OPTIMIZATION OF COAL PARTICLE FLOW PATTERNS IN LOW NO_x BURNERS

FINAL REPORT

Due December 28, 2000

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Submitted to

DOCUMENT CONTROL CENTER,
U.S. Department of Energy
University Coal Research Program
Pittsburgh Energy Research Center
P.O. Box 10940, MS 921-0940
Pittsburgh, PA 15236-0940.

Under Contract DE-FG26-97FT97269

September 4, 2001

ABSTRACT

It is well understood that the stability of axial diffusion flames is dependent on the mixing behavior of the fuel and combustion air streams. Combustion aerodynamic texts typically describe flame stability and transitions from laminar diffusion flames to fully developed turbulent flames as a function of increasing jet velocity. Turbulent diffusion flame stability is greatly influenced by recirculation eddies that transport hot combustion gases back to the burner nozzle. This recirculation enhances mixing and heats the incoming gas streams. Models describing these recirculation eddies utilize conservation of momentum and mass assumptions. Increasing the mass flow rate of either fuel or combustion air increases both the jet velocity and momentum for a fixed burner configuration. Thus, differentiating between gas velocity and momentum is important when evaluating flame stability under various operating conditions.

The research efforts described herein are part of an ongoing project directed at evaluating the effect of flame aerodynamics on NO_x emissions from coal fired burners in a systematic manner. This research includes both experimental and modeling efforts being performed at the University of Arizona in collaboration with Purdue University. The objective of this effort is to develop rational design tools for optimizing low NO_x burners. Experimental studies include both cold- and hot-flow evaluations of the following parameters: primary and secondary inlet air velocity, coal concentration in the primary air, coal particle size distribution and flame holder geometry. Hot-flow experiments will also evaluate the effect of wall temperature on burner performance.

Cold-flow experiments were conducted at Purdue University using a 0.45 meter square Plexiglas down-flow flow visualization chamber. The 1 meter tall chamber was equipped with X-Y-Z traversing laser doppler velocimetry/phase doppler particle analyzer (LDV/PDPA). The overall purpose of the cold-flow research is to investigate the gas-solid interactions and particle motion in an isothermal coaxial jet as a function of various inlet parameters including velocity ratio and of particle properties: particle size and the particle size distribution. By utilizing a bimodal mixture of large and small particles, we can more closely approximate the flow characteristics of pulverized coal streams, and enables us to study the possible changes in the particle behavior of different sizes in the presence one another. Lagrangian particle tracking was employed to help verify the proposed mechanism for particle motion and gas-solid interaction for different velocity ratios.

A computational study using the Fluent 4.5 CFD package was performed to evaluate the predictive capability of the standard k- ϵ turbulent model for the coaxial jet flow behavior and to calculate particle trajectories and compare these to the cold-flow results. The standard k- ϵ model relates the Reynolds stress using the Boussinesq eddy viscosity assumption. While, the Boussinesq eddy-viscosity model defines the Reynolds stress to be proportional to the mean velocity gradient, it lacks a way to distribute the turbulent kinetic energy among the different components; hence, it does not predict anisotropic turbulence. The ability of the model to predict anisotropic jet behavior is however, important since the radial gas fluctuating velocity, v_g' may be responsible for the radial motion of particles.

The trajectory calculations were performed to probe the factors affecting the radial and axial motion of particles. By comparing the calculation using between deterministic separated flow

(DSF) and stochastic separated flow (SSF) methods, the effect of the gas turbulence on the particle motion can be explored. The initial axial and radial location of particles was 1 mm downstream of the inlet and 2.54×10^{-4} m from the pipe wall. The initial conditions boundary conditions at the inlet were matched with the cold-flow experimental results. A no-slip boundary condition was applied at the solid wall and the wall function was used to approximate the near wall solution. A zero-gradient was applied for the outlet boundary condition. The calculation used 40x500 grid points in the radial and axial direction, respectively. This grid resolution yielded a grid-independent solution. The radial grid is clustered along all solid walls and stretched between the solid walls for the whole length.

The hot-flow experimental work was conducted at the University of Arizona in a 2M down-fired coal furnace. The near-flame furnace design includes electrically heated walls, full length quartz windows for flow visualization studies and emissions sampling and analysis capabilities. The use of a novel variable sleeve burner allows the investigators to independently evaluate the impact of these two parameters on heat recirculation, air/fuel mixing and the resultant flame stability. Experimental data for Type 0 axial diffusion flames with no swirl will be compared with cold-flow studies and Fluent CFD Modeling results.

The major conclusions that have been reached are as follows. Cold flow studies show that coal particles do not follow the gas flow and fine coal particles tend to concentrate at the outside of the jet. The location of these fines is important since they play a role in flame ignition and attachment. A test hot flow facility has been constructed and preliminary data on flame attachment has been obtained. Wall temperature, which was controllable through implanted heating elements, played a major role on flame stability. Future work will examine the separate effects of oxygen partial pressure in the transport fluid, of coal particle size, and of primary and secondary jet momenta.

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NOMENCLATURE

C_D	Drag coefficient
CRW	Continuous Random Walk
d_o	Diameter of annular jet
d_i	Diameter of central jet
d_p	Particle diameter
DSF	Discrete Separated Flow
DRW	Discrete Random Walk
F_D	Drag force
F_{vm}	Virtual mass force
g	Gravity acceleration
k	Turbulent kinetic energy
r	Radial direction
Re	Reynolds number
Re_p	Particle Reynolds number
SSF	Stochastic separated flow
St	Stoke number
T_L	Integral Lagrangian time constant
T_L^*	The modified integral Lagrangian time constant
t	Time
U_{gi}	Centerline gas-phase axial velocity at the inlet
U_{go}	Maximum gas-phase axial velocity of annular jet at the inlet
U_s	Slip Velocity
U	Axial mean velocity
U_e	Eddy's velocity scale
u	Instantaneous velocity'
$\overline{u'}$	Axial root mean square velocity
$\overline{u' v'}$	Reynolds stress
V	Radial mean velocity
v	$V + v'$
v'	Radial root mean square velocity
w'	Angular root mean square velocity
x	Axial direction
ε	Rate of energy dissipation
τ	Characteristic time
τ^*	Modified time scale
τ_e	Eddy time scale
μ	Viscosity

1.0 INTRODUCTION

It is well understood that the stability of axial diffusion flames is dependent on the mixing behavior of the fuel and combustion air streams. Combustion aerodynamic texts typically describe flame stability and transitions from laminar diffusion flames to fully developed turbulent flames as a function of increasing jet velocity (1). In practical flames, turbulent diffusion flame stability is greatly influenced by recirculation eddies that transport hot combustion gases back to the burner nozzle. Recirculation can be of two types-internal and external. Internal recirculation is caused by secondary air swirl or the wake behind a bluff body and is not discussed here. Rather this analysis focuses on external recirculation in pure Type 0 flames. Both types of recirculation enhance mixing and heats the incoming gas streams. Models describing these recirculation eddies utilize conservation of momentum and mass assumptions. Increasing the mass flow rate of either fuel or combustion air increases both the jet velocity and momentum for a fixed burner configuration. Thus, differentiating between gas velocity and momentum is important when evaluating flame stability under various operating conditions.

In general, flame stabilization can include operating with the flame attached to the burner or with the flame front at some stable distance from the burner tip. Although typical methods of stabilizing detached flames in both natural gas and coal combustors include inducing swirl, modification of the burner quarrel, or the addition of angled nozzles or flame holders (3,7,8), this research focuses on the effects of changes in the velocity, momentum or wall temperature on the stability of axial non-swirl flames in a systematic manner.

