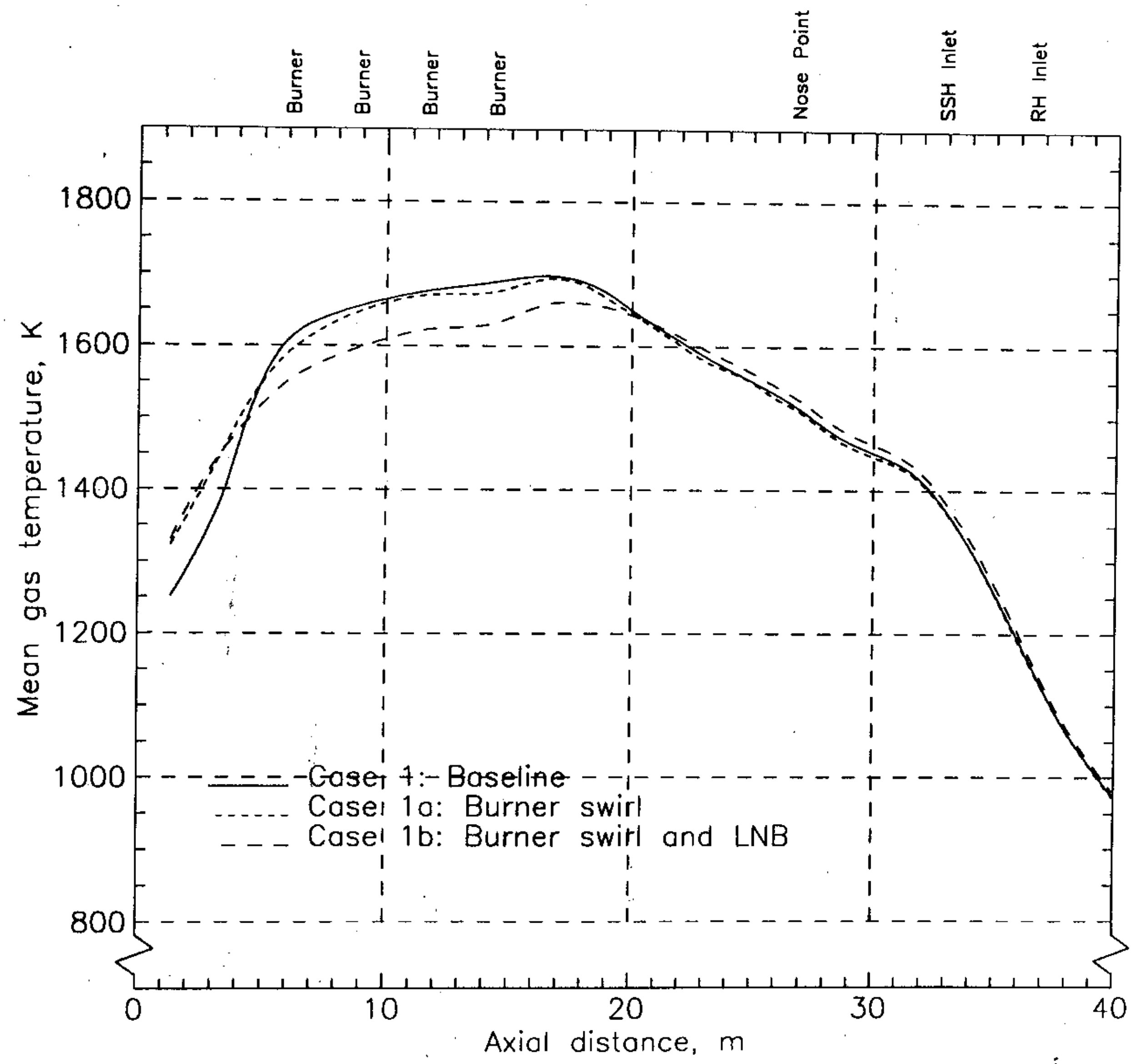


Figure 6–3. Impacts of burner swirl and LNB on mean gas temperature distributions at 100% load without GR.

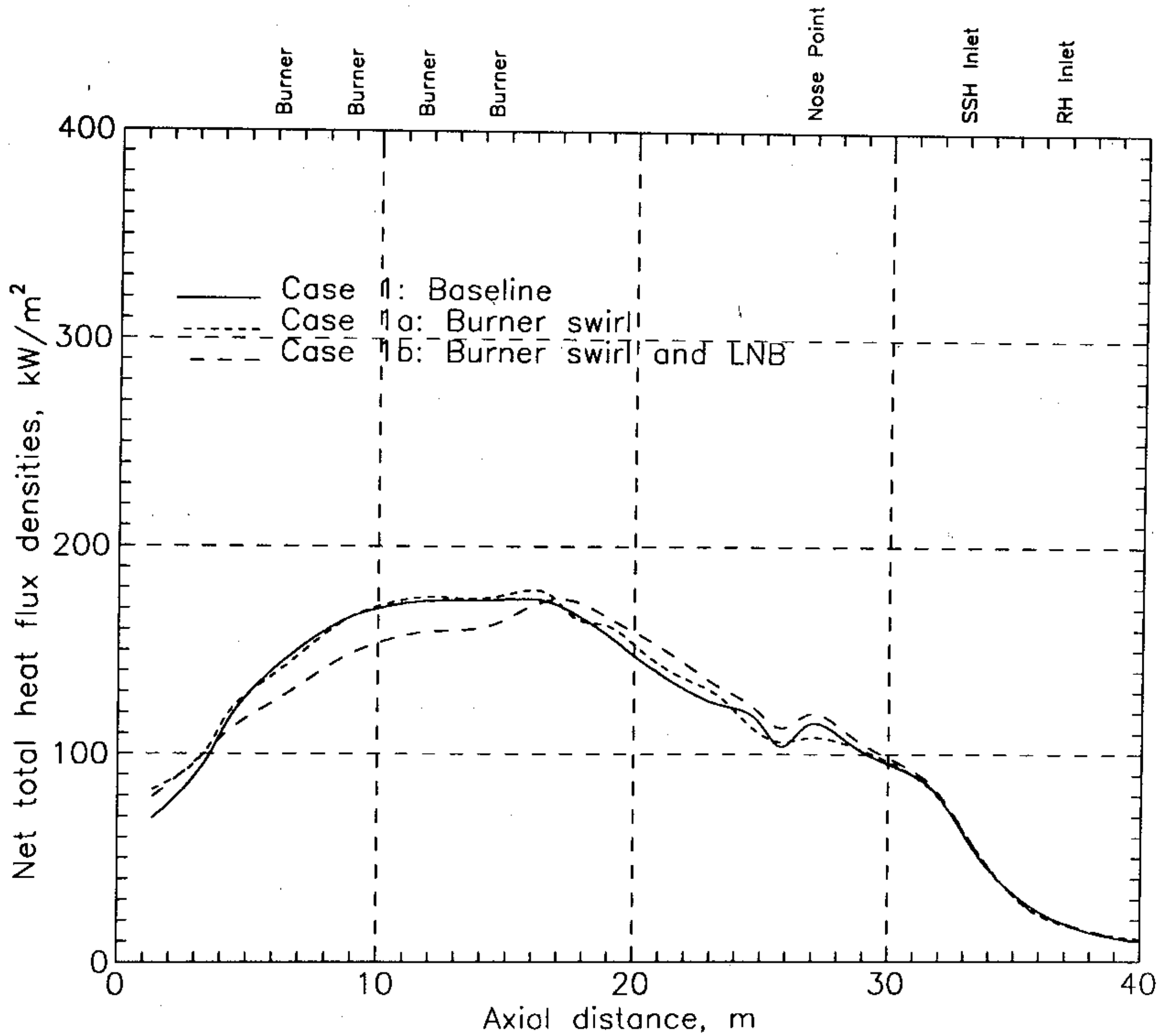


Figure 6–4. Impacts of burner swirl and LNB on net total heat flux densities at 100% load without GR.

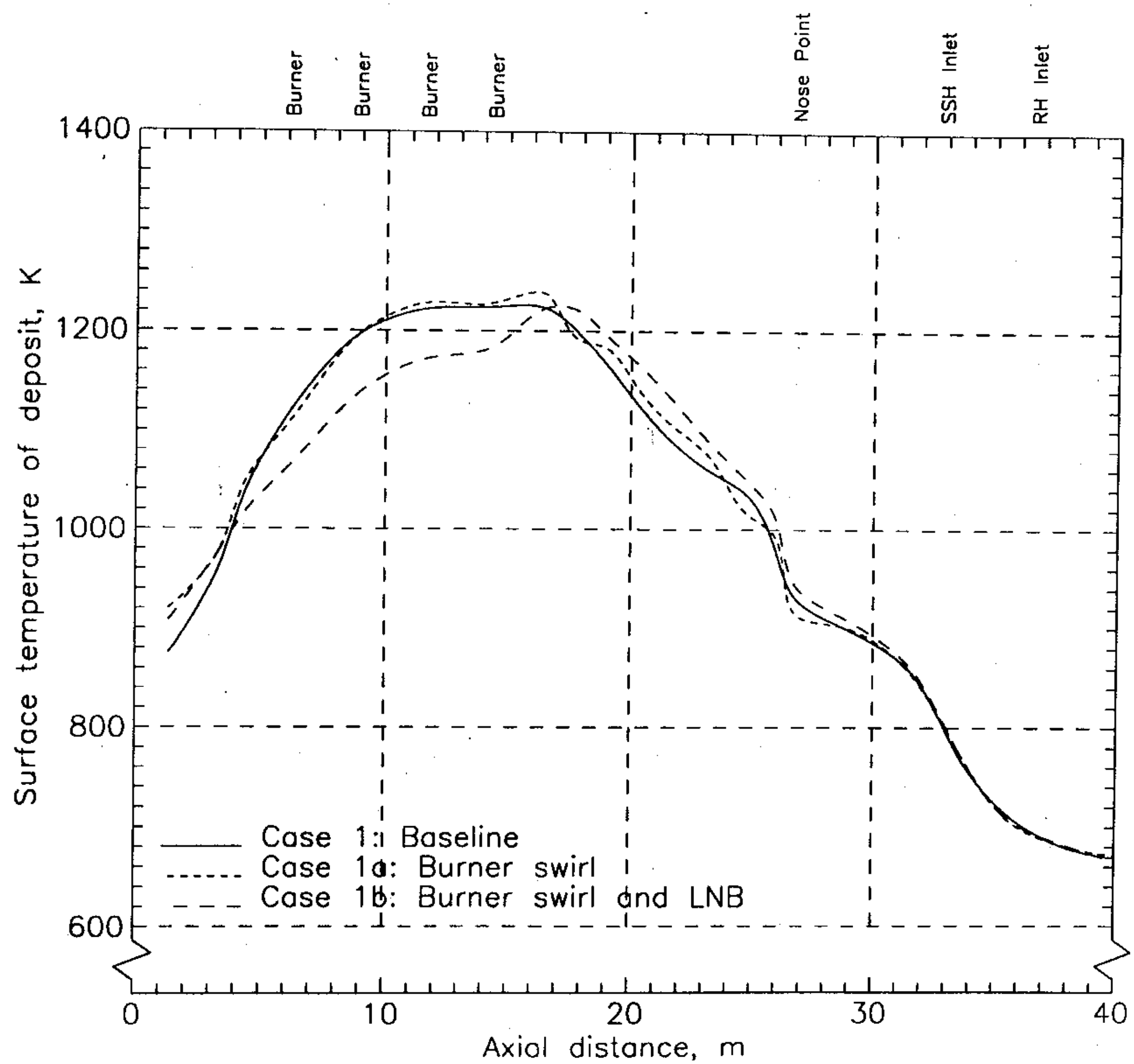


Figure 6–5. Impacts of burner swirl and LNB on surface temperatures of deposit at 100% load without GR.

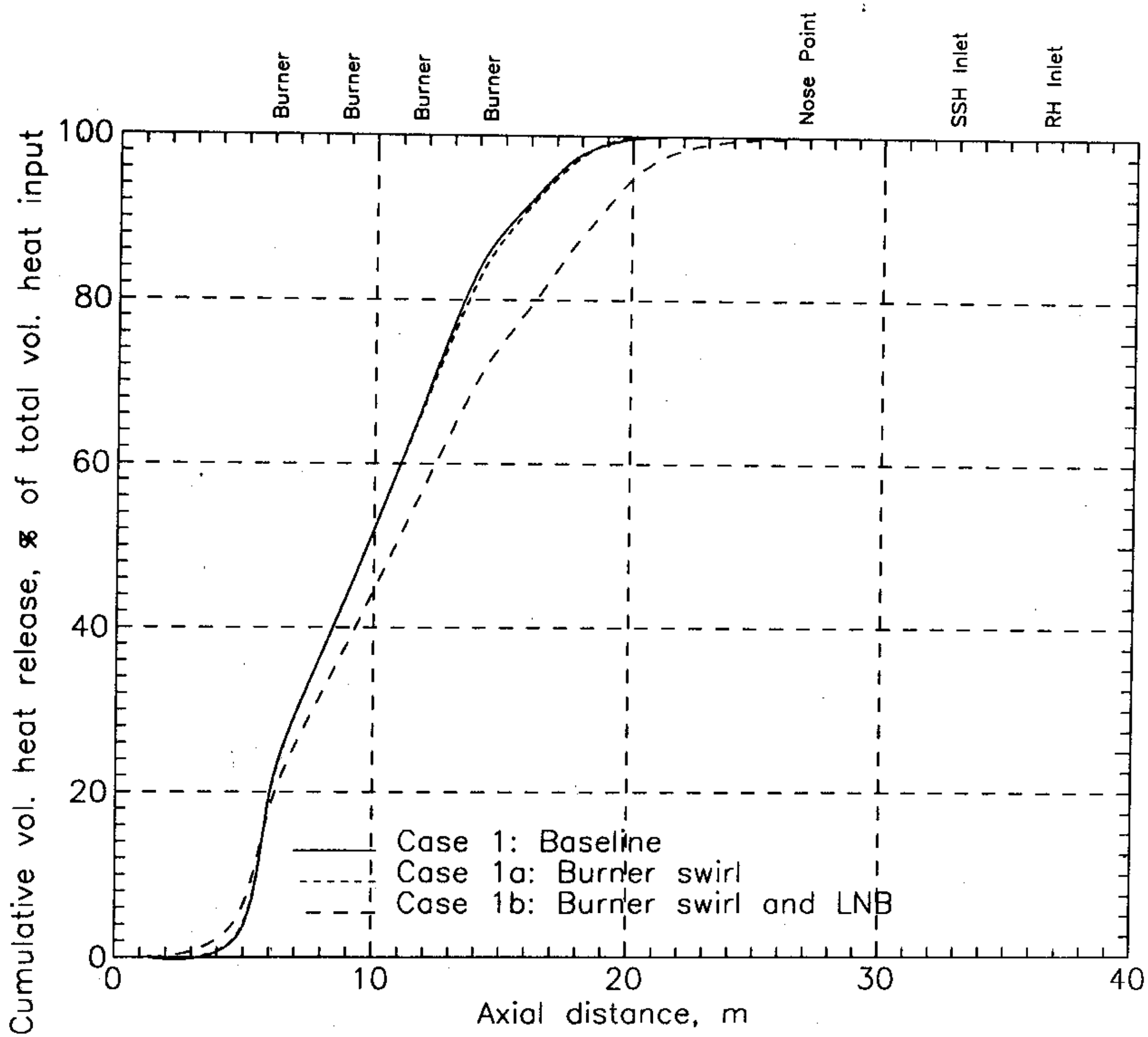


Figure 6–6. Impacts of burner swirl and LNB on volatile heat release in a percentage of total volatile heat input at 100% load without GR.

cumulative volatile heat releases as percentages of total volatile heat input. Adjusting burner swirl vanes does not have any significant impact on furnace thermal characteristics except that temperatures and heat fluxes in the hopper zones increase. This increase is caused by an increase in recirculation below the bottom burner row which recirculates more hot gas from the burner zones down into the hopper zones. Firing with LNB decreases the gas temperatures in the burner zones, shifts the heat absorption and maximum surface temperature upwards in the furnace, and also prolongs the completion of volatile heat release. These effects are due to the delay in volatile heat release caused by the LNB.

The impacts of burner swirl and LNB on boiler performance are displayed in Table 6-4. Increasing the burner swirl shifts heat absorption to the lower furnace which increases steam generation and reduces attemperation flow rates for both main and reheat steam cycles. Operating the unit with LNB in service delays fuel heat release. More heat, therefore, becomes available in the upper furnace and higher heat absorptions occur there. The upwards shift of heat absorption makes the main and reheat attemperation flow rates higher than those without LNB in operation. A small improvement in carbon burnout is caused primarily by the higher upper furnace gas temperatures with LNB in service (Case 1b) than without LNB in operation (Case 1c).

Table 6-5 lists boiler efficiency calculation for Cases 1, 1a and 1b, based on the ASME heat loss method, as described in "ASME Test Form for Abbreviated Test" (PTC 4.1-a and 4.1-b). Heat exchange through the air heater is considered. The method considers six categories of heat losses: heat loss due to dry gas, moisture in fuel, water vapor generated by combustion of hydrogen in a fuel, combustible matter in refuse, radiation and unmeasured. Water vapor in the flue gas is formed by both vaporization of moisture and combustion of hydrogen in the fuel. As long as the flue gas leaving the boiler carries the moisture in the vapor form, the latent heat of flue gas moisture becomes a loss to the unit. Heat losses are calculated based on 2D and Boiler Code input data and output results except that radiation and unmeasured heat losses are taken from the B&W boiler design data sheet. The boiler efficiency with LNB in operation is slightly greater than with LNB out of service due to slightly lower levels of unburned fixed carbon.

Flue gas residence time in the furnace plays an important role in the selection of GR locations. Sufficient time must be allowed for the coal to burn, and to ensure that the natural gas is added at the optimum location for controlling NO_x emissions. Figure 6-7 shows the temperature profiles versus residence time for Cases 1, 1a and 1b.

TABLE 6-4. IMPACTS OF BURNER SWIRL AND LNB ON BOILER PERFORMANCE FOR 100% LOAD COAL FIRING.

Case Number	Case 1	Case 1a	Case 1b
Case Definition	Baseline Prediction	Burner Swirl Only	Burner Swirl and Low NOx
Exit Gas Temperatures(K) of			
RH(rear)	974	972	980
PSH	672	671	674
Economizer	649	649	651
Air Preheater	429	429	430
Steam Flow(kg/s)			
Main Steam	142.33	143.49	142.66
RH Steam	116.71	117.40	117.36
Attemperation flow(kg/s)			
SH	1.354	0.611	2.779
RH	0.341	0.088	0.721
Water/steam temperatures(K)			
Economizer Inlet	526	526	526
Economizer Outlet	533	533	533
PSH Inlet	609	609	609
PSH Outlet	696	693	699
SSH Attemp. Outlet	690	690	687
SSH Outlet	803	803	803
RH Attemp. Outlet	613	615	610
RH Outlet	812	812	812
Heat absorptions(kW)			
Furnace	212756	215724	211028
Economizer	4846	4817	4895
PSH	66684	66307	67737
SSH	45925	46056	47333
RH	52080	51878	53131
Unburned Fixed Carbon (% of Total Fixed Carbon Input)	0.73	0.73	0.63

TABLE 6-5. IMPACTS OF BURNER SWIRL OR LNB ON BOILER EFFICIENCY, BASED ON ASME HEAT LOSS METHOD FOR 100% LOAD COAL FIRING.

Case Number	Case 1	Case 1a	Case 1b
Case Definition	Baseline Prediction	Burner Swirl Only	Burner Swirl and Low NOx
Heat Loss due to Dry Gas	5.1068	5.0920	5.1238
Heat Loss due to Moisture in Fuel	1.6947	1.6943	1.6952
Heat Loss due to H_2O from Combustion of H_2	4.1521	4.1513	4.1536
Heat Loss due to Combustible in Refuse	0.4446	0.4446	0.3837
Heat Loss due to Radiation*	0.2200	0.2200	0.2200
Unmeasured Losses*	1.5000	1.5000	1.5000
Total Heat Losses	13.1181	13.1021	13.0763
Boiler Efficiency	86.8819	86.8979	86.9237

* Use the value reported in the boiler design performance data sheet.

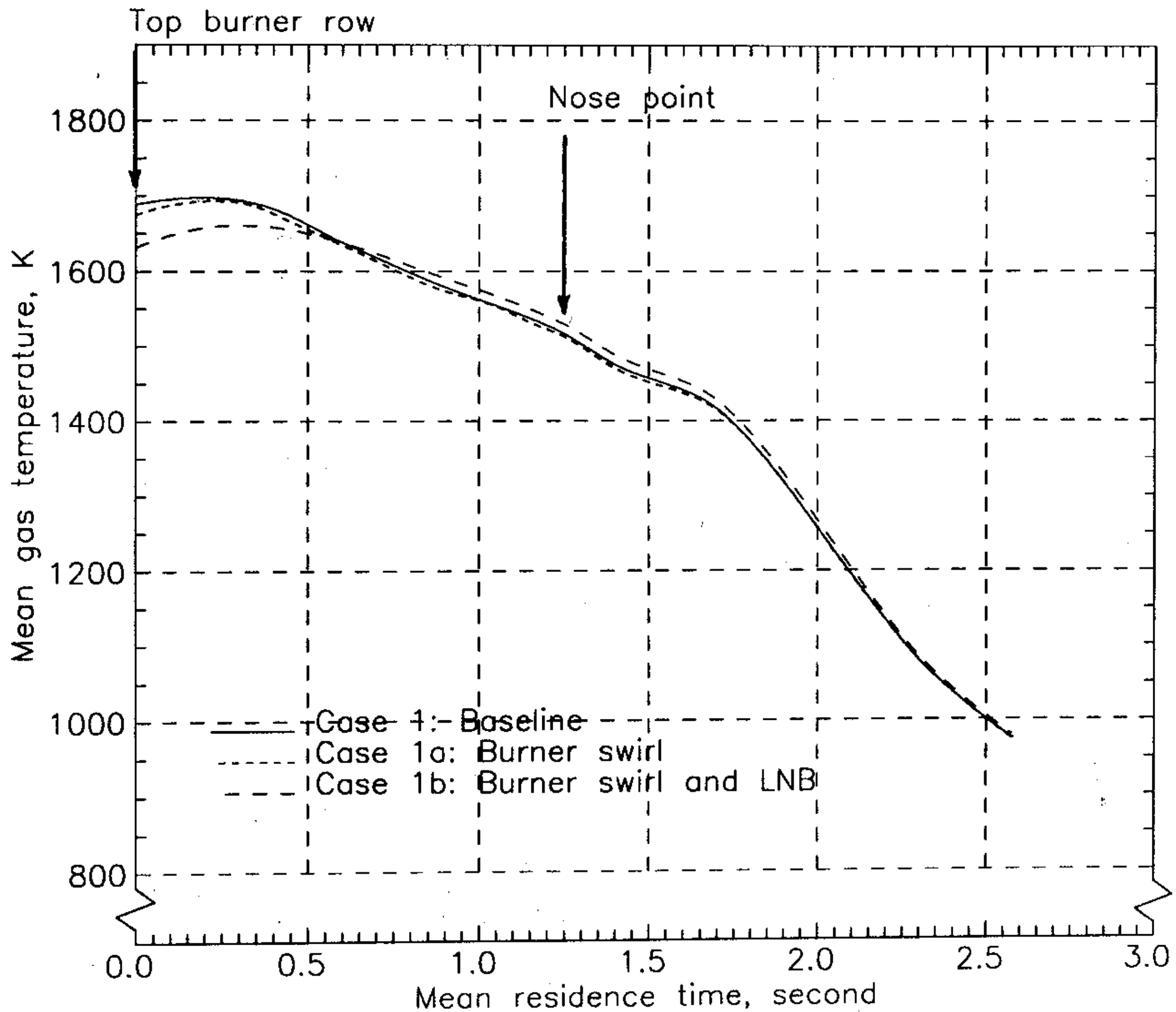


Figure 6–7. Time–temperature plots for Cases 1, 1a and 1b.

6.2.3 Impacts of GR at Full Load

Figures 6-8, 6-9, 6-10 and 6-11 plot the impacts of GR at full load on furnace mean gas temperature distributions, net total heat flux densities, surface temperatures of ash deposits and cumulative fuel heat release as functions of the 2D grid axial distance(Cases 1b and 1c). Most of the profile variations between GR and non-GR cases in the region of the over-fire air and burner zones are due to mass flow differences at these locations. Reburning zone(SR_2) and overfire air(SR_3) stoichiometries are maintained at constant values of 0.9 and 1.1661 respecively for the GR case.

The impacts of GR on the mean gas temperature profiles are shown in Figure 6-8. The lower heat input in the burner region for the GR case results in lower gas temperatures there. There is also a drop in temperature at the over fire air ports for GR case.

The effect of GR on the net total heat fluxes to the furnace wall is shown in Figure 6-9. Lower heat fluxes in the burner zones for the GR case are due to the lower heat input in the burner zones. Introducing over fire air for the GR case drops the local gas temperatures and consequently decreases heat absorption on the furnace wall. The same effect is illustrated in Figure 6-10 for the surface temperatures of ash deposits. The impact of GR on the cumulative heat release distributions as a percent of total chemical heat input is also displayed in Figure 6-11. Less heat is released below the natural gas injection ports due to the reduced thermal heat input to the burner zones for the gas reburning case. The heat release above the natural gas ports do not reach completion until the over-fire air level at which point sufficient air is provided to allow combustion to be completed.

The impacts of GR on boiler performance at full load are summarized in Table 6-6. Introduction of the GR system increases both main and reheat attemperation flows by about two times and decreases steam flow rates by less than one percent. This indicates that the heat absorption pattern is modified such that more heat is absorbed in the reheater and superheater sections and less heat is absorbed in the radiant furnace.

Fixed carbon in the coal, for GR application, is first burned in the fuel-lean, high-temperature burner zones where favorable conditions for the combustion of fixed carbon exist. The fixed car-

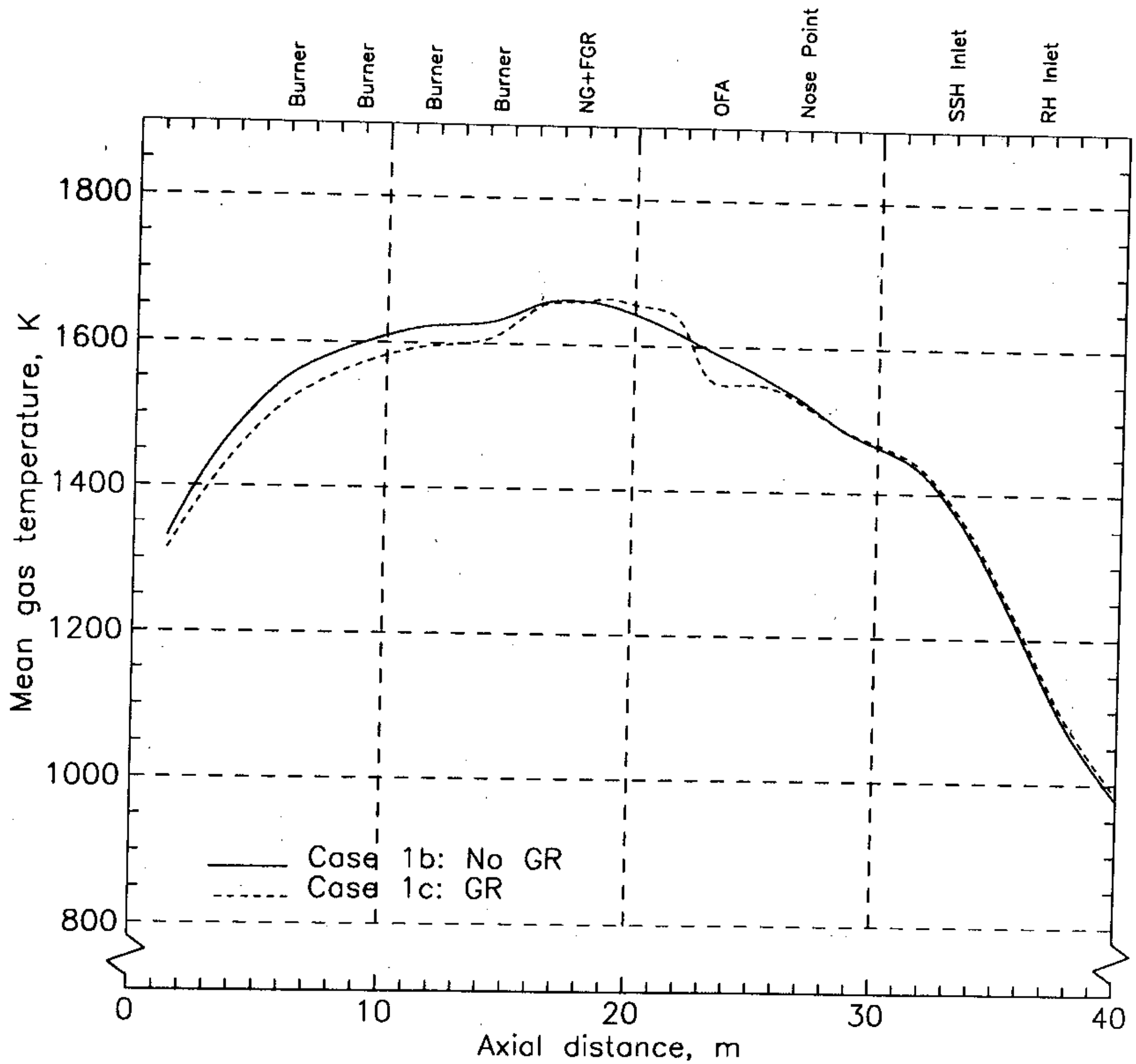


Figure 6–8. Impacts of GR on mean gas temperature distribution at 100% load.