The effect of mixing and flame attachment on pollutant formation, specifically NO_x emissions has been widely reported (3-7). Researchers have shown that the resultant NO_x emissions are highly dependent on flame attachment or the extent of pre-mixedness for detached flames, both of which are dependent on burner geometry (4-8). The resulting pollutant emissions greatly depend on flame position. This is especially true for combustion fuels containing significant amounts of nitrogen, such as coal and fuel oil. NO_x emissions from combusting these types of fuels involves a complex series of interconnected reaction pathways including both thermal (Zeldovich) and prompt (Fenimore) NO_x formation mechanisms. Figure 1 displays the reaction pathways for NO_x formation. Approximately, 50-90% of NO_x emission in coal combustion is produced during devolatilization. Therefore, methods for reducing the NO_x emission have focused on the volatile combustion characteristics.

While a detached flame produces less Thermal NO_x due to reduced flame temperatures (7), the NO_x emissions attributed to fuel nitrogen increases due to the increased oxidation of these nitrogenous species, specifically volatile nitrogen compounds (9). IFRF data obtained from European cement kilns shows that the single most important parameter for predicting NO_x emissions from pulverized coal Type 0 Axial diffusion flames (typical of cement kilns) was the flame standoff distance from the burner (10).

Figure 1 also shows that NO_x formation is preferred over N_2 formation when devolatilization occurs in a fuel lean environment. NO_x formation during devolatilization has been shown to be both kinetic and mixing limited (11). The dependence of NO_x formation on the local oxygen concentration suggests that combustion aerodynamics are important in controlling NO_x

emissions. Aerodynamics controls both the mixing between the fuel and the oxidizer and particle trajectories. In particular, near field aerodynamics have the greatest effect because coal devolatilization occurs in this region. The near field aerodynamics is controlled by jet inlet parameters, which for coaxial jet burners include:

1. Diameter ratio of the annular section to central nozzle (d_o/d_i)
2. Velocity ratio of the annular jet to the central jet in a coaxial jet ($VR = U_{go}/U_{gi}$)
3. Magnitude of the annular and central jet velocities
4. Swirl number
5. Burner geometry

Because these inlet parameters affect mixing between the central and annular jets, they control the partial pressure of oxygen in the near field region.. In addition, the jet inlet characteristics determine the fluid flow behavior, which affects the particle motion through inter-phase forces such as drag. Particle trajectories in the near field will determine whether the coal particles will travel through the rich oxygen environment near the fuel and oxidizer interface or remain in the fuel rich region of the central jet. The physical properties of the coal particles, including density, shape, particle size and size distribution also affect trajectories. During devolatilization, particle density decreases and particles swell (12). These physical changes affect the gas-solid interaction, hence changing the particle trajectories and resultant NO_x emissions.

Particle size has also been shown to affect NO_x emissions, which tends to increase with decreasing particle size (13, 14, 15). It was also observed that the smaller particles were transported radially outward towards the high temperature zone by the turbulent eddies in the gas phase while the large particles resisted turbulent dispersion and remained confined within the cooler zones of the central jet (13). It is proposed that the larger dispersion of the smaller particles increase the mixing between the combustion air and fuel streams. This aerodynamic effect when combined with the faster heatup of the smaller particles increases volatile combustion in oxygen-rich atmospheres producing higher NO_x emissions. The observed combustion behavior and emission rates were due to the interaction between two factors: the local oxygen partial pressure and particle spacing.

The interactions between the two concentric jets are also affected by the inlet parameters of both jets. And while the gas flow behavior in a coaxial jet is similar to that of a single jet far from the entrance, the near field region in a coaxial jet has a complex flow behavior because of the presence of the inner and outer shear layers. The inner shear layer is formed by the velocity gradient between the annular and central jets, and the outer shear layer is formed by the velocity gradient between the annular jet and the ambient fluid. The inner shear layer is responsible for the turbulent mixing between fuel and oxidizer in combustor flows. The turbulent mixing will be more intense as the velocity gradient increases. Varying the velocity ratio between the central and annular jets, varies the mixing intensity, thereby changing the local oxygen concentration and particle motion. These changes can significantly affect NO_x emissions. Many isothermal single-phase experimental investigations have been performed to understand the details of the flow pattern in the near field region for a coaxial jet. Recently, Buresti *et al.* has investigated the

mean and turbulent gas behavior in a coaxial jet using laser Doppler velocimetry (LDV) for velocity ratios of 3.3 and of 1.5 (16). Their observation in the gas fluctuating velocities and the Reynolds stress shows that the two variables increased with an increase in the velocity ratio. This increase verifies the higher turbulent mixing for velocity ratios greater than 1.

Turbulent mixing in the inner shear layer of a coaxial jet occurs due to the role of turbulent eddies. Many flow visualization experiments have been undertaken to understand the behavior of the turbulent eddies in the inner shear layer. Dahm *et al.* performed flow visualization experiments in the near field region of a coaxial jet for different velocity ratios and velocity magnitudes using laser-induced-fluorescence (LIF) technique with still photographs (17). Two of the velocity ratios they investigated, 1 and 2.56, are of particular interest because they are similar to the operating conditions in the research discussed here. For the velocity ratio of 1, the velocity jump between the two jets was zero across the inner shear layer. The zero velocity jump forms a wake-like instability that never rolls up to form its own distinct vortical structures before interacting with the outer shear layer. For the velocity ratio of 2.56, the inner shear layer rolled up into a shear layer-like vortex, which developed quite well before interacting with the outer shear layer. Flow visualization demonstrated that a high level of interaction exists between the vortical structures of the inner and outer shear layers. They observed that the vortex structure and dynamics in the near field of a coaxial jet depended not only on the velocity ratio, but also on the absolute velocities of the two streams. The behavior of vortical structures is crucial in the particle cross-stream dispersion.

There are not many experimental investigations of two-phase coaxial jets in the literature. One of them is by Mostafa *et al.* (18). They investigated the particle behavior in a coaxial jet with the velocity ratio of 1 and found that the particle motion in a coaxial jet was different from that in a single jet. The presence of the annular jet produces a more-uniform axial mean particle velocity across the central jet. It is important to note that they observed a narrower particle number density profile in coaxial jets than in single jets. The reduction in particle number density is due to the entrainment and mixing of additional air from the annular jet. Their investigation did not cover the effect of the annular jet on the cross-stream particle dispersion. The cross-stream motion of particles is necessary to determine radial particle dispersion. The radial particle dispersion defined here is the same as particle spreading, which is quantified according to the difference between a particle's radial position at a given axial location and its initial radial position. The change in particle dispersion will change the distribution particle number density and will affect the NO_x formation during the devolatilization (13).

It has been well documented, first by Crowe and Champagne (19), that large-scale vortical structures exist in a single-phase round jet because of the Kelvin-Helmholtz instability. These vortical structures dominate a single-phase jet up to downstream axial location of $x/d_i = 4$, where d_i is the diameter of the jet nozzle. These turbulent structures may affect and possibly enhance the particle dispersion in turbulent jets. A conceptual model of particle dispersion in the shear layer flow is the entrapment of particles in the circulating velocity fields of the turbulent structures and the subsequent centrifuging of the particles beyond these structures. The experimental results in air turbulent round jets (20) and in mixing layers (21, 22, 23) have verified the enhancement of particle dispersion exceeding the fluid dispersion. In a coaxial jet, the presence of both inner and outer shear layer may change the interaction between particle and

turbulent eddies from that observed in round jet cases. Crowe *et al.* established criteria under which the turbulent structures play a dominant role in particle dispersion (24). They proposed the use of the Stokes number (St) which is the ratio of particle time scale, τ_p to the fluid time scale, τ_f to assess the importance of these structures. The particle response time is calculated by assuming Stokes flow

$$\tau_p = \frac{\rho_p d_p^2}{18\mu} \quad (1)$$

Where ρ_p is the particle density, μ is the fluid viscosity, and d_p is particle diameter. A fluid time scale for the turbulent structures is calculated as follows

$$\tau_f = \frac{L_e}{U_e} \quad (2)$$

Where L_e is the eddy length scale and U_e is the eddy velocity scale. Based on the value of particle St, the flow behavior of the particles can be categorized as follows

- St $\ll 1$, the particle velocities will be nearly in equilibrium with the fluid velocity, thus the particles will disperse in the same way as the fluid.
- St $\gg 1$, the large structures will not have sufficient time to influence the particle trajectories,
- St ≈ 1 , the particles may be trapped in the vortices and flung beyond these structures. This effect has been verified by many experimental investigations as mentioned previously.