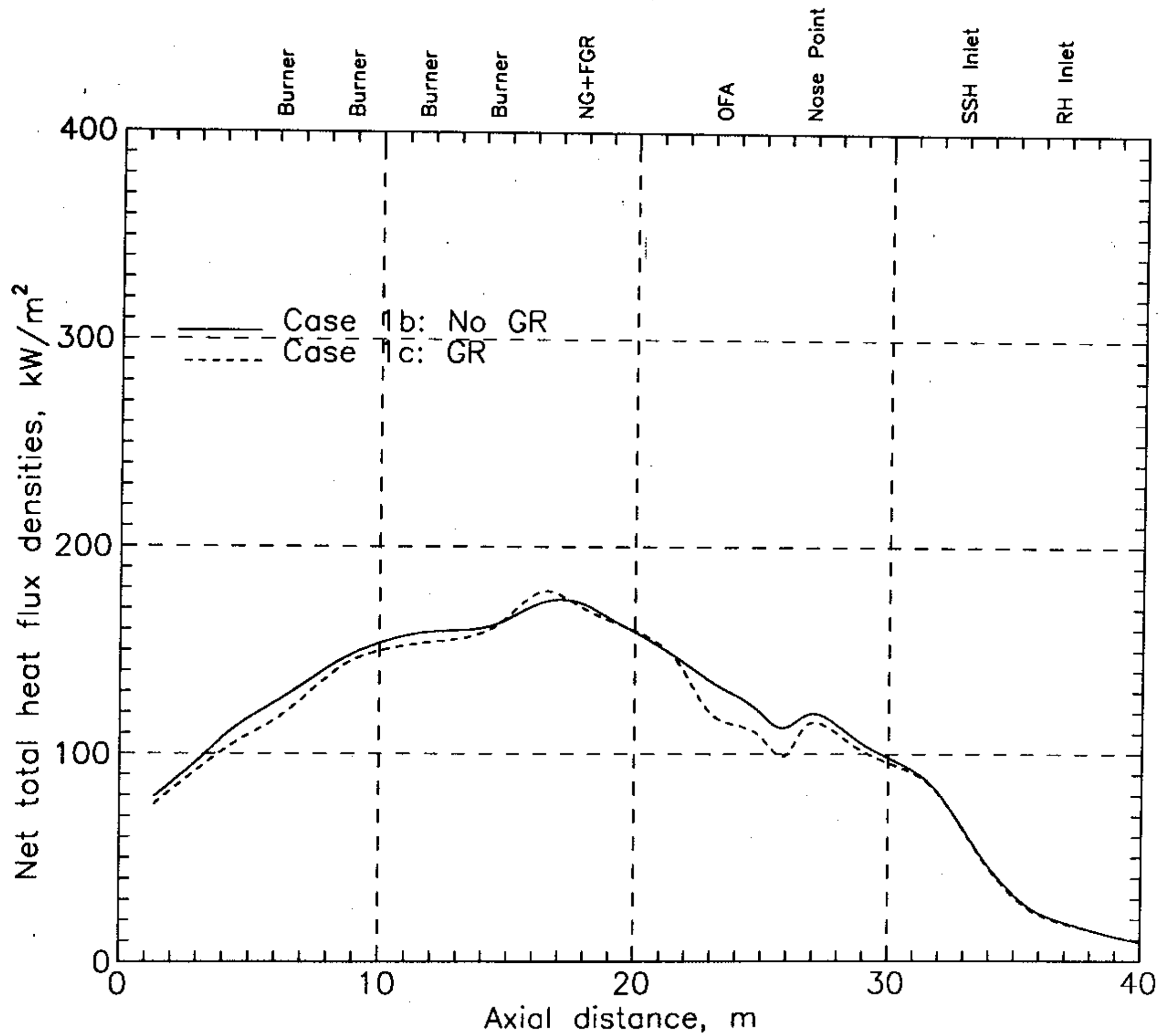


Figure 6–9. Impacts of GR on net total heat flux densities at 100% load.

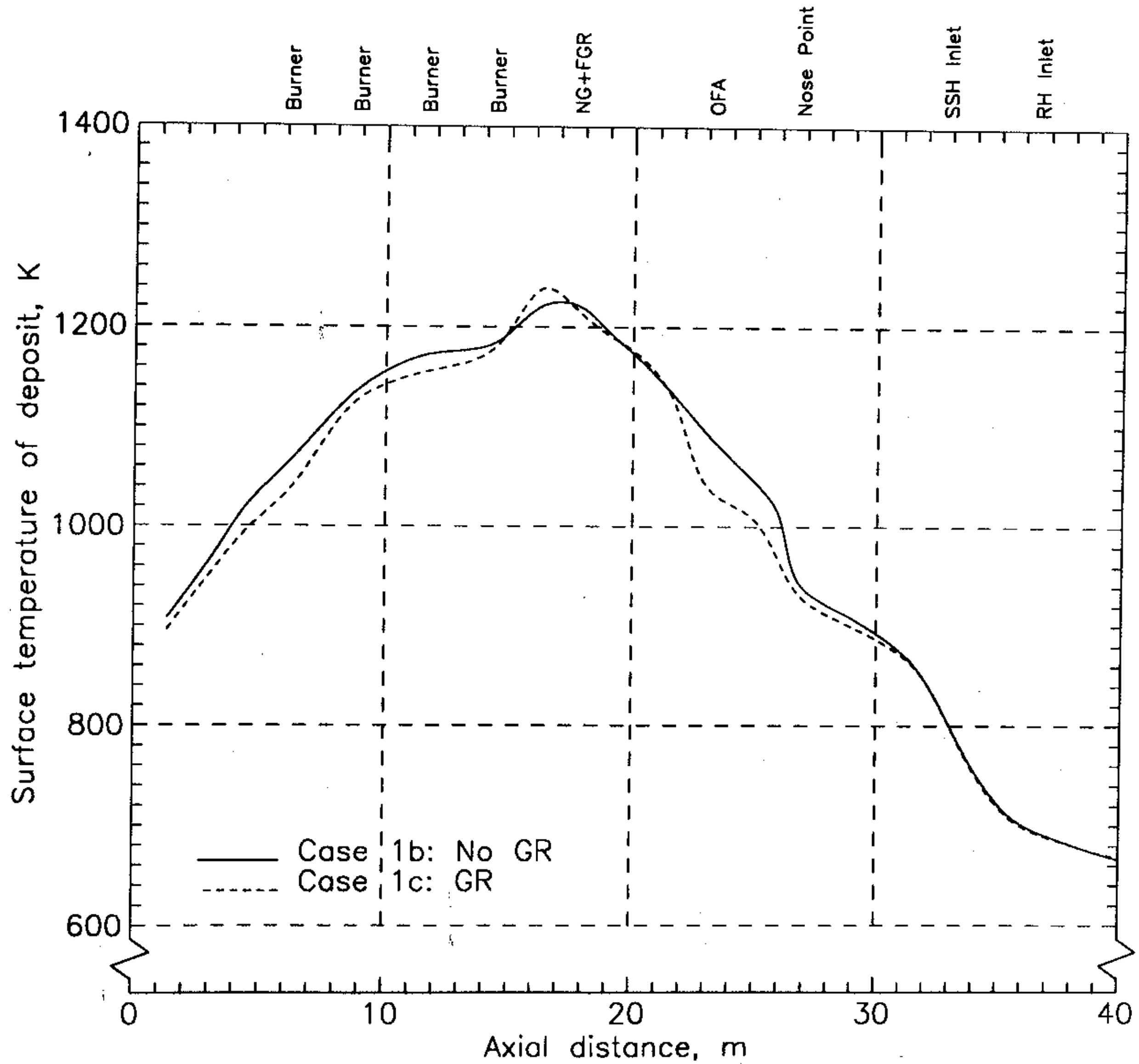


Figure 6–10. Impacts of GR on surface temperatures of deposit at 100% load.

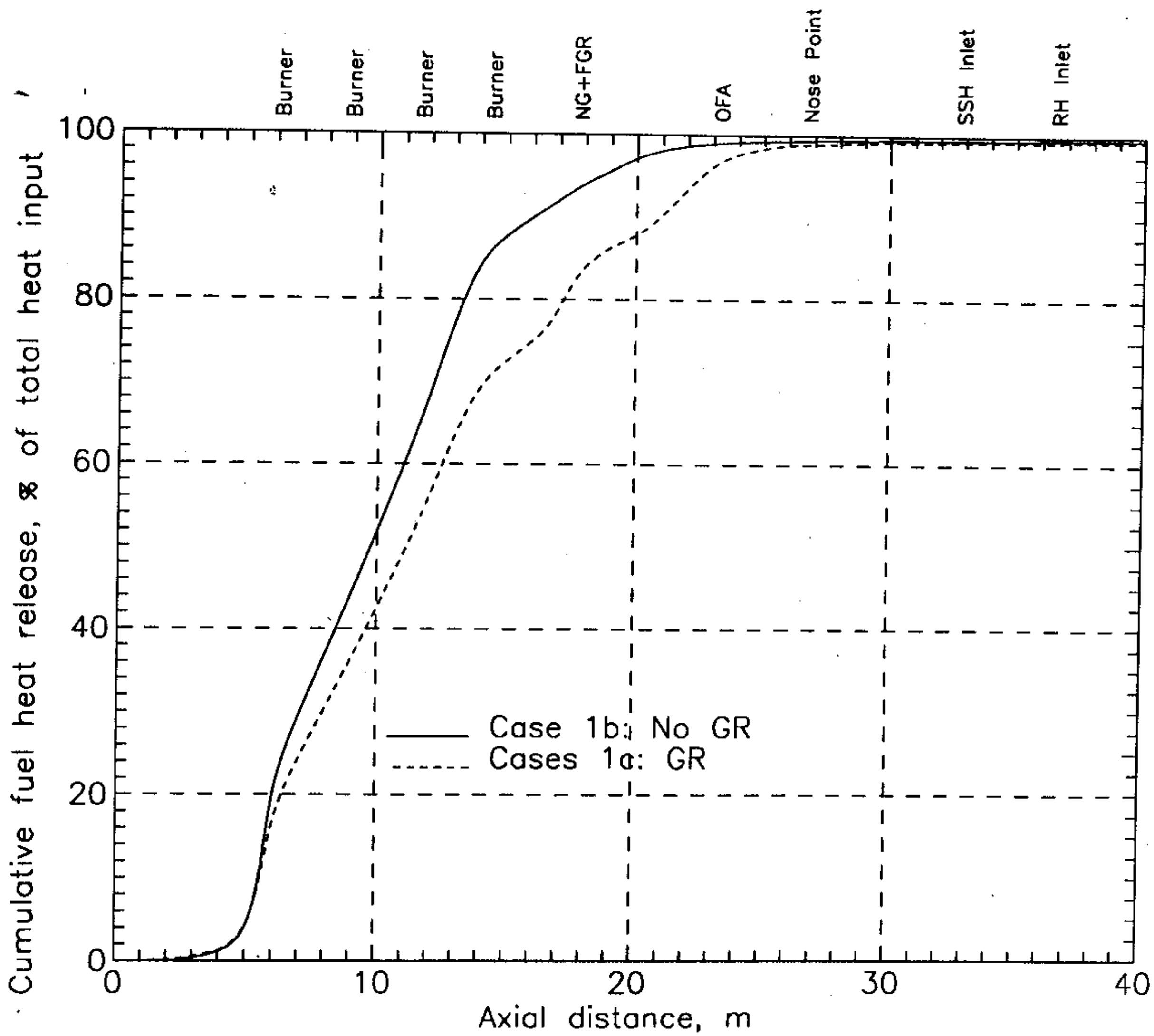


Figure 6–11. Impacts of GR on cumulative total fuel heat release as percentage of total chemical heat input at 100% load.

TABLE 6-6. IMPACTS OF GR ON BOILER PERFORMANCE AT 100% LOAD.

Case Number	Case 1b	Case 1c
Case Definition	No GR	GR
Exit Gas Temperatures(K) of		
RH(rear)	980	988
PSH	674	679
Economizer	651	654
Air Preheater	430	432
Steam Flow(kg/s)		
Main Steam	142.66	141.61
RH Steam	117.36	117.43
Attemperation flow(kg/s)		
SH	2.779	5.741
RH	0.721	1.651
Water/steam temperatures(K)		
Economizer Inlet	526	526
Economizer Outlet	533	534
PSH Inlet	609	609
PSH Outlet	699	711
SSH Attemp. Outlet	687	686
SSH Outlet	803	803
RH Attemp. Outlet	610	603
RH Outlet	812	812
Heat absorptions(kW)		
Furnace	211028	204411
Economizer	4895	5316
PSH	67737	71063
SSH	47333	47764
RH	53131	55026
Unburned Fixed Carbon (% of Total Fixed Carbon Input)	0.63	1.22

bon then enters the fuel-rich reburning zone and, further downstream in the burnout zone, low-temperature fuel-lean zones where conditions are not favorable for conversion of the fixed carbon. Table 6-6 shows a higher percentage of unburned fixed carbon for the GR case. This is due to insufficient residence time available for fixed carbon conversion in fuel-lean high-temperature primary burner zone.

The impact of GR on boiler efficiency is summarized in Table 6-7. The decrease in boiler efficiency for the GR case is primarily caused by an increase in heat loss due to flue gas moisture resulting from the higher hydrogen content of the natural gas and by an increase in heat loss due to combustibles in the ash. Heat loss due to fuel moisture is reduced for the GR case due to decreasing coal flow rates.

In conclusion, the operation of a GR system results in minor reductions of steam flow rates and boiler efficiency and increases the attemperation flows and unburned fixed carbon in the ash. A 1% to 2% increase in thermal load would be required to maintain steam generation at full capacity. However, the attemperation flows need to be evaluated to determine whether there is sufficient capacity available and the limit on the content of unburned fixed carbon in the ash needs to be verified to ensure that it does not exceed allowable limits.

6.3 Low Load Cases

The impacts of varying the amount of FGR introduced into the hopper bottom and increasing SR_1, while GR is applied, are discussed first, followed by a discussion of the impacts of the GR system at low load. This section concludes with a study of reducing SR_3 at low load in order to improve the overall boiler performance and reduce ducting and compressor costs.

6.3.1 Impacts of Introducing FGR into Hopper Bottom at Low Load

The introduction of FGR into the hopper bottom is commonly applied as a means of reducing furnace heat absorption and increasing the absorption in the convective sections in order to maintain reheater and superheater steam temperatures. The impact of FGR introduced into the hopper bottom on mean gas temperature distributions, net total heat flux densities and surface temperatures of ash deposits at low load without GR in operation are shown in Figures 6-12 through 6-14. All three mean properties (gas temperatures, heat fluxes and surface temperatures) drop in the lower

TABLE 6-7. IMPACTS OF GR ON BOILER EFFICIENCY, BASED ON ASME HEAT LOSS METHOD AT 100% LOAD.

Case Number	Case 1b	Case 1c
Case Definition	No GR	GR
Heat Loss due to Dry Gas	5.1238	5.0662
Heat Loss due to Moisture in Fuel	1.6952	1.3787
Heat Loss due to H_2O from Combustion of H_2	4.1536	5.3860
Heat Loss due to Combustible in Refuse	0.3837	0.6037
Heat Loss due to Radiation*	0.2200	0.2200
Unmeasured Losses*	1.5000	1.5000
Total Heat Losses	13.0763	14.1546
Boiler Efficiency	86.9237	85.8454

* Use the value reported in the boiler design performance data sheet.