The difficulty in calculating the Stokes number is to approximate the correct fluid time scale, τ_f . Table 1 lists L_e and U_e that are used in the literature to calculate the fluid time scale. Most of the approximations have been used to specify the fluid time scale at far field region of the jet. Only Longmire and Eaton have performed the near field investigation of a particle laden round jet (25). The fluid length scale is approximated by measuring the distance between two adjacent vortical structures (obtained from flow visualization) and the velocity scale is approximated by half of the centerline axial mean gas velocity.

The particle Stokes number defined previously is defined for Stokes flow conditions. Therefore, it is valid when the particle slip velocity is small. When there is a significant slip velocity, the time interaction between particles and the turbulent structures will be reduced, hence increasing the particle Stokes number (26).

Detailed measurements of the gas and particle flow variables in a combustor are needed to accurately model particle trajectories in pulverized coal combustion and thus be able to develop a strategy to reduce NO_x emissions. Unfortunately, the high temperature environment and the combustion products in a conventional hot-flow furnace complicate flow measurements using conventional methods such as hot wire anemometry or pitot tubes. The hot wire anemometer cannot not sustain the high temperatures are easily coated by the combustion products. Pitot tubes cannot take account flow reversal. Recently, several researchers have confirmed that they could make flow parameter measurements of the gas by using LDV (27, 28). They reported that the small coal particles follow the detailed motion of gas in their hot flow system. They did not

report any flow measurements of the large coal particles. The property changes of coal particles interfere with Phase Doppler Anemometer Analyzer (PDPA) measurements since PDPA requires spherical particles with constant index of refraction for size discrimination.

Simultaneous changes in the size, shape, and density of the coal particles during devolatilization complicate the study of particle flow behavior since the changes are also accompanied by changes in the gas-solid interaction. Therefore, in order to have a good understanding in the NO_x formation in the near-field region, firstly, we need to understand the effect of particle size, particle diameter, particle size distribution, and particle shape on gas-solid interaction under cold-flow conditions. Understanding how the inlet parameters and particle properties affect the gas-solid interaction separately in cold flow will help us in studying the motion of particles in real combustion environments, as devolatilization occurs and changes both the particle properties and the gas flow behavior.

The overall purpose of the cold-flow research is to investigate the gas-solid interactions and particle motion in an isothermal coaxial jet as a function of various inlet parameters including velocity ratio and of particle properties: particle size and the particle size distribution. By utilizing a bimodal mixture of large and small particles, we can more closely approximate the flow characteristics of pulverized coal streams, and enables us to study the possible changes in the particle behavior of different sizes in the presence one another. This report will summarize only the recent accomplishment in studying the role of velocity ratio on gas-solid interaction and on particle motion. In addition, complementary computational studies using the commercial computational fluid dynamic (CFD) package Fluent 4.5 are compared to the experimental results. Lagrangian particle tracking was employed to help verify the proposed mechanism for particle motion and gas-solid interaction for different velocity ratios.

In order to use the experimental results to optimize the design and operation of pulverized coal combustors, computational studies are needed to capture the correct physics of the gas and the particles. In coal combustion, the average mass loading of coal particles is about 0.5 and this will corresponds to a volume fraction of 10^{-4} . The low volume fraction will minimize the role of particle-particle interaction in determining the particle motion. For this reason, a Lagrangian model was employed for this research. This method is also known as the trajectory or tracking method. In this approach, the gas-phase flow field (single-phase solution) is needed to calculate the particle trajectories. Two types of modeling studies exist based on the single-phase solution, the time averaged and the time-dependent methods. The time-averaged method treats the flow as a steady flow. The k - ϵ and Reynolds stress turbulence models are commonly used to obtain the single-phase solution in the time-averaged methods. The time-dependent models simulate the instantaneous eddies, and particle trajectories are predicted in this time-dependent field. These models include Discrete Vortex Element and Large Eddy Simulation models. Additional source terms are needed to account for the presence of the particles and the ability of the models to represent these effects correctly is very crucial since the particle motion depends solely on the gas-phase “seen” by the particles. In this research, the time –averaged method will be used.

The time-averaged methods have been under extensive development in the past three decades. Faeth has categorized the time-averaged methods into two categories that identify the approach used to treat the interphase transport rates (29). The first category is the locally homogeneous

flow (LHF) analysis, where interphase transport rates are assumed to be infinitely fast. This model assumes that the particle-phase is in dynamic and thermodynamic equilibrium with the gas-phase. The second category is the separated flow analysis where finite interfacial transport rates are considered. The separated flow analysis is divided into two categories: (1) deterministic separated flow (DSF) analysis, where dispersed-phase/turbulent interactions are ignored and (2) stochastic separated flow (SSF) analysis, where dispersed-phase/turbulence interactions are considered using a random walk computation. In the SSF analysis, the particles are assumed to interact with a succession of energy containing eddies that have statistically independent properties. Properties inside the eddy are assumed to be uniform, but to change in a random manner from eddy to eddy. Eddy properties are approximated from the calculated turbulent kinetic energy of the single-phase flow. Faeth concluded that the DSF are deficient for quantitative estimates of the dispersed-phase flow structure, but the SSF model is very promising based on results for variety of particle-laden jets.

The research efforts described herein are directed at evaluating the effect of flame aerodynamics on NO_x emissions from coal fired burners in a systematic manner. This research includes both experimental and modeling efforts being performed at the University of Arizona in collaboration with Purdue University. The objective of this effort is to develop rational design tools for optimizing low NO_x burners to the kinetic emissions limit (below 0.2 lb./MMBTU). Experimental studies include both cold- and hot-flow evaluations of the following parameters: primary and secondary inlet air velocity, coal concentration in the primary air, coal particle size distribution and flame holder geometry. Hot-flow experiments will also evaluate the effect of wall temperature on burner performance.

As the scope of this project entails three distinct components-CFD modeling, cold-flow and hot-flow experiments, each will be described independently. The Cold-Flow Studies will be followed by Modeling, followed by the Hot-Flow Studies. Each Chapter includes materials & methods and results & discussion details.

2.0 COLD FLOW STUDIES

2.1 Materials and Methods

The two-phase system utilized for this study is an axisymmetric particle-laden coaxial jet. The spherical glass beads particles used in the experiments have a nominal diameter of 70 microns with 80% between 63 to 85 microns. Glass beads with a nominal diameter of 4-5 microns are used as the seed particles for the LDV measurements of the gas phase. The specific gravity of both particles and seeds is 2.5. The schematic of the coaxial jet flow apparatus is shown in Figure 2. The airflow to the central nozzle is mixed with the particles stored in the hopper through a venturi-type inductor and with the seed through a reverse cyclone. The central nozzle is a 14.22 mm (0.56 inch) ID copper pipe with a wall thickness of 1.65 mm (0.65 inch). The length to diameter ratio for the central nozzle is 100 to ensure a fully developed, turbulent pipe flow at the pipe exit. The flow to the annular nozzle is mixed with the seed particles before it is discharge to the 0.46 x 0.46 m (18x18 inches) chamber. The inner diameter ratio between the annular and the central nozzle is 2.24. Flow straighteners in the form of perforated plates are inserted in order to eliminate any swirl in the annular section. Figure 3 displays a detailed schematic of the central and annular section. The particles are collected at the bottom of the chamber and weighed with a load cell so that the mass flow rate of particles can be monitored.