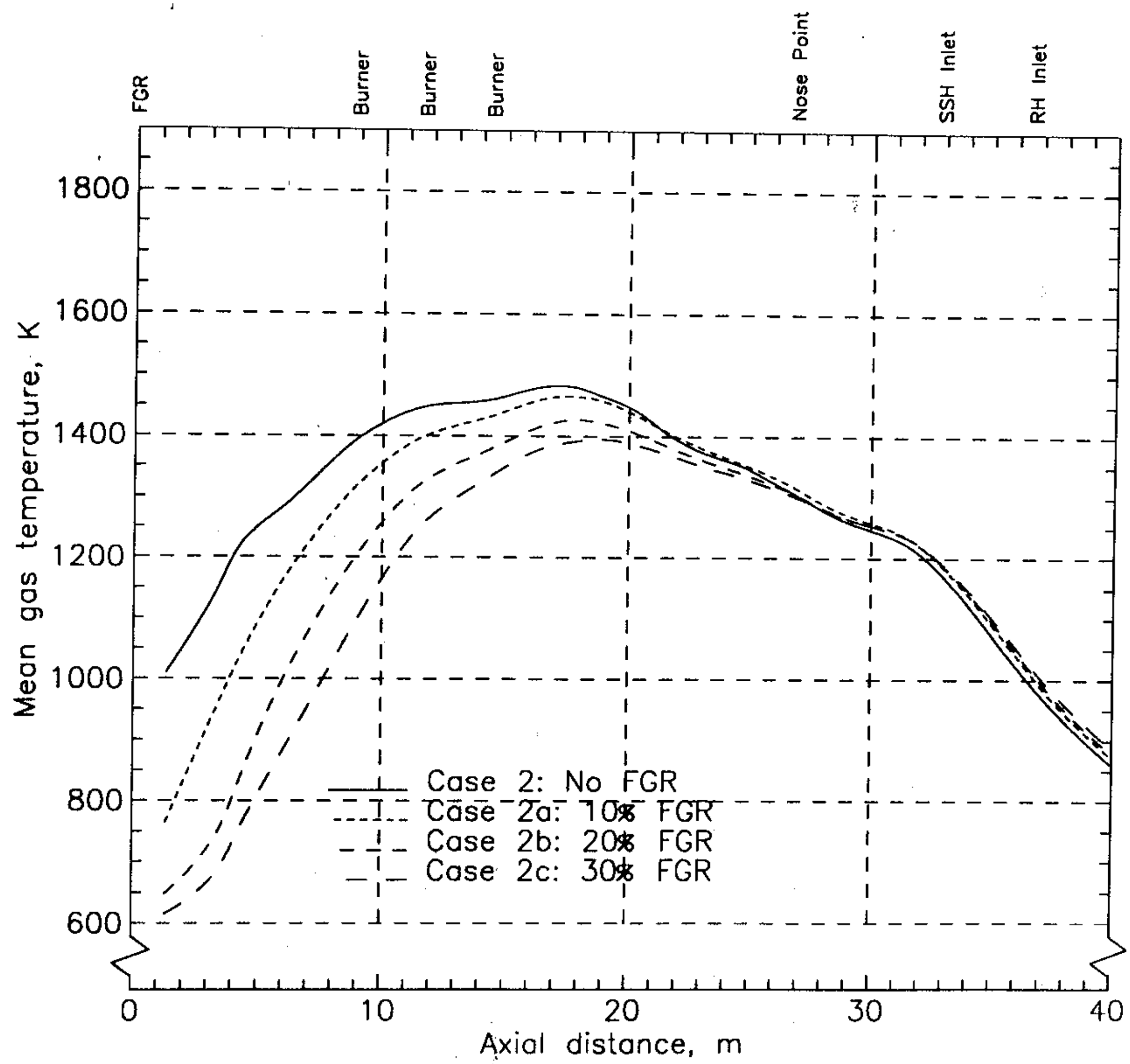


Figure 6–12. Impacts of amount of FGR introduced into hopper bottom on mean gas temperature distributions at 50% load.

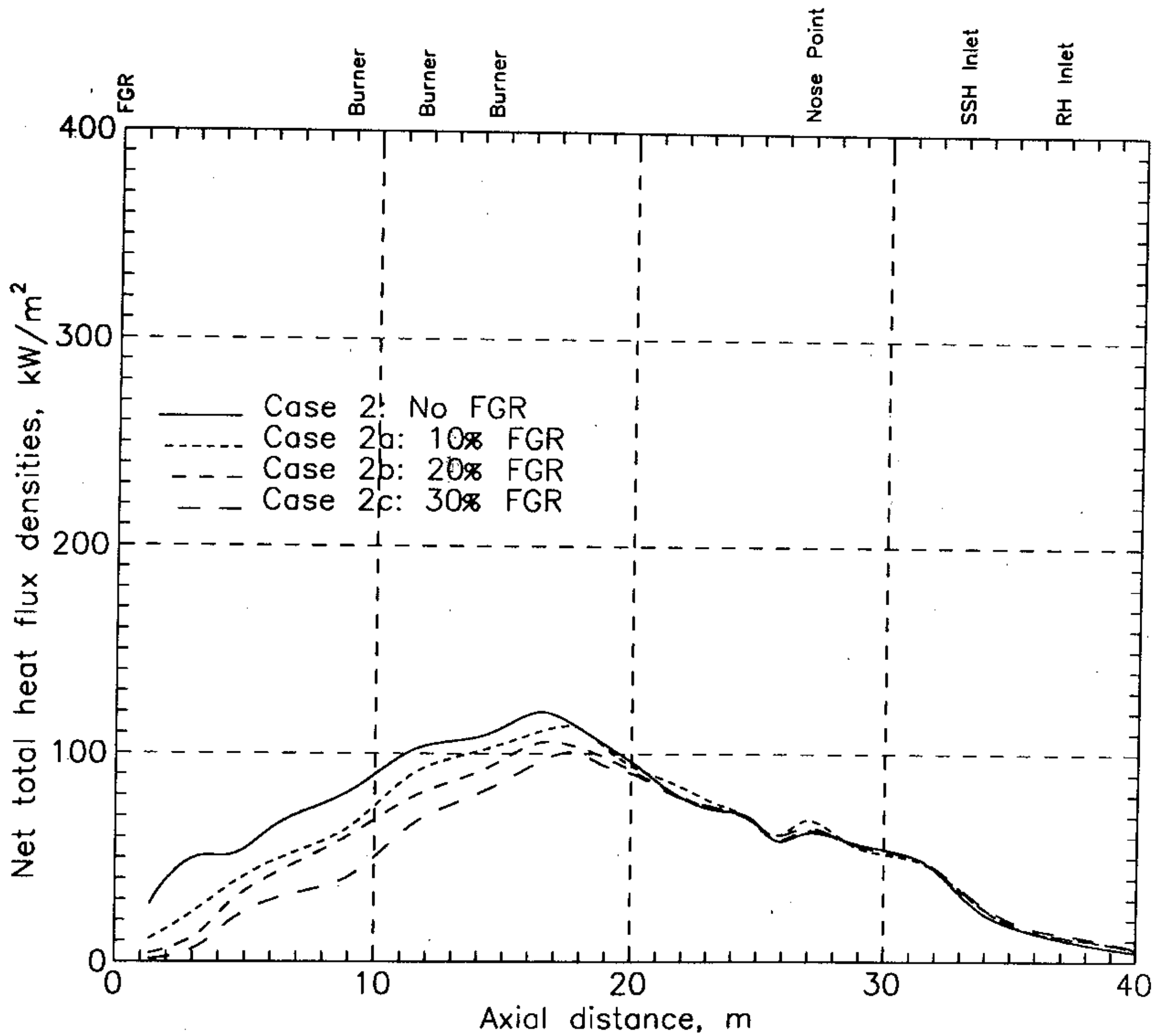


Figure 6–13. Impacts of amount of FGR introduced into hopper bottom on net total heat flux densities at 50% load.

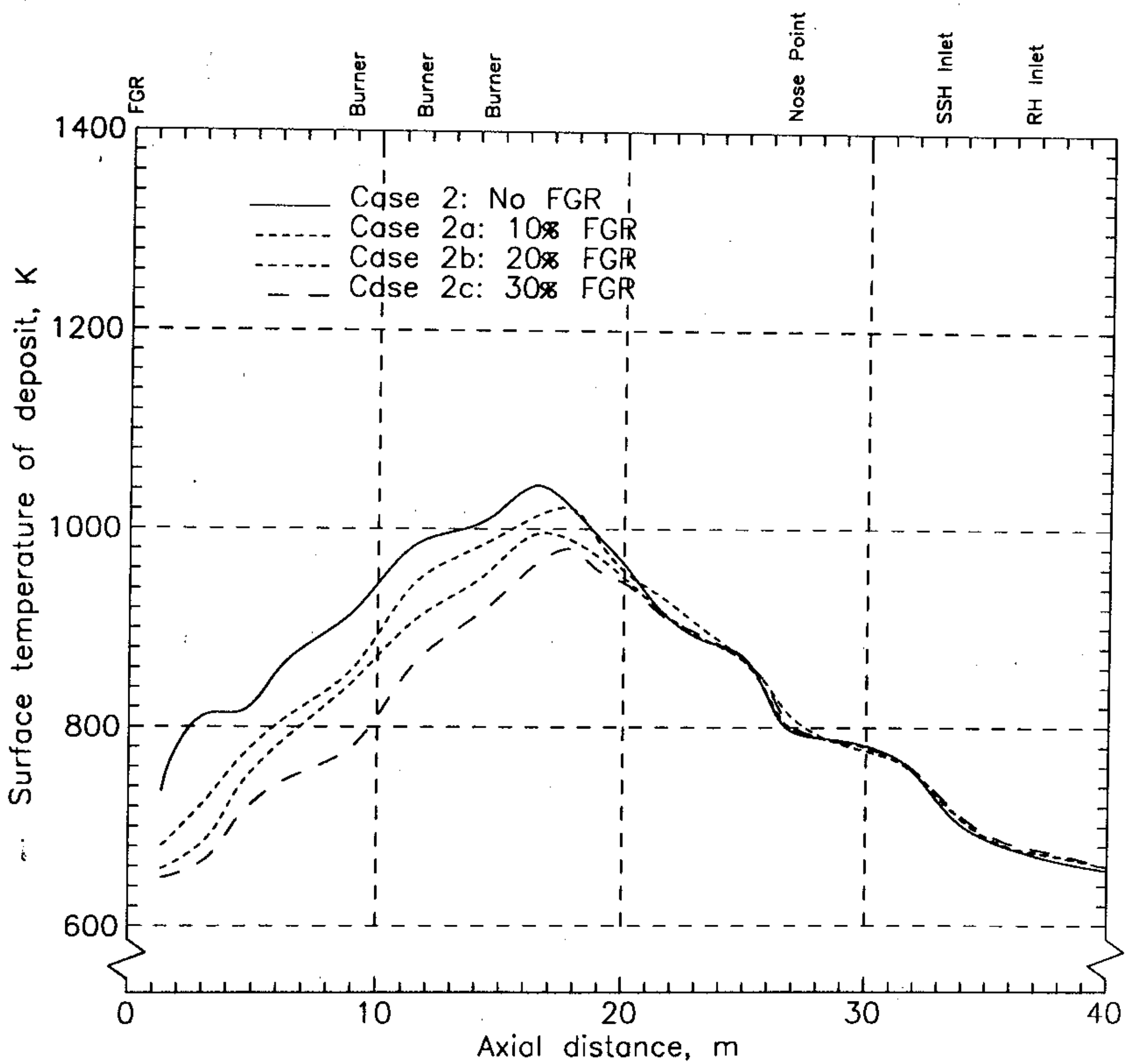


Figure 6–14. Impacts of amount of FGR introduced into hopper bottom on surface temperatures of deposit at 50% load.

furnace as the amount of FGR increases. Gas temperatures at the exit of the reheater (rear) section increase by 27K as FGR is increased from 0% to 30%. This is caused by the reduction in total heat absorbed by the furnace, secondary superheater and reheater sections (Figure 6-15).

The variation in heat absorption for different sections with flue gas recirculation introduced into the hopper is shown in Figure 6-16. Introduction of flue gas at this location produces a marked reduction in heat absorption in the furnace and increases absorption in the convection section. Heat transfer is dominated by radiation in the furnace. Furnace heat absorption is therefore primarily a function of gas temperatures and gas temperature profiles throughout the furnace. Hence the introduction of gas recirculation into the furnace hopper reduces furnace absorption by reducing gas temperatures. The major portion of the heat absorbed in the superheater, reheater and economizer is transferred by convection which makes heat absorption in these regions dependent primarily on gas temperature and gas velocity. Both of these parameters are affected by flue gas recirculation. The gas temperature entering the secondary superheater is relatively unchanged by flue gas recirculation as illustrated in Figure 6-12. Increasing the amount of flue gas recirculation increases the gas velocity and therefore heat absorption in the secondary superheater. Heat absorption in the reheater, primary superheater, and economizer also increases with the greatest increase occurring at the cold end of the unit. These changes are typical of those that occur when the flue gas recirculation amount is varied.

Table 6-8 shows that 10% FGR helps to maintain the exit steam temperature of the superheater section. Heat absorption in the radiant furnace section drops and the heat absorption of both the reheater and superheater sections increases as FGR amount increases. As a consequence, steam generation is reduced and the superheater and reheater attemperation flows are increased. The increase in unburned fixed carbon is primarily caused by the drop in gas temperatures in the burner zones. Table 6-9 shows that the drop in boiler efficiency is primarily caused by increases in the dry gas and combustible heat losses as FGR increases.

6.3.2 Impacts of Increasing SR_1 with GR at Low Load

The results of variations of the SR_1 at low load with constant SR_2 and SR_3 are plotted in Figures 6-17 through 6-20. The total fuel heat input is the same for the three cases. Increasing SR_1 therefore represents a shift in the thermal input contributed by the coal burners to the reburning zone as shown in Table 6-1.

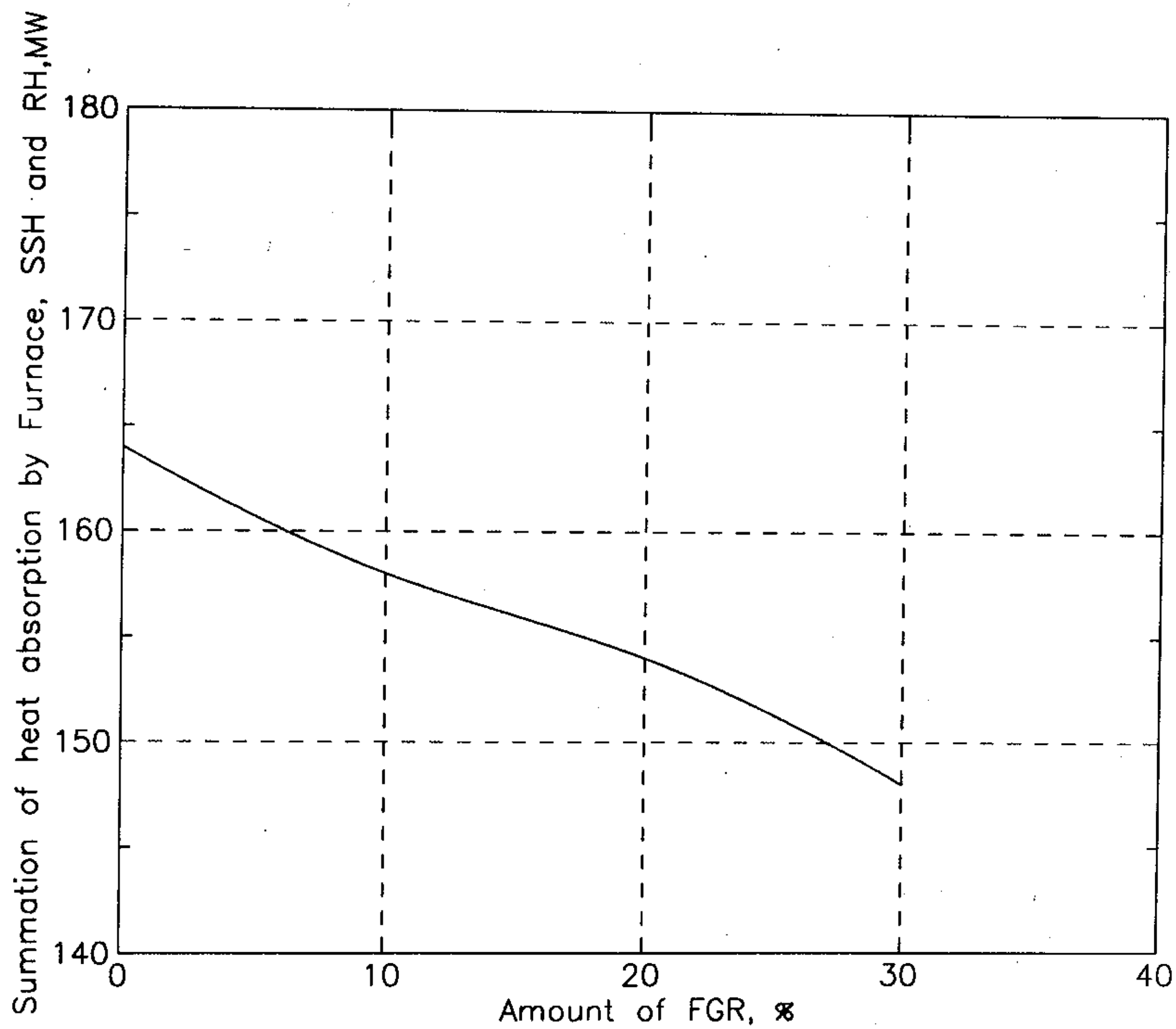


Figure 6–15. Impact of amount of FGR introduced into hopper bottom on summation of heat absorption by Furnace, SSH and RH.

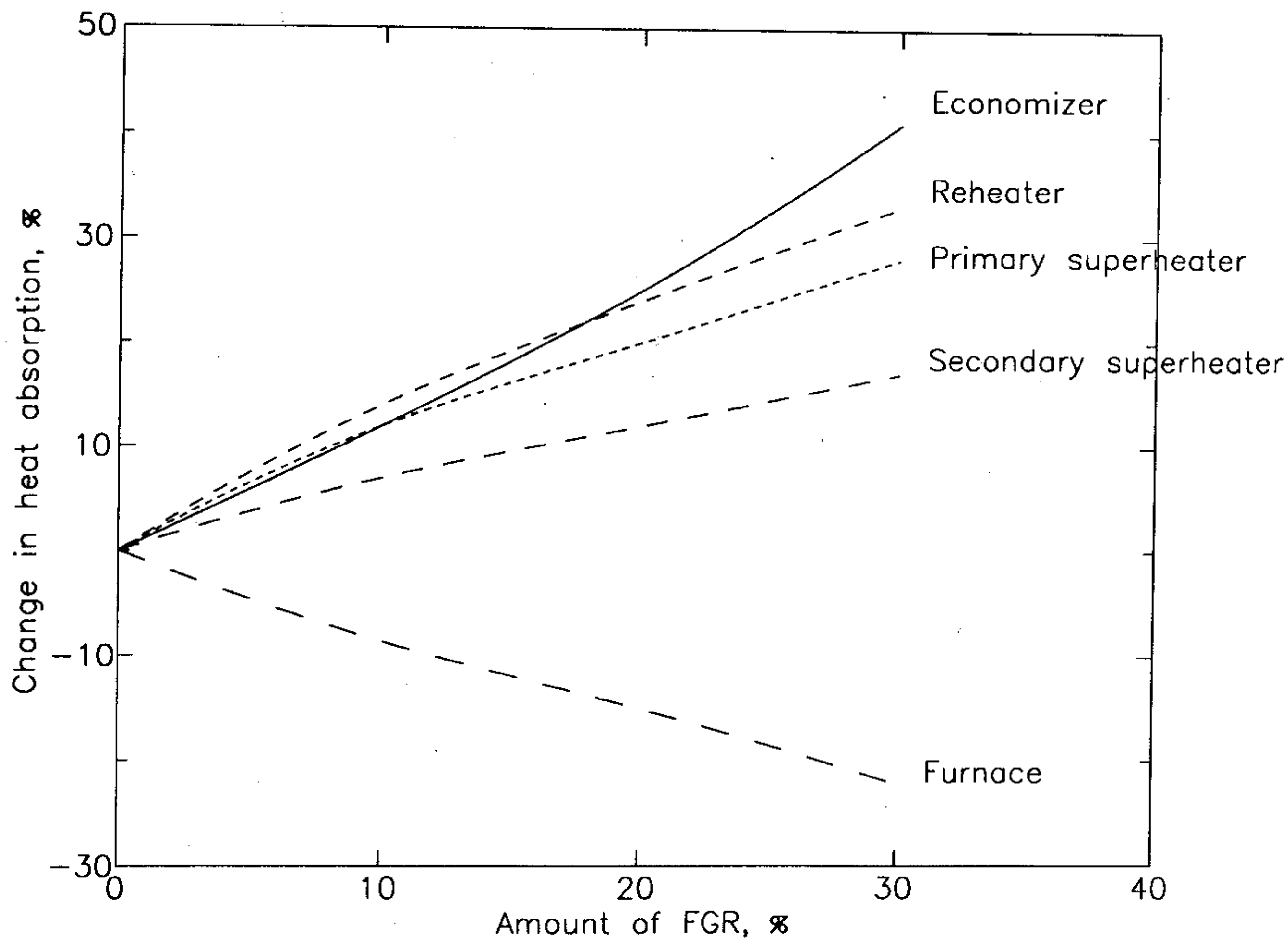


Figure 6-16 Impact of amount of FGR introduced into hopper bottom on change in heat absorption of each section.

TABLE 6-8. IMPACTS OF FGR AMOUNT ON BOILER PERFORMANCE WHILE FGR IS INTRODUCED INTO HOPPER BOTTOM FOR 50% LOAD COAL FIRING CASES.

Case Number	Case 2	Case 2a	Case 2b	Case 2c
Case Definition	Baseline 0% FGR	10% FGR	20% FGR	30% FGR
Exit Gas Temperatures(K) of				
RH(rear)	862	875	881	889
PSH	637	646	656	670
Economizer	608	617	626	639
Air Preheater	382	384	387	391
Steam Flow(kg/s)				
Main Steam	70.45	67.19	65.68	63.76
RH Steam	58.20	55.51	54.26	53.24
Attemperation flow(kg/s)				
SH	0.000	2.355	4.960	7.620
RH	0.000	0.000	0.000	0.561
Water/steam temperatures(K)				
Economizer Inlet	484	484	484	484
Economizer Outlet	495	498	500	504
PSH Inlet	606	606	606	606
PSH Outlet	679	704	728	759
SSH Attemp. Outlet	679	681	675	667
SSH Outlet	784	803	803	803
RH Attemp. Outlet	565	565	565	555
RH Outlet	739	773	795	806
Heat absorptions(kW)				
Furnace	119961	109700	102033	93455
Economizer	3487	3915	4368	4927
PSH	28361	31730	34092	36260
SSH	21987	23633	24685	25716
RH	21751	24847	26961	28927
Unburned Fixed Carbon (% of Total Fixed Carbon Input)	0.21	0.38	0.71	1.21

TABLE 6-9. IMPACTS OF FGR AMOUNT ON BOILER EFFICIENCY, BASED ON ASME HEAT LOSS METHOD WHILE FGR IS INTRODUCED INTO HOPPER BOTTOM FOR 50% LOAD COAL FIRING CASES.

Case Number	Case 2	Case 2a	Case 2b	Case 2c
Case Definition	Baseline 0% FGR	10% FGR	20% FGR	30% FGR
Heat Loss due to Dry Gas	3.8278	3.9487	4.0695	4.2270
Heat Loss due to Moisture in Fuel	1.6379	1.6409	1.6441	1.6483
Heat Loss due to H_2O from Combustion of H_2	4.0130	4.0205	4.0283	4.0385
Heat Loss due to Combustible in Refuse	0.1279	0.2314	0.4324	0.7369
Heat Loss due to Radiation*	0.4500	0.4500	0.4500	0.4500
Unmeasured Losses*	1.5000	1.5000	1.5000	1.5000
Total Heat Losses	11.5565	11.7916	12.1242	12.6006
Boiler Efficiency	88.4435	88.2084	87.8758	87.3994

* Use the value reported in the boiler design performance data sheet.

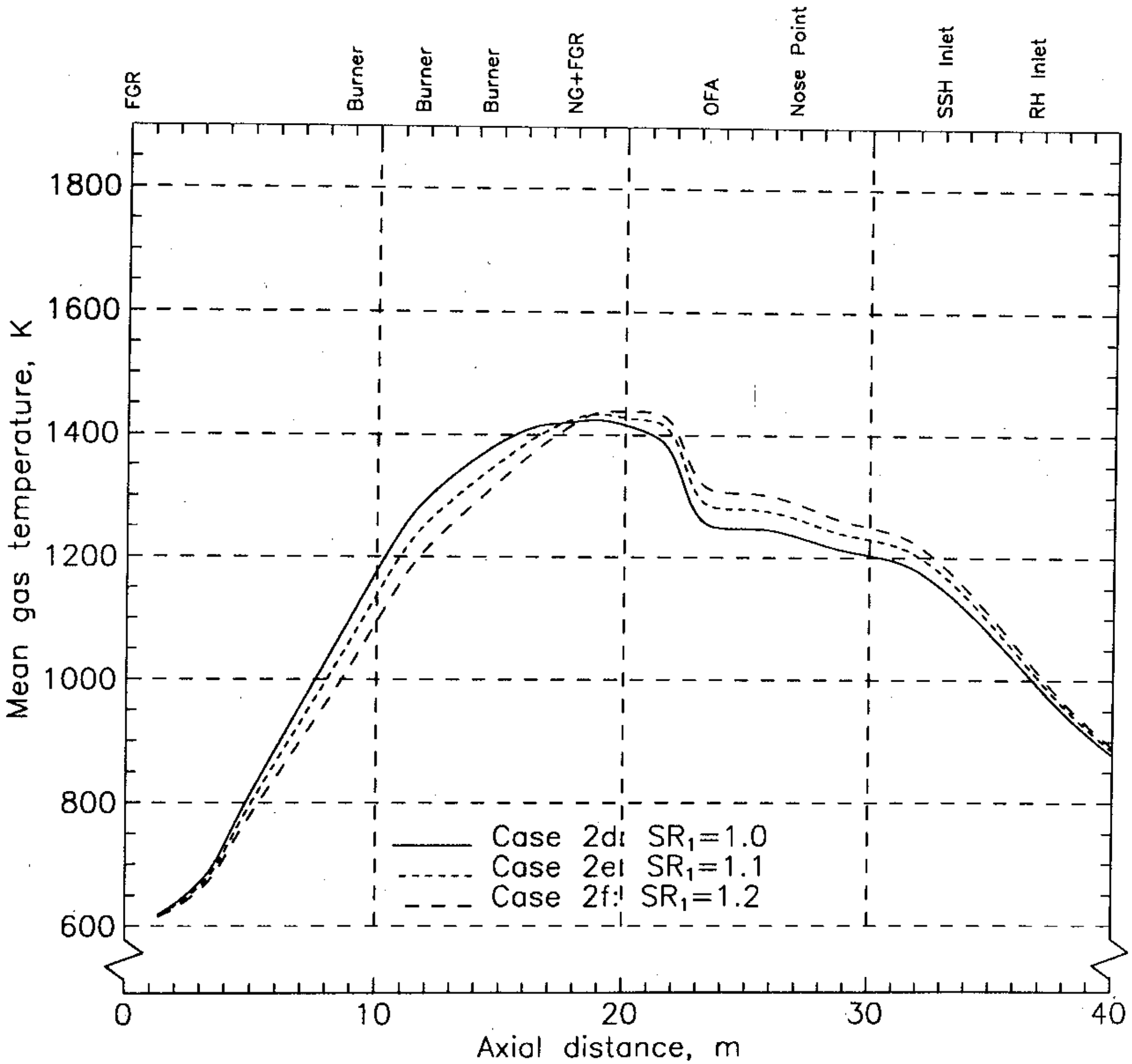


Figure 6–17. Impacts of SR_1 on mean gas temperature distributions at 50% load while 30% FGR is introduced into hopper bottom.