2.2 Laser Doppler Velocimetry / Phase Doppler Particle Analyzer (LDV/PDPA)

A two-component laser Doppler velocimetry / phase Doppler particle analyzer (LDV/PDPA) was used to make measurements of the mean and fluctuating velocity of particles, as well as the particle size. The LDV/PDPA system is placed on a X-Y-Z traversing system with a 10^{-4} inch accuracy. A 5-W argon ion laser, with blue and green beam wavelengths of 488 and 514 nm, respectively, is used as the light source. The system uses a Bragg cell unit that created a 40 MHz frequency shift for one of each of the color beams to avoid directional ambiguity. The size of the optical measurement volume is 100 x 233 x 233 μm .

For the single-phase measurements, the high voltage of the photomultiplier tube (PMTs) was set at 350 V and the burst threshold was set at 0.05 mV for the seed particles. To measure the 70 μm particles in two-phase flow, the PMTs were set at 250 V with a burst threshold of 1 mV. A minimum of 1,000 coincident samples is collected for both phases to minimize the statistical error. The coincidence mode is employed in order to ensure that both the burst signal from the green and blue beams originate from the same particle. Hardalupas *et al.* stated that the random errors for the mean axial velocity, the axial and radial fluctuating velocities are 1%, 3%, and 5%, respectively (30). The manufacturer of the LDV/PDPA system reported that the random error in the particle size measurement is 1%.

2.3 Experimental Flow Conditions

The central jet was loaded with spherical glass beads at a mass loading, m of 0.5. The solid loading m is defined as the mass flow rate ratio of particles to air. Three velocity ratios of 0, 1, and 1.8 were undertaken to investigate the effect of the annular jet on the particle motion. The Reynolds number for the central jet at the nozzle exit was maintained constant at 8,300

and the annular flow velocity was varied. Table 2 lists the experimental conditions for the single-phase and two-phase flows.

2.4 Experimental Results and Discussions

2.4.1 Particle Stokes number (St)

Table 3 lists the particle St for different axial positions for the three velocity ratios. For simplicity, the fluid time scale, τ_f , is calculated based on the mean flow properties. The length scale of the fluid is taken as the half-jet width, and the velocity scale of the fluid is the centerline axial mean gas velocity.

It is important to evaluate the particle St in order to investigate the effect of the gas turbulence on the particle fluctuation velocities and the particle radial motion. The particle St at the nozzle exit is greater than 10, indicating that the particle is unresponsive to gas turbulence. By the axial location, $x/d_i = 10$, the St is on the order of ten, still corresponding to unresponsive particles. Based on this value of the particle St, the particle motion in the measurement distance will not be affected by the turbulent eddies.

2.4.2 Nozzle Exit Measurements

The nozzle exit measurements were taken at distance of 1 mm downstream of the nozzle exit. Figure 4a displays the radial profiles of the axial mean gas velocity, U_g and the axial mean particle velocity, U_p at the nozzle exit. The profile of the axial mean gas velocity represents the 1/7 power law profile of turbulent pipe flow. The axial mean particle velocity has a flat profile across the pipe diameter with particle slip condition on the pipe wall. The particles in the pipe interact with the gas through a drag force, causing the presence of a slip velocity, $U_s = U_g - U_p$. The mean slip velocity is 2.8 m/s in the pipe centerline and - 8.5 m/s on the pipe wall.

The radial motions of particles and gas at the nozzle exit are displayed in Figure 4b. The radial mean particle velocity, V_p is about zero across the central jet diameter, but it increases significantly near the wall. This may be due to the interaction between the particles and turbulent eddies and due to the lift force in the wall boundary layer. It is interesting to note that particles move radially with a higher radial velocity than that of the gas even though the particle St is higher than 10.

The radial variations in the axial and radial fluctuating velocities for both gas and particles are displayed in Figures 4c and 4d, respectively. The particles have higher axial and radial fluctuating velocities than those of the gas near the pipe centerline. This behavior has been observed previously (30). This may be due to the particle-particle collisions in the pipe centerline because the particle volume fraction has a normal distribution in the fully developed pipe.

2.4.3 Axial Mean Gas and Particle Velocity

Figures 5a and 5b show the experimental data of the axial mean velocities for both gas and particles at axial locations, x/d_i , of 3 and of 10, respectively. Comparing the profiles of the axial mean gas velocity for different velocity ratios, it increases with an increase in the velocity ratio. When the velocity ratio is 1 or larger, the annular jet fluid will move inward and transfer its mass and its momentum to the central jet fluid through the inner shear layer. The higher the velocity ratio, the more momentum will be transferred to the central jet. The mixing process will increase the volumetric flow rate in the region where particles are present and will influence the volume fraction profile.

Due to the change in the gas velocity, the gas-solid interactions will also change. As the velocity ratio increases, the gas will accelerate the particle axial mean velocity with a higher drag force. The gas will stop accelerating the particles when both the magnitudes of the axial mean gas velocity and the axial mean particle velocity are the same. Then, as the particle moves faster than the gas as shown by the case with velocity ratio of 0 at x/d_i of 10, the particle will accelerate the gas.

The increase in the velocity ratio will influence two things that are important for NO_x emissions. First, the velocity ratio increase will increase the partial pressure of oxygen in the fluid surrounding the particles, which increases the stoichiometric ratio as the particle mass flow rate is the same for all velocity ratios, assuming no particle loss. Second, the increase in the velocity ratio will increase particle axial mean velocity, therefore decreasing particle residence time in the devolatilization region.

2.4.4 Radial Mean Gas and Particle Velocity

Figure 6 displays the radial mean particle velocity, V_p at x/d_i of 3 and 10 and Figure 7 displays the radial mean gas velocity, V_g . The radial profile of the radial mean gas velocity increases with radial distance. The radial mean gas velocity for the case with velocity ratio of 0 is higher near the jet centerline at x/d_i of 3 than the other cases since it has a narrower jet width. By x/d_i of 10, the radial mean gas velocity increases with an increase in the velocity ratio. The radial mean particle velocity is an important parameter because it determines the radial dispersion of the particles, changing the particle concentration distribution across the flow. The radial mean particle velocity is about the same for all velocity ratios at x/d_i of 3, but at x/d_i of 10, the radial mean particle velocity is higher for the case with a higher velocity ratio.

It is a question of interest to answer how the radial mean particle velocity changes with the velocity ratio. The effect of gas turbulence should be minimum since the particle St is on the order of ten for the axial distances observed. Figure 8 shows that the axial evolution of the axial fluctuating gas and particle velocity along the jet centerline between the inlet and $x/d_i = 10$. There is no change in the axial particle fluctuating velocity although the particles are exposed to higher turbulence for the higher velocity ratio case. This verifies that gas turbulence does not affect the particle motion, as predicted by the particle Stokes number. Therefore, the only source for the observed behavior in the radial mean particle velocity is the interaction between the particles and the mean gas radial velocity. Comparing Figures 6 and 7, the radial mean velocity of the

particles is higher than that of the gas. Consequently, the gas will reduce the radial mean particle velocity via a drag force.

There are two reasons for the observed trend in the radial mean particle velocity at $x/d_i = 10$. First, the radial mean gas velocity increases with increasing velocity ratio, the drag force applied on the particles by the gas will be smaller with increasing velocity ratio. The smaller the drag force will cause less reduction on the radial mean particle velocity. Second, the particle residence time decreases with an increase in the velocity ratio. Since the drag force which is responsible for the particle radial motion is a function of the radial mean velocity difference between the particle and gas, the decrease in the particle residence time will reduce the total time used to integrate the equation of motion. And this will cause less reduction in the radial mean particle velocity with an increase in the velocity ratio.

2.4.5 Radial Particle Dispersion

The higher radial particle velocity does not always indicate a higher particle dispersion. One way to measure the radial dispersion is to measure the directional angle of the particle velocity vector, $\alpha = \tan^{-1}(V_p/U_p)$. Figure 9 displays the radial profile of α for all velocity ratios at x/d_i of 3 and 10. The plots show that at x/d_i of 3 and of 10, the directional angles are the same for all velocity ratios. The reduction in the radial mean gas velocity occurs proportionally with the increase in the axial mean particle velocity for all velocity ratios. The result indicates that the particle dispersion is the same for all velocity ratios. This result is important because it will tell us that there is no change in the radial distribution of the particle number density for the flow condition and particle properties used in this research. Thus, one would not expect a change in the NO_x emissions due to the particle spacing as shown by Masutani et al (13).

Figure 10 compares the directional angel of velocity vector between the gas and the particle at $x/d_i = 3$ and 10. For velocity ratio of 0, the directional angle for the gas is larger than the particles, indicating that the radial dispersion of gas is larger than the particles. This trend is verified by the calculation of a particle Stokes number larger than 10. This result also confirms the use of the half jet width and the gas centerline velocity as the length and velocity scales of the fluid in calculating particle Stokes number. For velocity ratios of 1 and 1.8, it seems that the radial dispersion of particle is higher than the gas. This is not caused by the ejection of particles by the large eddies structures because the particle Stokes number is larger than 10 for both velocity ratios. This behavior is contributed by the presence of the annular stream that prohibits the expansion of the central jet and by the initial radial velocity of the particles.

2.4.6 Gas Turbulence

Figure 11 displays the axial and radial fluctuating gas velocities, u_g' and v_g' at x/d_i of 3 and of 10 for the three velocity ratios. As the velocity ratio increases, the turbulent mixing in the outer and inner shear layers becomes more intense due to the increase in the gradient of the mean axial gas velocity. This is shown by the increase in the radial and

axial fluctuating gas velocities. The presence of the inner shear layer is shown by a local maximum in the axial fluctuating gas velocity for the velocity ratio of 1.8 (Figure 9a). The absence of local maximum in the axial fluctuating gas velocity in the inner shear layer for the velocity ratio of 1 is due to the zero velocity jump between the annular and the central jets. This indicates that the case with velocity ratio of 1 behaves in a similar way to the velocity ratio of 1, but with the jet diameter equal to the annular diameter.

2.4.7 Particle Turbulence

It is desired to understand the mechanism for the generation of the particle turbulence when we want to use the two-fluid model. Figure 12 displays the axial and radial fluctuating particle velocities, u_p' and v_p' with the 5% error bars at x/d_i of 3 and of 10. It is interesting to note that the axial fluctuating particle velocity behaves differently from the radial fluctuating particle velocity for different velocity ratios.

At x/d_i of 10, the case with velocity ratio of 0 has the highest value of axial fluctuating particle velocity followed by the case with velocity ratio of 1.8 and 1, respectively. By considering this trend and the particle St number, it is not expected that the gas turbulence plays a role in generating particle turbulence, rather that the particle turbulence is generated by the gradient of the axial mean particle velocity. As the gradient of the axial mean particle velocity increases, the axial fluctuating particle velocity also increases, similar to the gas turbulence. The dependency of the axial fluctuating particle velocity on the gradient of the axial mean particle velocity has been observed previously in particle-laden jet experiments (31) and mixing layer experiments (21, 32) have also explained the generation of particle turbulence by the gradient particle velocity using the fan spreading theory. The fan spreading theory states that particles arriving at one point in a downstream location originate from different points upstream and bring with them axial mean velocity information. The difference in the axial mean particle velocity at different radial positions will contribute to a higher value of the axial fluctuating particle velocity.

The profile of radial fluctuating particle velocity is different from that of the axial fluctuating particle velocity. At x/d_i of 10, the radial fluctuating particle velocity increases with an increase in the velocity ratio, similar to the gas turbulence. This behavior is not due to gas turbulence, but it is due to the generation by the gradient of the radial mean particle velocity (Figure 6). At x/d_i of 3, the radial fluctuating velocities are the same since the radial mean particle velocities are the same for all velocity ratios.

Anisotropic behavior is observed in the particle turbulence in which the axial fluctuating particle velocity is higher than the radial one. One source for this difference is due to the difference in the magnitudes variation between the axial mean particle velocity and the radial mean particle velocity.

2.4.8 Gas Reynolds Stress

The Reynolds stress $\overline{u_g' v_g'}$, of the gas-phase is the time-averaged rate of momentum transfer due to turbulence and is correlated to the turbulent production by multiplying it with the gradient of the axial mean velocity. The radial profiles of the normalized Reynolds stress at x/d_i of 3 and of 10 are displayed in Figures 13a and 13b. At $x/d_i = 3$, the local maximum for velocity ratio of 1.8 shows the more intense of turbulent mixing than the velocity ratio of 1. The Reynolds stress in the outer shear layer increases with an increase in the velocity ratio. This indicates that the gas Reynolds stress is related to the gradient in the axial mean gas velocity. Moreover, the production of gas turbulence will follow the same pattern.

2.4.9 Particle Reynolds Stress

Figure 14 displays the radial profiles of the particle Reynolds stress, $\overline{u_p' v_p'}$, for all velocity ratios at x/d_i of 3 and 10. The shapes of the radial profiles are different for different velocity ratios. There is a change in the sign between the case with velocity ratio of 0 and the other cases and this is expected due to the change in the profile of the axial mean particle velocity. For a velocity ratio of 1.8, the gradient of the axial mean particle velocity changes from a negative (at the nozzle exit) to a positive value in the periphery of the jet centerline. Besides the sign change, the trend of the magnitude for the particle Reynolds stress is similar to the axial particle fluctuating velocity, in which the velocity ratio of 1 has the lowest magnitude. This result indicates that there is a strong relationship between the particle Reynolds stress and the gradient of the axial mean particle velocity. This information will be useful to close the particle Reynolds stress in the two-fluid model.

3.0 COMPUTATIONAL RESULTS AND DISCUSSIONS

3.1 Single-Phase Coaxial Jet

A computational study using the Fluent 4.5 CFD package was performed to evaluate the predictive capability of the standard k- ϵ turbulent model for the coaxial jet flow behavior. It is important to have an accurate solution for the single-phase flow since it will be used to calculate the particle trajectory. The Fluent 4.5 CFD package discretizes the governing equation in space using a finite volume formulation. The standard k- ϵ model relates the Reynolds stress using the Boussinesq eddy viscosity assumption. The Boussinesq eddy-viscosity model defines the Reynolds stress to be proportional to the mean velocity gradient. The model also lacks a way to distribute the turbulent kinetic energy among the different components; hence, it does not predict anisotropic turbulence. The ability of the model to predict anisotropic jet behavior is however, important since the radial gas fluctuating velocity, v_g' may be responsible for the radial motion of particles.

The boundary conditions at the inlet were matched with the experimental results. A no-slip boundary condition was applied at the solid wall and the wall function was used to approximate the near wall solution. A zero-gradient was applied for the outlet boundary condition. The calculation used 40x500 grid points in the radial and axial direction, respectively. This grid resolution shown in Figure 15 has yielded a grid-independent solution. The radial grid is clustered along all solid walls and stretched between the solid walls for the whole length.