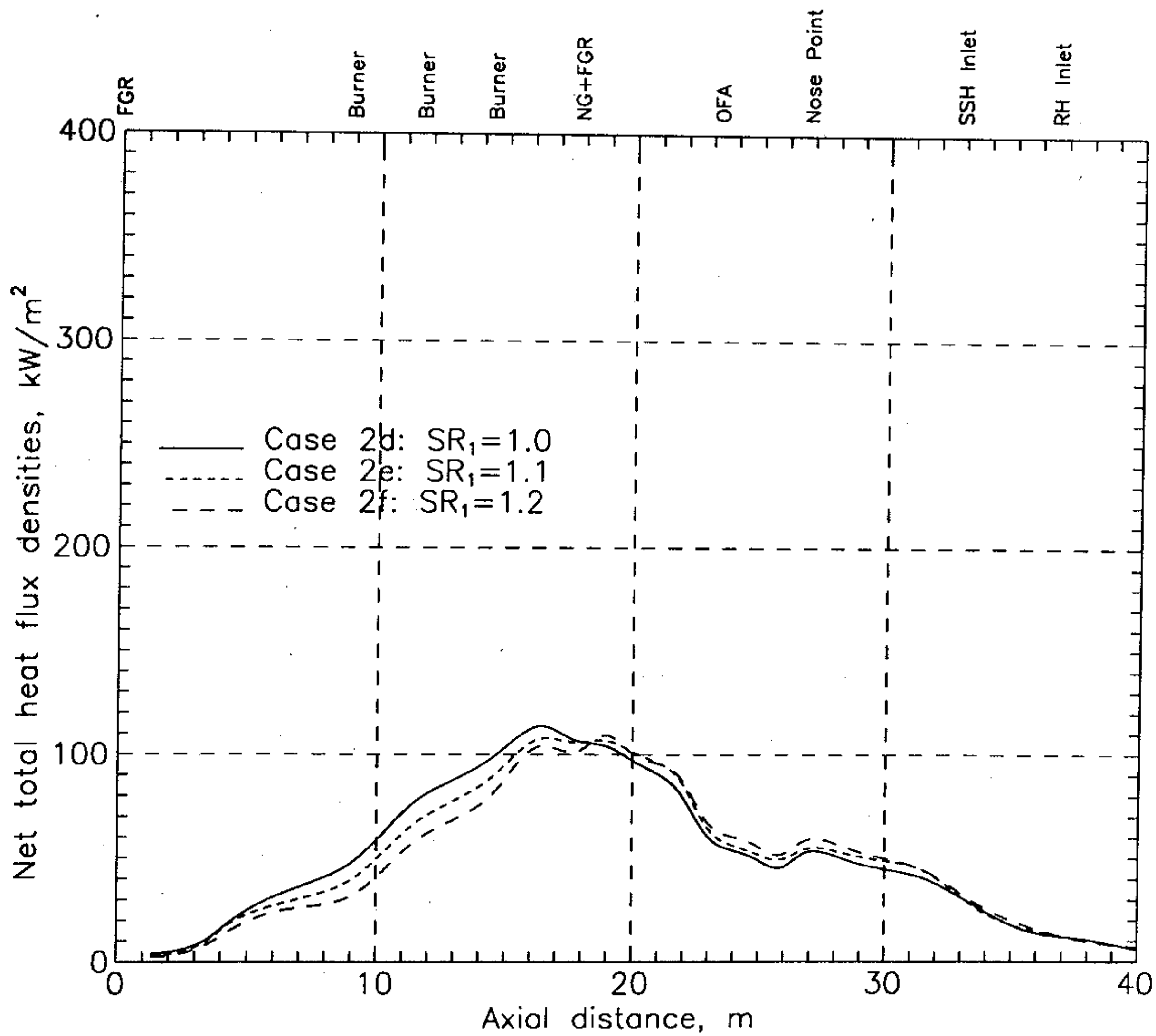


Figure 6–18. Impacts of SR_1 on net total heat flux densities at 50% load while 30% FGR is introduced into hopper bottom.

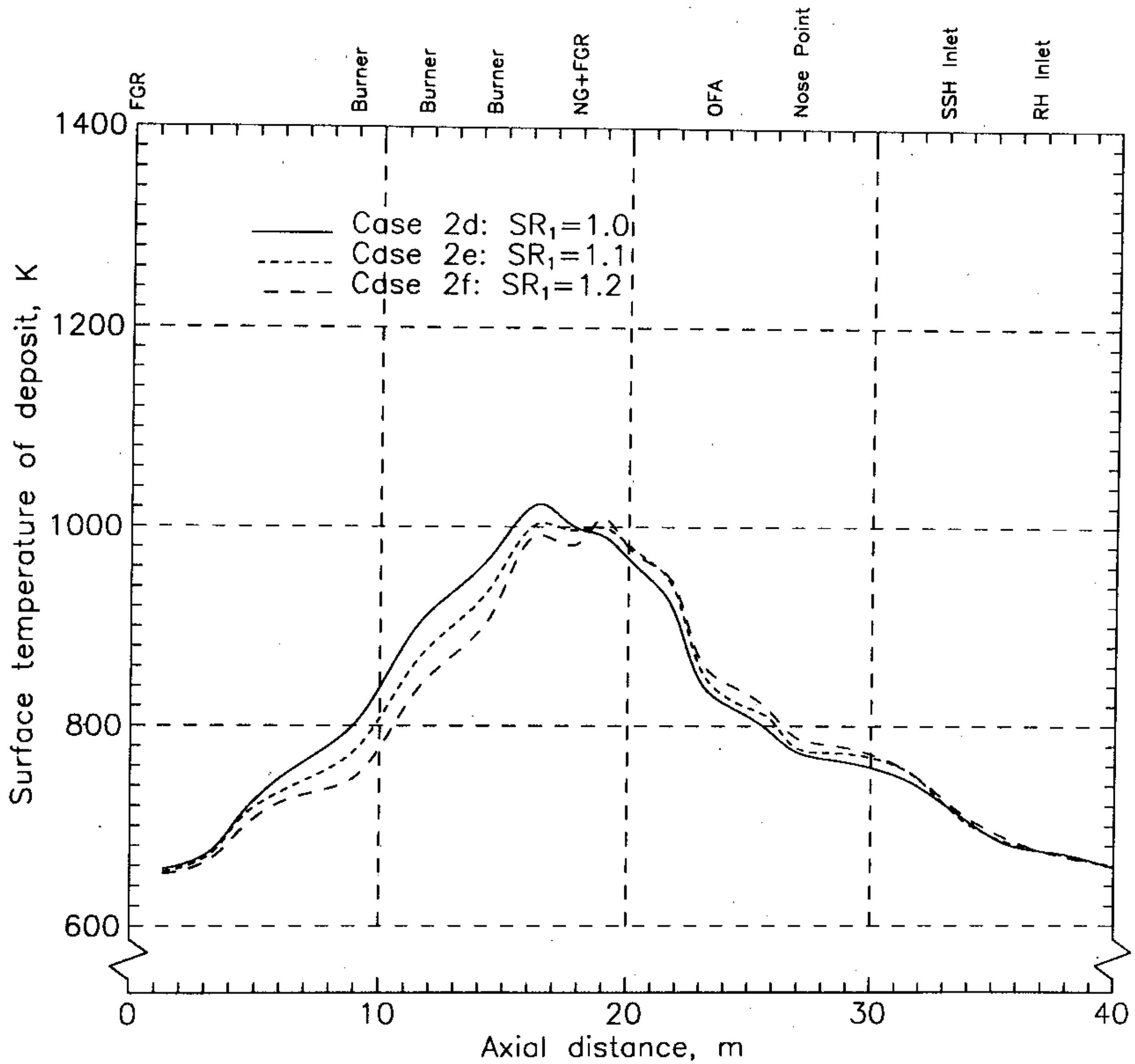


Figure 6–19. Impacts of SR_1 on surface temperatures of deposit at 50% load while 30% FGR is introduced into hopper bottom.

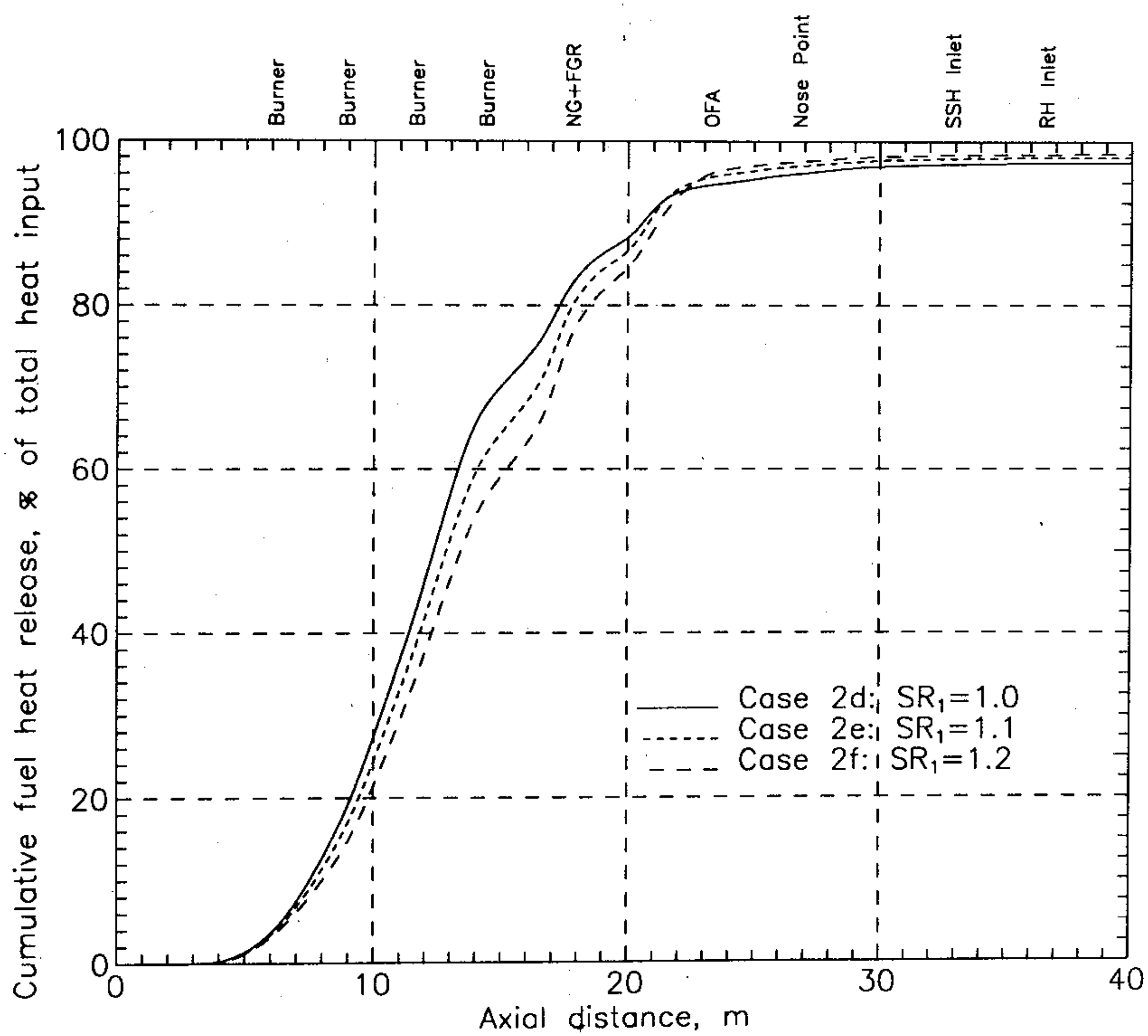


Figure 6–20. Impacts of SR_1 on cumulative total fuel heat release as percentage of total chemical heat input at 50% load.

The mean gas temperature profiles in Figure 6-17 show that temperatures below the natural gas ports drop and the temperatures above the natural gas ports increase as SR_1 is increased. The drop in lower furnace gas temperatures is due to the reduced thermal input in that region as well as the increased excess air while the increase in upper furnace gas temperatures is due to the higher natural gas thermal input as SR_1 increases. There is also a drop in temperature upon the introduction of over fire air. The same trends are shown in Figures 6-18 and 6-19 for the net total heat flux densities and surface temperatures of ash deposits respectively. Figure 6-20 compares cumulative fuel heat releases as SR_1 is varied.

The impacts of primary zone stoichiometry on boiler performance are shown in Table 6-10. Increasing SR_1 reduces steam generation because less heat is absorbed in the furnace and increases attemperation requirements since more heat is available in the upper furnace. The improvement in carbon burnout is caused by higher gas temperatures in the upper furnace as SR_1 increases.

The improvement in boiler efficiency as primary zone stoichiometry increases is shown in Table 6-11. The increased efficiency is primarily caused by changes in the fuel moisture, unburned fixed carbon, and heat input due to natural gas. As a consequence heat loss due to combustibles in ash decreases when SR_1 increases. The increase in the fraction of heat input contributed by natural gas increases the heat loss due to H_2O from the combustion of hydrogen and reduces the heat loss due to coal moisture as SR_1 is increased.

6.3.3 Impacts of GR at Low Load

Figures 6-21 through 6-23 show the impacts of GR on mean gas temperatures, net total heat flux densities, and surface temperatures of ash deposits for both non-GR and GR cases (Cases 2c and 2f). Tables 6-12 and 6-13 summarize the impacts of GR at low load on boiler performance and efficiency, respectively, while 30% FGR is introduced into hopper bottom and SR_1 and SR_2 are maintained at design values of 1.2 and 0.9 respectively. Most of the trends observed in Section 6.2.3 for the full load case can be observed for the low load case except that the increase in the main attemperation flow rate is less significant than at full load.

TABLE 6-10. IMPACTS OF PRIMARY ZONE STOICHIOMETRY ON BOILER PERFORMANCE WHILE 30% FGR IS INTRODUCED INTO HOPPER BOTTOM FOR 50% LOAD GAS REBURNING CASES.

Case Number	Case 2d	Case 2e	Case 2f
Case Definition	GR SR_1 = 1.0	GR SR_1 = 1.1	GR SR_1 = 1.2
Exit Gas Temperatures(K) of			
RH(rear)	879	887	892
PSH	668	672	660
Economizer	638	641	644
Air Preheater	390	391	392
Steam Flow(kg/s)			
Main Steam	62.99	62.99	62.74
RH Steam	52.07	52.62	52.91
Attemperation flow(kg/s)			
SH	6.233	7.473	8.654
RH	0.037	0.588	1.083
Water/steam temperatures(K)			
Economizer Inlet	484	484	484
Economizer Outlet	504	504	506
PSH Inlet	606	606	606
PSH Outlet	752	763	772
SSH Attemp. Outlet	676	671	665
SSH Outlet	803	803	803
RH Attemp. Outlet	565	555	546
RH Outlet	806	806	806
Heat absorptions(kW)			
Furnace	94484	92196	89549
Economizer	4969	5087	5228
PSH	35720	36486	36951
SSH	23234	24642	26056
RH	27207	28665	29865
Unburned Fixed Carbon (% of Total Fixed Carbon Input)	4.48	3.67	3.14

TABLE 6-11. IMPACTS OF PRIMARY ZONE STOICHIOMETRY ON BOILER EFFICIENCY, BASED ON ASME HEAT LOSS METHOD WHILE 30% FGR IS INTRODUCED INTO HOPPER BOTTOM FOR 50% GAS REBURNING CASES.

Case Number	Case 2d	Case 2e	Case 2f
Case Definition	GR $SR_1 = 1.0$	GR $SR_1 = 1.1$	GR $SR_1 = 1.2$
Heat Loss due to Dry Gas	4.1417	4.1577	4.1706
Heat Loss due to Moisture in Fuel	1.4773	1.3397	1.2262
Heat Loss due to H_2O from Combustion of H_2	4.6975	5.2336	5.6797
Heat Loss due to Combustible in Refuse	2.4463	1.8160	1.4204
Heat Loss due to Radiation*	0.4500	0.4500	0.4500
Unmeasured Losses*	1.5000	1.5000	1.5000
Total Heat Losses	14.7129	14.4969	14.4468
Boiler Efficiency	85.2871	85.5031	85.5532

* Use the value reported in the boiler design performance data sheet.