3.1.1 Results and Discussion

Figure 16 and 17 shows the axial evolution of the axial mean gas velocity and the gas turbulence kinetic energy along jet centerline, respectively. It shows that the standard k- ϵ model overpredicts the gas turbulent kinetic energy and causes the underprediction of the axial mean gas velocity for the case with velocity ratio of 0. For the velocity ratio of 1.8, the standard k- ϵ underpredicts the gas turbulent kinetic energy along the centerline between $0 < x/d_i < 10$. Consequently, the axial mean gas is not accelerated as fast as that observed in the experiment. For the case with velocity ratio of 1, the standard k- ϵ model underpredict the axial evolution of gas turbulent kinetic energy so that the standard k- ϵ model overpredicts the axial mean gas velocity along jet centerline.

Figure 18 and 19 displays the radial profiles of the axial mean gas velocity and the turbulent kinetic energy at $x/d_i = 10$, respectively. The radial profiles agree with the observation in the axial evolution. For velocity ratio of 1 and 1.8, the standard k- ϵ model seems to have a comparable prediction in the turbulent kinetic energy in the outer radial position. This will cause the overprediction of the radial fluctuating gas velocity, which is important parameter in the Lagrangian particle tracking.

For velocity ratio of 0, the flow is a round jet and the standard k- ϵ model has been known to overpredict the spreading rate because of the overprediction in the gas turbulent kinetic energy. Wilcox noted that a round jet-spreading rate calculated by the standard k- ϵ model

is approximately 18% higher than the experimental data (33). He suggested the drastic surgery of the ε transport equation is responsible for the poor performance of the standard k- ε model. The correct form of the dissipation rate transport equation obtained from the time-averaged Navier-Stokes equation is far more complicated than the k transport equation and involves many unknown double and triple correlations of fluctuating velocity, pressure, and velocity gradient. Many researchers have suggested some modification in the standard k- ε equation to increase the dissipation rate including adding a cross diffusion term in the k transport equation (34, 35) or modifying C_μ as a function of the ratio of production of gas turbulent kinetic energy to dissipation rate (36).

The poor performance of the standard k- ε model to approximate the flow behavior in a coaxial jet is expected since the presence of the inner shear layer will add more complexity. The inner shear layer can be described similarly as mixing layer or axisymmetric wake. It has been shown that the standard k- ε model tends to underpredict the spreading rate (33, 36). This indicates that the standard k- ε model underpredicts the turbulent mixing in these flows. Since the jet centerline is in the region where the gas flow behaves as that of mixing layer or wake flow, it is the reason we observed the underprediction in the axial evolution of the gas turbulent kinetic energy. The addition of the cross diffusion term improves the predictability in the mixing layer (35). Modification of C_μ also improves the predictability of the standard k- ε model in the axisymmetric wake (36).

The other important flow variables is the radial mean gas velocity and the radial fluctuating gas velocity, the accuracy of the standard k- ε model is important to predict the radial motion of the particles. Figure 20 shows the radial profiles of the radial mean gas velocity at $x/d_i = 3$ and 10. The standard k- ε model underpredicts the radial gas mean velocity both in the inner and outer shear layer, causing the overprediction in the calculated drag force. On the other hand, due to the isotropic characteristic of the k- ε model, the standard k- ε seems to overpredict the radial fluctuating gas velocity in the outer shear layer (Not shown). This will overpredict the role of gas turbulence in the radial motion of the particles.

3.2 Particle-Phase Flow

3.2.1 Particle Trajectory

Lagrangian particle tracking simulations were performed using the FLUENT 4.5 CFD package. The trajectory of a particle is predicted by integrating the force balance on a particle

$$\frac{\partial u_p}{\partial t} = F_D (u_f - u_p) + g \frac{(\rho_p - \rho_f)}{\rho_p} + F \quad (3)$$

Where $F_D (u_f - u_p)$ is the drag force per unit particle mass and

$$F_D = \frac{18\mu}{\rho_p d_p^2} \frac{C_D Re_p}{24} \quad (4)$$

The drag coefficient, C_D , is a function of the Re_p given by

$$C_D = a_1 + \frac{a_2}{Re_p} + \frac{a_3}{Re_p^2} \quad (5)$$

Where a_1 , a_2 , and a_3 are constants that apply over a range of Re_p (37). Subscript f in this section corresponds to the fluid-phase in the presence of particle. The u_p is the instantaneous velocity of a particle. The second term on the rhs of equation (3) is the gravity force. The third term on the rhs of equation (3) represents the other forces, such as the virtual mass force, F_{vm} , and the force that arises due to the pressure gradient in the fluid, F_p

$$F_{vm} = \frac{1}{2} \left(\frac{\rho_f}{\rho_p} \right) \frac{\partial}{\partial t} (u_f - u_p) \quad (6) \text{ and } (7)$$

$$F_p = \left(\frac{\rho_f}{\rho_p} \right) u_p \frac{\partial u_i}{\partial x_i}$$

These two forces are not important for gas-solid flow since the density ratio is small. The other force that is not included in this study is the Saffman Lift force. This force is due to the pressure gradient developed on a particle due to rotation induced by a velocity gradient and can be important in the case with a steep gradient in the gas mean axial velocity.

Two different trajectory methods are considered. The first one is the deterministic-separated flow analysis, DSF, where particle-gas turbulence interactions are ignored. This model represents the fan spreading theory. The second one is the stochastic separated-flow analysis, SSF, where particle-turbulence interactions are considered using a random walk computation for the particles. The continuous Random Walk model, CRW in FLUENT 4.5 was used for the SSF analysis. Particles are assumed to interact with turbulent eddies that have statistically independent properties. Properties inside the eddy are assumed to be uniform, but to change in a random manner from eddy to eddy. Each eddy is characterized by a Gaussian distributed random velocity fluctuation u_f' , v_f' , and w_f' (angular rms velocity), and by the eddy lifetime, τ_e :

$$\tau_e \approx 2 T_L \quad (8) \text{ and } (9)$$

$$T_L \approx C_L \frac{k_f}{\varepsilon_f}$$

Where T_L is the integral Lagrangian time constant and C_L is a constant that is set to be 0.15. The effect of crossing trajectory is included in CRW. The crossing trajectory effect arises from the existence of a relative velocity between the gas-phase and particles. The relative velocity will reduce the interaction time between an eddy and particles. CRW in Fluent 4.5 uses the Csanady equation to calculate the reduction of a time interaction (38).

$$\frac{T_L^*}{T_L} = \frac{1}{\left(1 + C \frac{(U_f - U_p)^2}{k_f}\right)^{\frac{1}{2}}} \quad (10)$$

Where T_L^* is the modified fluid Lagrangian integral time, and C is a constant which is set to be 1.5.

Both the DSF and SSF methods need the flow predictions of the gas-phase in the presence of particle. Since the mass loading is low, the modulation of gas-phase by the particles is neglected. Therefore, the single-phase solution was used for the calculation of particle trajectories for all velocity ratios.

3.2.2 Results and Discussion

The trajectory calculations were performed to probe the factors affecting the radial and axial motion of particles. By comparing the calculation using between the DSF method and the SSF, the effect of the gas turbulence on the particle motion can be explored. The initial axial and radial location of particles was 1 mm downstream of the inlet and 2.54×10^{-4} m from the pipe wall. The initial conditions were matched with the particle-phase data obtained from the LDV measurements.

Figure 21 displays the comparison between the SSF and DSF predictions in the axial particle velocity, u_p for the three velocity ratios. The u is defined as the instantaneous velocity. The results show that the particle axial velocity calculated by the DSF and SSF methods is about the same, verifying that axial particle velocity is controlled mainly by the axial mean gas velocity.