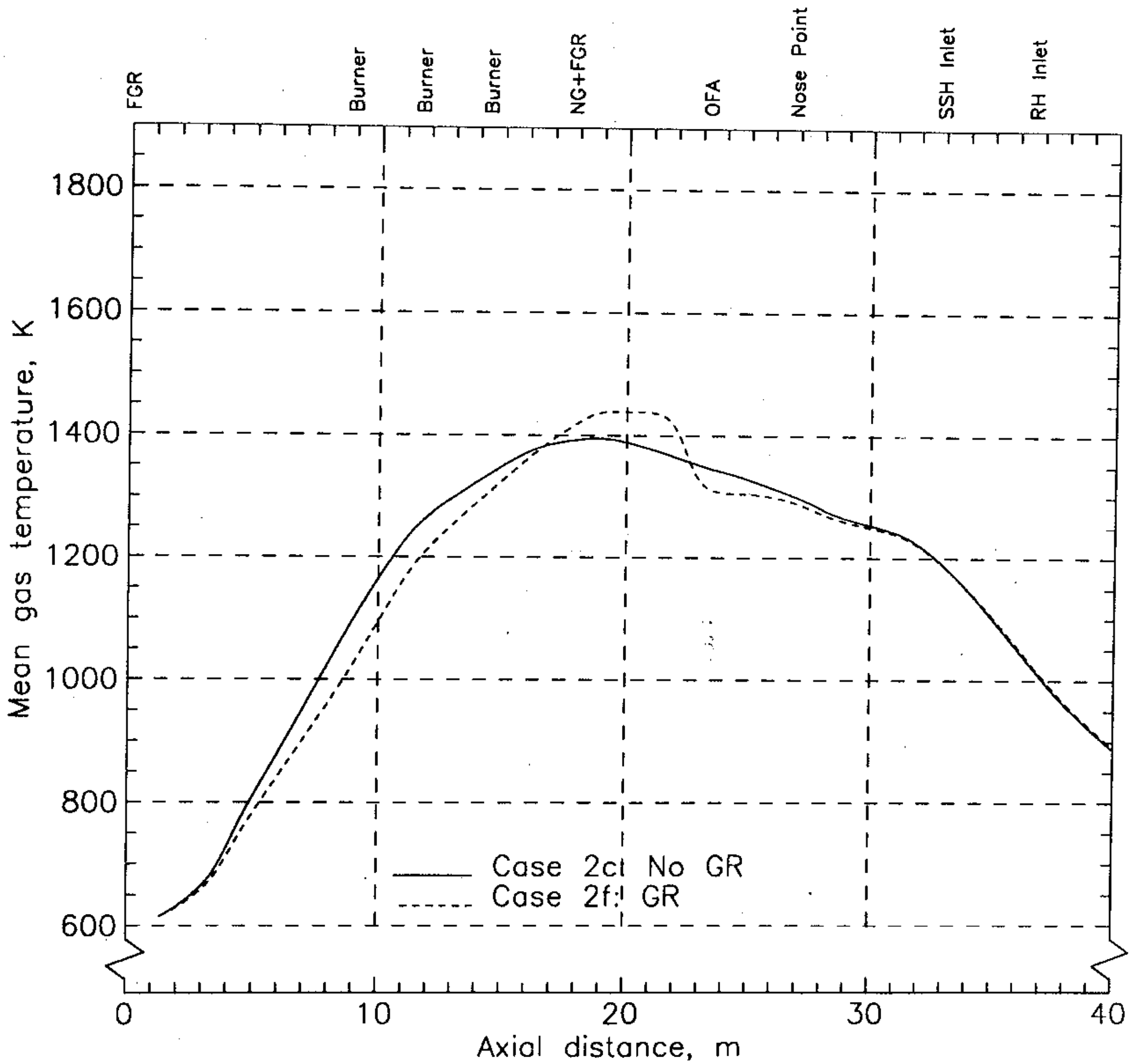


Figure 6–21. Impacts of GR on mean gas temperature distributions at 50% load while 30% FGR is introduced into hopper bottom.

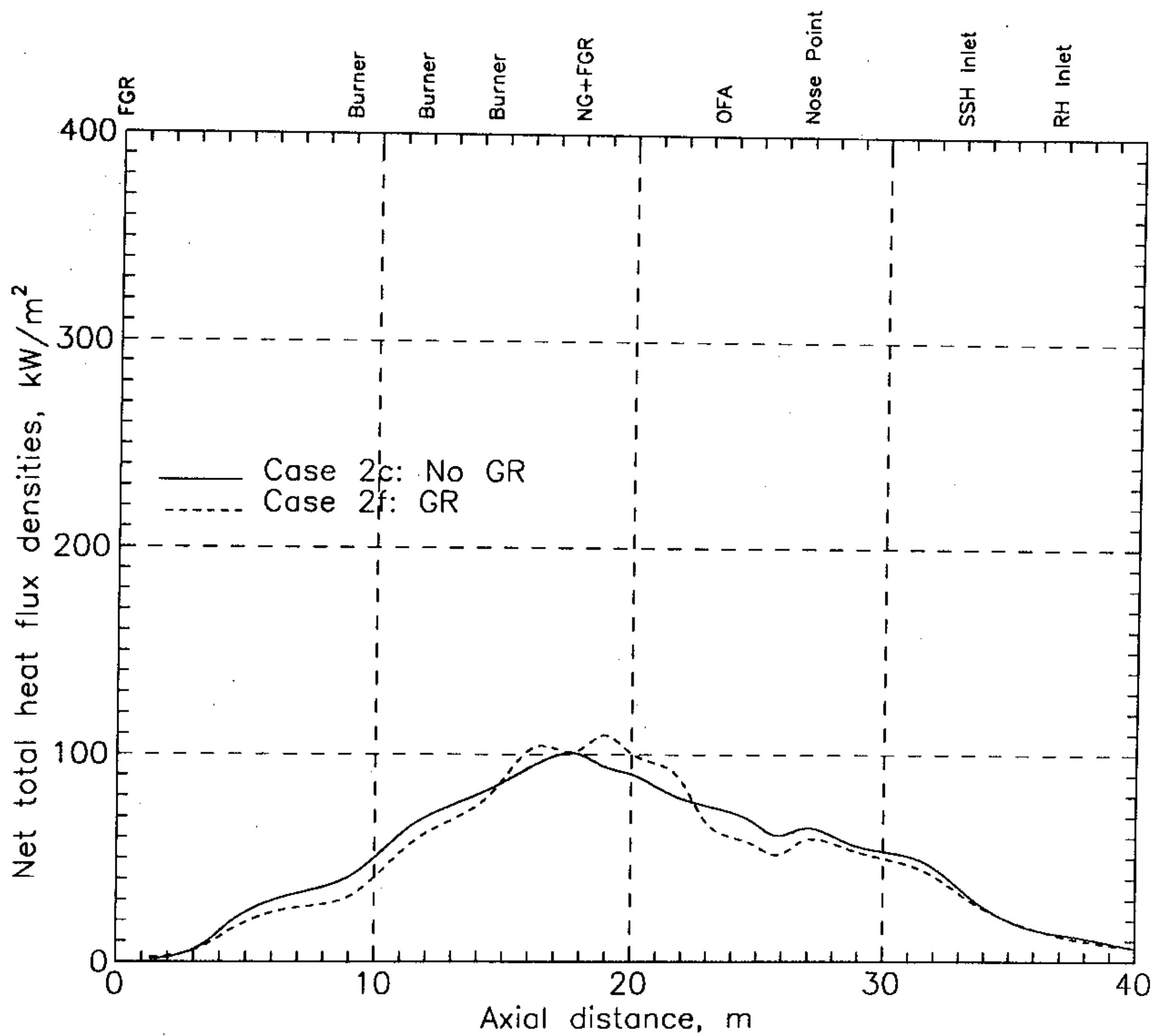


Figure 6–22. Impacts of GR on net total heat flux densities at 50% load while 30% FGR is introduced into hopper bottom.

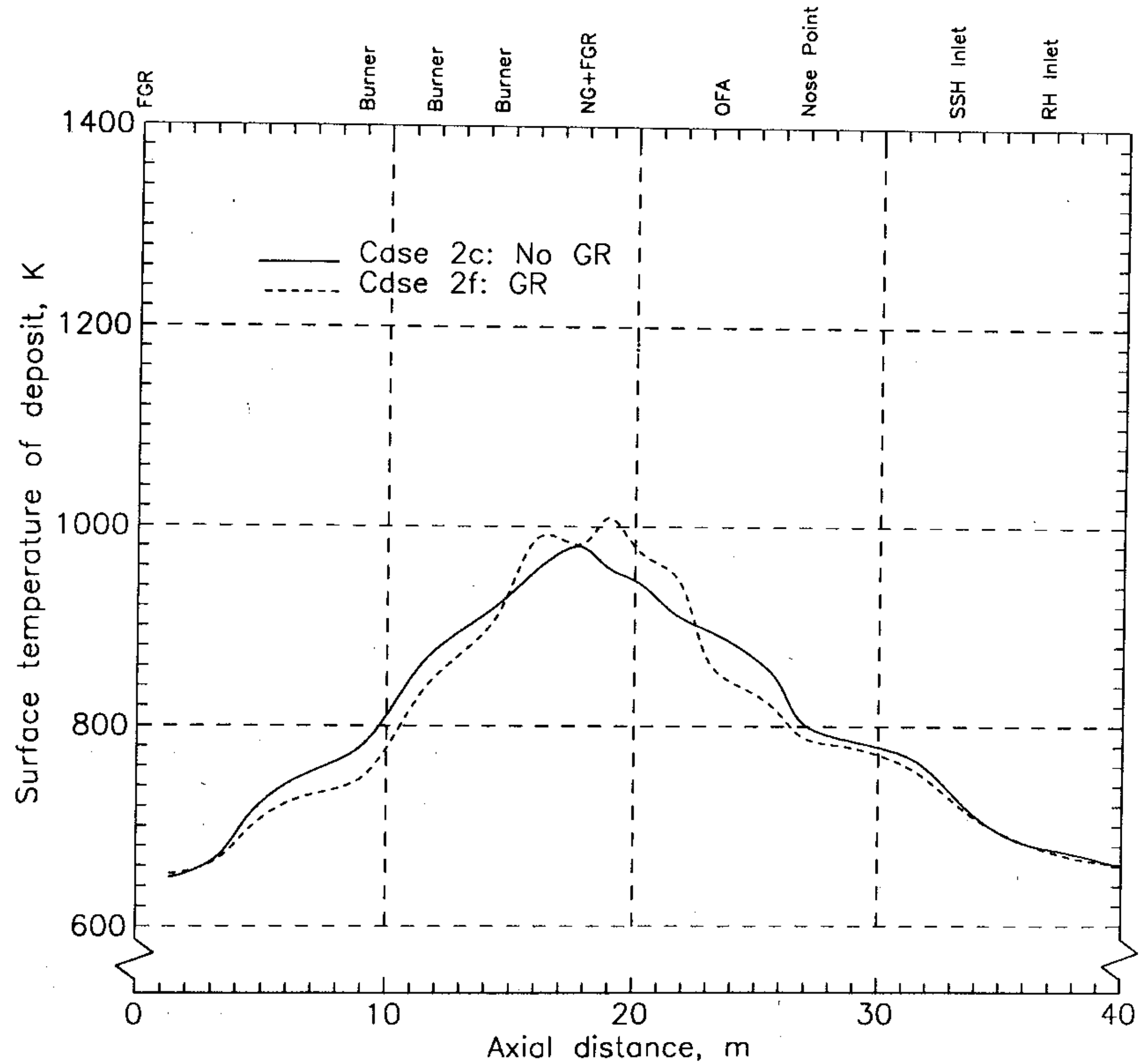


Figure 6–23. Impacts of GR on surface temperatures of deposit at 50% load while 30% FGR is introduced into hopper bottom.

TABLE 6-12. IMPACTS OF GR ON BOILER PERFORMANCE AT 50% LOAD WHILE 30% FGR IS INTRODUCED INTO HOPPER BOTTOM.

Case Number	Case 2c	Case 2f
Case Definition	No GR	GR
Exit Gas Temperatures(K) of		
RH(rear)	889	892
PSH	670	660
Economizer	639	644
Air Preheater	391	392
Steam Flow(kg/s)		
Main Steam	63.76	62.74
RH Steam	53.24	52.91
Attemperation flow(kg/s)		
SH	7.620	8.654
RH	0.561	1.083
Water/steam temperatures(K)		
Economizer Inlet	484	484
Economizer Outlet	504	506
PSH Inlet	606	606
PSH Outlet	759	772
SSH Attemp. Outlet	667	665
SSH Outlet	803	803
RH Attemp. Outlet	555	546
RH Outlet	806	806
Heat absorptions(kW)		
Furnace	93455	89549
Economizer	4927	5228
PSH	36260	36951
SSH	25716	26056
RH	28927	29865
Unburned Fixed Carbon (% of Total Fixed Carbon Input)	1.21	3.14

TABLE 6-13. IMPACTS OF GR ON BOILER EFFICIENCY, BASED ON ASME HEAT LOSS METHOD AT 50% LOAD WHILE 30% FGR IS INTRODUCED INTO HOPPER BOTTOM.

Case Number	Case 2c	Case 2f
Case Definition	No GR	GR
Heat Loss due to Dry Gas	4.2270	4.1706
Heat Loss due to Moisture in Fuel	1.6483	1.2262
Heat Loss due to H_2O from Combustion of H_2	4.0385	5.6797
Heat Loss due to Combustible in Refuse	0.7369	1.4204
Heat Loss due to Radiation*	0.4500	0.4500
Unmeasured Losses*	1.5000	1.5000
Total Heat Losses	12.6006	14.4468
Boiler Efficiency	87.3994	85.5532

* Use the value reported in the boiler design performance data sheet.

6.3.4 Impacts of Decreasing SR_3 at low Load

Lowering SR_3 may reduce over fire air requirements and influence the design of the over fire air ducting and compressor capacity. A new value of SR_3 was determined by interpolating the B&W design stoichiometry at low load (approximately 1.22). Table 6-2 indicates that decreasing SR_3 significantly reduces over fire air mass flow rate and results in 16.29 kg/s less total mass input to the boiler. Figures 6-24 through 6-26 show the impacts of changing SR_3 from 1.3909 to 1.2200 on mean gas temperature distributions, net total heat flux densities, and surface temperatures of ash deposits. There was no difference on mass flow rates of coal, natural gas and air to the coal burners between these two cases although the unit was modeled with high and low SR_3 (Cases 2f and 2g). However the same percentage of FGR (30%) specified in the low SR_3 case actually represented 3.66 kg/s less FGR mass than the high SR_3 case due to less total mass input for low excess air. An earlier discussion in Section 6.3.1 concluded that increasing the amount of FGR reduced temperatures and heat fluxes in the lower furnace. Therefore, it is observed that gas and surface temperatures and wall heat fluxes in the lower furnace for the low FGR amount case (Case 2g) are higher than those for the high FGR amount case (Case 2f). Lower over fire air rates (Case 2g) result in a smaller impact on temperatures and heat fluxes around the over fire air injection location.

The impacts of decreasing SR_3 on boiler performance are summarized in Table 6-14. Steam flow is increased at low SR_3 values due to greater rates of heat absorption in the lower furnace and attemperation flow is decreased as a result of less heat being absorbed in the upper furnace. The higher temperatures in the lower furnace are largely responsible for the improved carbon burnout at low SR_3 values.

Table 6-15 indicates that boiler efficiency improves with decreased SR_3. The improved carbon conversion of low SR_3 reduces the heat loss due to unburned combustibles in the ash. Lower total mass input to the boiler also results in a significant reduction of heat loss due to dry gas.

6.4 Thermal Performance Summary

Thermal performance models were used to evaluate Cherokee Unit 3 to determine the impacts of GR and LNB on the boiler performance. Model studies were conducted for baseline and GR and LNB cases with burner swirl effect included. Initially, the models were calibrated against boiler

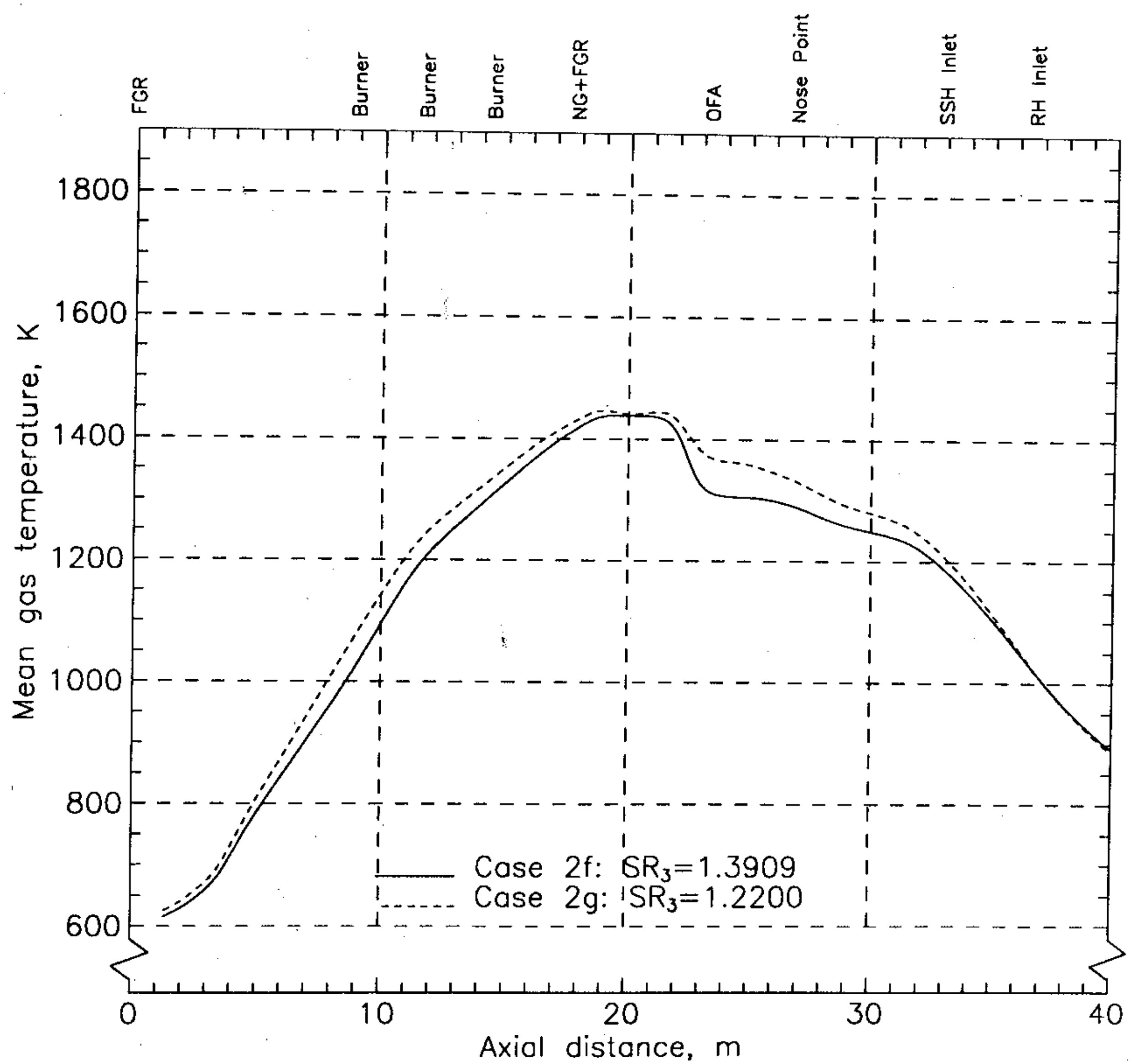


Figure 6–24. Impacts of SR_3 on mean gas temperature distributions at 50% load while 30% FGR is introduced into hopper bottom.

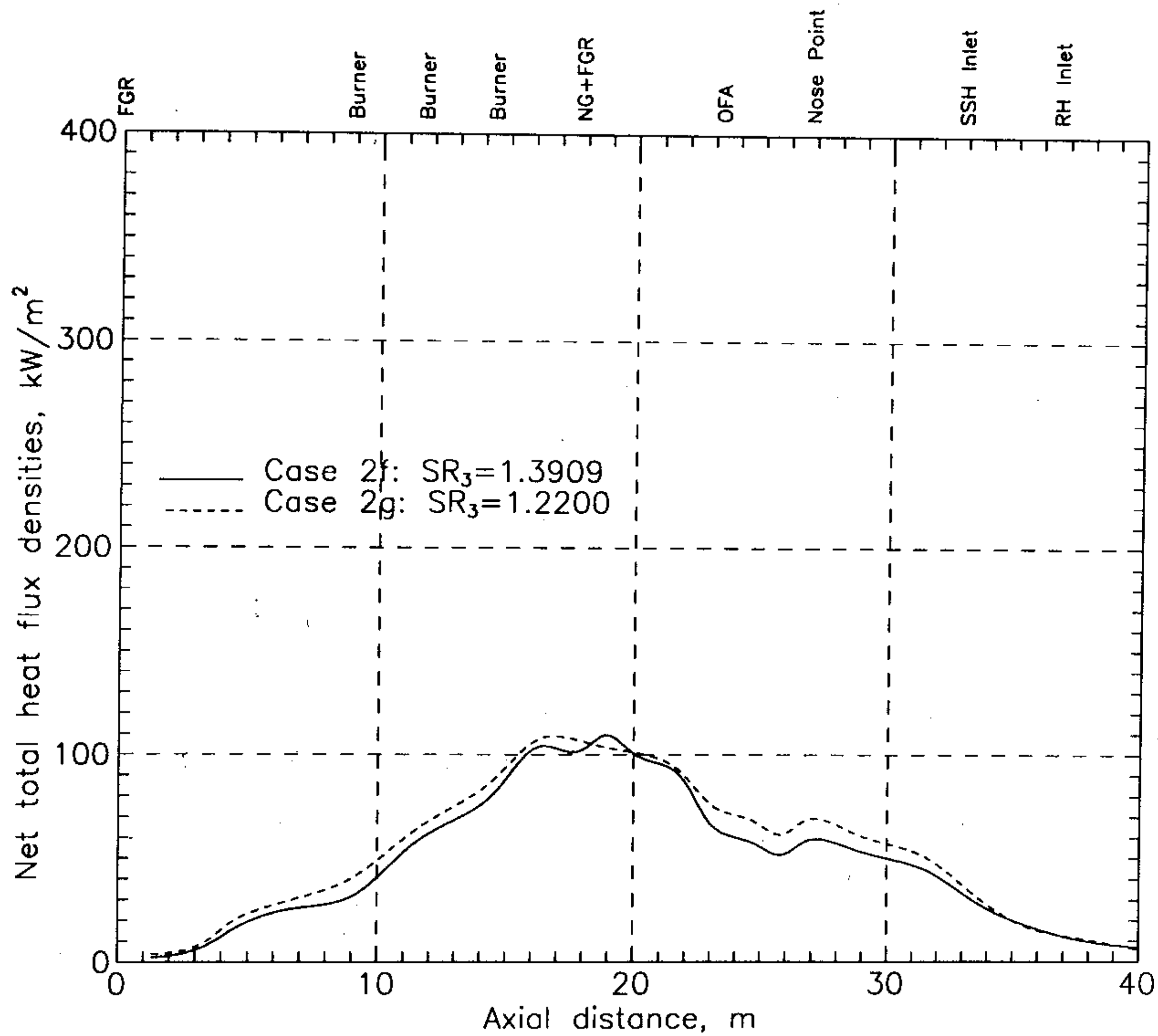


Figure 6–25. Impacts of SR_3 on net total heat flux densities at 50% load while 30% FGR is introduced into hopper bottom.

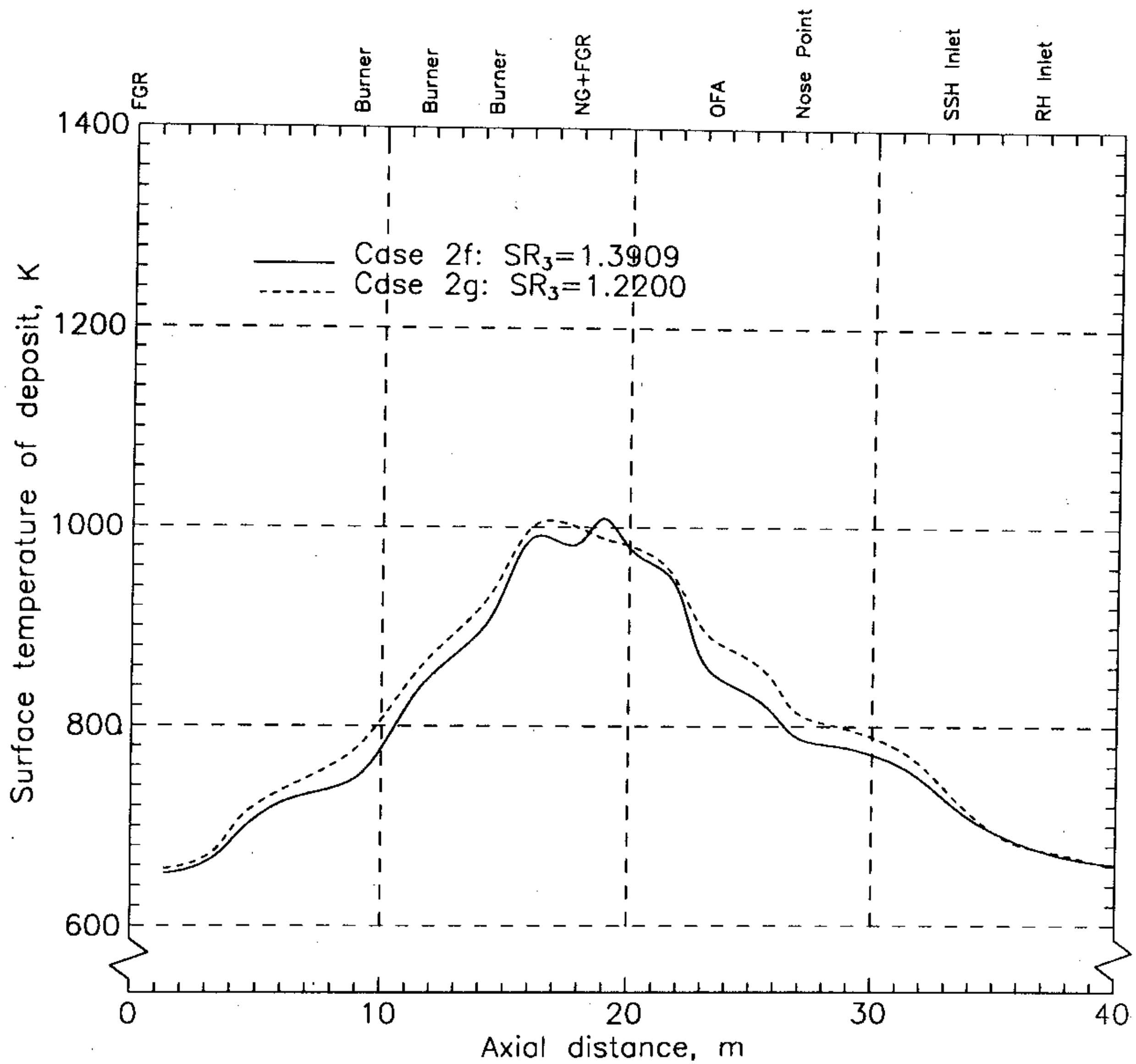


Figure 6–26. Impacts of SR_3 on surface temperatures of deposit at 50% load while 30% FGR is introduced into hopper bottom.

TABLE 6-14. IMPACTS OF SR_3 ON BOILER PERFORMANCE AT 50% LOAD WHILE 30% FGR IS INTRODUCED INTO HOPPER BOTTOM.

Case Number	Case 2f	Case 2g
Case Definition	GR SR_3=1.3909	GR SR_3=1.2200
Exit Gas Temperatures(K) of		
RH(rear)	892	888
PSH	660	659
Economizer	644	627
Air Preheater	392	384
Steam Flow(kg/s)		
Main Steam	62.74	65.11
RH Steam	52.91	53.87
Attemperation flow(kg/s)		
SH	8.654	6.613
RH	1.083	0.081
Water/steam temperatures(K)		
Economizer Inlet	484	484
Economizer Outlet	506	502
PSH Inlet	606	606
PSH Outlet	772	740
SSH Attemp. Outlet	665	667
SSH Outlet	803	803
RH Attemp. Outlet	546	564
RH Outlet	806	806
Heat absorptions(kW)		
Furnace	89549	97841
Economizer	5228	4664
PSH	36951	34784
SSH	26056	26454
RH	29865	28237
Unburned Fixed Carbon (% of Total Fixed Carbon Input)	3.14	2.90

TABLE 6-15. IMPACTS OF SR_3 ON BOILER EFFICIENCY, BASED ON ASME HEAT LOSS METHOD AT 50% LOAD WHILE 30% FGR IS INTRODUCED INTO HOPPER BOTTOM.

Case Number	Case 2f	Case 2g
Case Definition	GR SR_3 = 1.3909	GR SR_3=1.2200
Heat Loss due to Dry Gas	4.1706	3.3437
Heat Loss due to Moisture in Fuel	1.2262	1.2190
Heat Loss due to H_2O from Combustion of H_2	5.6797	5.6464
Heat Loss due to Combustible in Refuse	1.4204	1.3118
Heat Loss due to Radiation*	0.4500	0.4500
Unmeasured Losses*	1.5000	1.5000
Total Heat Losses	14.4468	13.4709
Boiler Efficiency	85.5532	86.5291

* Use the value reported in the boiler design performance data sheet.

data for coal firing at full load. Case studies were then conducted to evaluate the impacts of GR and LNB operation on boiler performance at full and low loads.

The results of the performance models are expected to correctly simulate trends and changes in boiler performance although the exact magnitudes of any changes may not be accurately predicted due to assumptions made in the modeling work. The model results should therefore be viewed as predicting trends which can be used to assess the potential changes in boiler performance upon GR and LNB application rather than being viewed as predicting the exact magnitude of changes in boiler operation. This assessment then allows corrective or preventative action to be included into the design and retrofit.

Overall the effect of installing a gas reburning system and low NO_x burners will be minor. The results of the performance models indicate that operating the boiler at GR conditions with LNB is expected to increase attemperation flows and slightly increase the carbon content of the ash. Slightly increasing the boiler thermal input would be required to maintain steam generation at full capacity while the boiler is fired with GR at full load. The proposed approach for applying GR at low load with reduced overall excess air may actually increase steam generation and reduce the main and reheat attemperation requirements. It is recommended that the current main and reheat steam temperature control requirements and attemperation usage be closely evaluated to determine whether sufficient capacity is available.