Figure 22 displays the DSF predictions in the radial particle velocity, v_p for the three velocity ratios. The DSF predicts that the drag force plays an important role in reducing the particle radial velocity as is observed in the experiments. The decay rate is about the same for all velocity ratios between the nozzle exit and axial location x/d_i of 2, but they are different between x/d_i of 2 and 10 due to the difference in the velocity ratio. Qualitatively, this Lagrangian predictions agree with the proposed mechanism for the radial mean particle velocity behavior as described previously. The quantitative accuracy of the DSF prediction on the radial particle velocity depends on the accuracy of prediction in the radial mean gas velocity.

The DSF and SSF calculations for the radial particle velocity do not agree with one another. The difference between the DSF and SSF methods in predicting radial particle velocity indicates that the radial fluctuating gas velocity can play a role in the radial motion of particles. The difference in the prediction by the DSF and SSF is surprising since the particle St is larger than 10, but this may be caused by the overprediction of the radial fluctuating gas velocity.

Figure 23 displays the prediction of particle trajectories by both methods for the three velocity ratios. The DSF calculations predict that as the velocity ratio increases, the particle will travel further radially. The SSF prediction shows the same behavior as the DSF prediction by x/d_i of 10. The inaccuracies in the DSF and SSF predictions are caused by the inaccuracy in the standard k - ϵ turbulence model.

4.0 HOT FLOW STUDIES

Section 4 describes the 2M hot-flow experimental furnace and presents experimental data obtained with the 2M furnace firing natural gas. Included in the Materials and Methods section are furnace design and construction details, key features including the coaxial burner, sampling system and data acquisition system. This is followed by results of a series of natural gas experiments conducted using the 2M furnace.

A novel 2 meter tall furnace was designed and constructed for conducting the hot-flow experimental activities of this project. The 2M furnace complements two existing furnaces at the University of Arizona Combustion Research Group (CRG), a 6 meter tall, 15 cm ID down-fired unit mainly utilized for kinetic studies under Post-Flame conditions, and a medium (3 m) sized 16 cm ID unit also utilized for post-flame work. The 2M unit was specifically built to evaluate combustion characteristics in the near-flame region. It is designed to burn a wide variety of fuels including pulverized coal, liquid fuels and natural gas. The 2M furnace is integrated into the facility's compressed air, emission sampling and safety systems to allow great flexibility in experimental design and operation.

Construction of a new furnace offers unique opportunities for the project team as they are not constrained by physical limitations or shortcomings often associated with existing equipment. However, it also requires extensive planning and vision to ensure that the new furnace configuration has sufficient flexibility for future projects. In addition to incorporating the current project's need to evaluate operating conditions in the near-flame region, flame stability, and resulting NO_x emissions, major design constraints included furnace startup/shutdown requirements, controlling furnace emissions, laboratory safety and project budget. A 17 to 20 kW nominal firing rate was chosen for the new furnace to correspond with the laboratory's two existing furnaces.

4.1 Materials and Methods

4.1.1 Furnace Design

The engineering, design and construction details of building the 2M furnace are detailed in this section including determination of the furnace wall materials, specification of the furnace heaters, translating stage, exhaust, and optical window.

4.1.1.1 Design/materials of construction

To arrive at the final design, the project team first asked the following questions:

What do we want to measure and how will we do it?

What are current operational deficiencies associated with existing furnaces

Where will the furnace be located?

Answers to these questions formed the basis for the furnace design. Specifically, the goals and needs of the current project provided the fundamental design. Operation, controls and sampling must focus on the near-flame region and wall effects should be minimized. Based on these criteria, the furnace sizing was

chosen to be 18" (0.5 m) internal diameter and 3' (1 m) reaction section. The total furnace length was set at 6 ft (2 m) to allow for burnout and minimize the effects of the furnace exhaust on the flow characteristics in the reactive section.

Given the furnace dimensions, materials of construction were then evaluated. With an 0.5 m internal diameter, it was determined that the heat of combustion could not maintain sufficiently high wall temperatures to sustain the combustion process. Thus, the furnace design had to incorporate external heating, thereby greatly affecting wall material selection. Several gas-fired heater configurations and electric resistance heater designs were evaluated. Gas-fired designs included multiple firing tubes embedded in the furnace walls and a gas-fired annulus. These designs were not practical due to the heating area requirements of the large diameter furnace, the need for high temperature thermally conductive, but non-porous refractory materials separating the gas heater and the reaction zone, and the weights of these castable materials. This issue was compounded by the desire for full-length observation and sampling capabilities as described below. Externally heating the furnace electrically became the obvious solution. This allowed the furnace walls to be constructed from light-weight insulating refractory board (Thermcraft 2300BRD). Benefits of using the refractory board include:

- Lightweight construction
- Low thermal conductivity (0.22 W/m-K)
- Low Thermal Mass and negligible thermal expansion
- Board materials are readily machinable, thus repairs and modifications of the furnace can be completed easily.

The furnace configuration, depicted in Figure 24 provided the least expensive electric heater requirements, allowed for symmetrical construction and provided easy access for the observation and sampling ports. The furnace is constructed of 2 layers of 1" thick refractory board surrounded by 4 inches of Kaowool blanket insulation. The furnace is supported by an exterior unitstrut frame structure. The walls of the hot section are embedded in a 2" thick base made from the Kaowool board. The base rests on stainless steel plates loosely supported by the frame structure. The top of the furnace is constructed from a single 2" thick rigid Kaowool board and fits tightly onto the hot section walls. The cool section is pressed into the base of the hot section and is supported by a stainless steel ashtrap that is secured to the furnace frame. The use of the rigid insulation board with its extremely small thermal expansion coefficient and the sliding stainless steel plates minimizes the potential for leaks due to thermal cycling. The entire weight of the furnace is supported on the ashtrap frame which is the only furnace component bolted to the support structure. The furnace walls are painted with a coating cement (Fiberfrax QF-180) to prevent decomposition and minimize leakage into or out of the furnace. Exterior seams are taped and sealed with high temperature ceramic fabric dipped in the cement. The cool section is lined with a

Nextel 312 fabric skirt to prevent molten ash from collecting on the walls and causing damage upon removal. Construction details are presented in Figure 25.

4.1.1.2 Wall heaters

The furnace is heated electrically by ceramic heaters (Thermcraft FPH204). These heaters, 6" wide by 10" tall are mounted to the interior octagonal furnace walls by titanium brackets. Twenty four heaters are used to provide a maximum of 27 kW of heat in 3 zones. The embedded heating elements can operate continuously under oxidizing conditions at temperatures up to 1100°C. This design allows for easy replacement of individual heaters in the event of failure. Each zone is equipped with four K-type thermocouples for monitoring, control and over-temperature protection. Two dual thermocouples are utilized per zone to provide simultaneous temperature measurements for control and monitoring.

The heaters are wired for 208volt, 3 phase operation. The temperature of each zone can be controlled independently by individual 935 series controllers (Watlow) connected to a 3 phase Dinamite Power controller. (Watlow DC3C series). Three 147 series limit switches (Watlow) are connected in series with a mechanical contactor to prevent overheating the furnace. An electrical wiring diagram for the furnace heaters is presented in Figure 26.

4.1.1.3 Translating stage

In addition to four stationary ports equally spaced down the side of the hot section, the furnace is equipped with a translating sampling stage. The translating stage runs the full length of the hot section of the furnace and is located at a right angle to the stationary ports and directly opposite the optical widow described below. This configuration essentially cuts the furnace in half which complicated construction of the hot section as both sections had to be carefully positioned to mate into the furnace lid and base.

The translating stage, pictured in Figure 27 consists of a sample port welded to the center of a 6' long 3" wide stainless steel plate. This plate is sandwiched between parallel sections of angle iron mounted to the support structure. The angle iron mating faces are covered with Fiberfrax felt to provide a seal between the stage and furnace. Stage movement is provided by means of a 12 volt linear actuator (BSA AA250D12) with a 36 inch travel. The stage includes a 1M scale to determine the sampling distance from the burner nozzle in 0.5 cm increments. The translating stage can traverse the full length of the furnace in approximately 30 seconds and will be used to determine axial and radial temperature and concentration profiles. The sample port can also be equipped with a UV detector for flame length studies.

4.1.1.4 Exhaust system

The combustion gases exit the furnace through a water cooled 2" exhaust duct. The duct conveys the gases to a centrifugal exhaust fan/cyclone where the majority of particulate matter entrained in the exhaust stream are removed. The cyclone is sized to remove 95-99 % of particles greater than 20 microns and 60% greater than 7 microns. Thus the system will remove virtually all of the unburned coal and the majority of flyash not collected by the ash trap. The exhaust includes a dilution air system in addition the water-cooled jackets to further cool the exhaust gases before entering the cyclone.

4.1.1.5 Optical window

As the current experimental analysis includes evaluation of the near-flame region, it was desirable to include a full-length observation port in the furnace. This will allow researchers to conduct flow visualization studies, flame shape and length analyses without being limited by the field of view of conventional 2" sample ports. The flow visualization window is constructed of three sections of optical quartz glass 1/4" inch thick. The glass is fitted into channels cut into the furnace walls and cemented in-place using moldable Kaowool caulking. The gaps between the quartz sections are also sealed with moldable Kaowool. While the current windows are not suitable for laser diagnostics, they can be easily changed for future experiments.

A process flow diagram for the 2m Furnace is presented in Figure 28. The furnace can fire natural gas, pulverized coal fed via a loss in weight feeder system (K-Tron, KT20) or liquid fuels.

4.1.2 Burner

A novel burner system was constructed to complement the 2M furnace. A axial jet burner was utilized for the project to produce Type O turbulent diffusion flames. The burner consists of a fuel injection tube surrounded by the secondary air. The dual flow arrangement produces recirculation zones downstream of the burner which depend on velocity of the primary and secondary air flows and degree of secondary air swirl. The burner design depicted in Figure 28 includes variable sleeves for both the primary (transport) and secondary (combustion) air streams. Thus one can adjust the velocity of the combustion air by selecting a wider or narrower combustion air sleeve without affecting the primary air flow conditions. The velocity of the primary air-coal stream can be controlled in a similar fashion; however, both the primary and secondary air sleeves must be adjusted to maintain a constant secondary air velocity. The burner system includes 5 secondary air sleeves from 1.1 to 2.1" in diameter and 5 primary air tubes 3/16 to 5/8" in diameter as shown in Table 4.

The primary and secondary air streams were sized based on a maximum coal firing rate of 6 pounds per hour (2.7 kg/hr) and an air to coal stoichiometric air to fuel ratio (SR) of 1.2. Based on these parameters the maximum combustion air requirements for the furnace are 15.3 SCFM. The primary air (transport air) flow of 3 SCFM is

based on 20% of the total combustion air. The secondary air flow is 12.3 SCFM. Table 5 presents the range of combustion air velocities that can be obtained using various combustion air sleeves and the 5/8" diameter primary fuel tube.

Another method of adjusting the flow conditions of the secondary combustion air is to preheat the combustion air. Air preheat is provided by a 6 kW circulation heater (Chromalox #GCHCIS-01-006P-E1XX) specifically designed for the U of A combustion lab. The heater is sized to provide combustion air at the burner inlet at temperatures up to 1000°F.

4.1.3 Sampling System

The 2M sampling system includes both gas and temperature sampling devices. A water cooled quench probe is utilized for extractive gas sampling. The probe is constructed from a 1/2" and 3/8" cooling sheaths welded to a 1/4" sample tube. A 1/16" quench needle is inserted into the sample tube to wash soot from the tube. The quench needle is supported by triangular braces along its length to minimize blockage. The quench probe is depicted in Figure 30. Two quench probes were constructed, one 36" long for use in the fixed ports, and one 54" long for use in the translating stage. The longer probe has 1/2" gradations marked along its length to allow operators to measure how far into the furnace the probe is inserted.

Quench water is controlled by means of a rotameter and can flow at up to 120 ml/min. The gas sample exiting the probe flows into a water knockout pot connected to a 15 foot hydrostatic drain leg where the bulk of the quench water is removed. The knockout pot is fitted with a float valve that seals the sample line outlet to prevent water from flowing into the sample system. The gas sample then flows through a sample conditioning system consisting of a second knockout pot and refrigerated condensation coil and dual 0.5 micron coalescing filters to remove any remaining water and particulates. The sample then passes through a diaphragm pump and through a series of gas manifolds upstream of each analyzer. The manifolds are used to direct sample, calibration or zero gases to the analyzers. Fairchild Model 10 low pressure regulators prevent analyzer over-pressurization, and rotameters (Matheson 603) control flow rates to the analyzers. Each analyzer is also fitted with a 2 micron metal frit filter.

Continuous gas analysis includes CO and CO₂, oxygen and NO_x. All analysis is conducted on California Analytical instruments. CO and CO₂ are analyzed via infrared spectroscopy (Model ZRH). Oxygen is measured paramagnetically (Model 100P) and NO_x as NO is measured by chemiluminescent techniques (Model 300 CLD).

Gas temperatures in the furnace are measured by means of a water cooled R-type thermocouple probe. The probe is constructed from 1/2", 3/8" and 1/4" stainless sheaths welded together. The bare thermocouple wire is threaded into ceramic insulators inserted into the probe. Again two probes were constructed, one for the fixed sample

ports and one for the translating stage. Temperature data is transmitted directly into the data acquisition system and displayed on the control panel. K-type thermocouples measure the temperature at the furnace exhaust downstream of the cooling jackets and downstream of the dilution air system. The furnace top is also equipped with a K-type thermocouple for safety monitoring.

4.1.4 DAQ system

The zoned wall temperatures, furnace temperature and gas composition data are collected and displayed at the control panel by means of a computer data acquisition (DAQ) system (Labtech v. 12). The DAQ includes sample averaging and data-logging features for monitoring and evaluation of experimental data. Additional data streams of the DAQ include flow rates of combustion air, dilution nitrogen and other gas streams that may be injected into the furnace.

4.2 Hot-Flow Natural Gas Experiments

A series of combustion experiments were conducted in the 2M furnace to independently evaluate the impact of velocity and momentum on heat recirculation, air/fuel mixing and the resultant flame stability for turbulent axial diffusion flames firing natural gas. The original experimental design included conducting a 2^4 half-factorial design series of experiments with additional experiments for those parameters that show large effects in the initial runs. This would include a total of 8 experimental runs and replicates.

The parameters to be varied included: velocity of natural gas, velocity of combustion air, combustion air temperature and wall temperature. System effects to be analyzed included flame standoff distance, flame stability and combustion efficiency. However initial experimental attempts to achieve stable flames were unsuccessful. The scope of experiments was revised to those presented in Table 6. Attempts were made to maintain the air to fuel stoichiometry (SR) between 1.1 and 1.3.

4.2.1 Results

The 2M furnace produced attached flames under a wide range of firing rates with the 5/8" diameter fuel tube and 1.69" ID annulus. Lower firing rates produced lazy laminar flames. Increasing the firing rates to produce turbulent combustion air flows ($Re > 2200$) produced turbulent attached diffusion flames even though the natural gas flow rates were laminar ($Re < 1000$). Figure 31 presents a typical flame under such conditions. Heating the furnace walls to 900°C produced stable attached flames for higher flows. However, it was not possible to attain a stable flame for flows with gas velocities exceeding 6.2 fps ($Re\ 1674$) or combustion air velocities exceeding 10.2 fps ($Re\ 5400$).

Decreasing the combustion air annulus to 1.18" produced stable detached flames as shown in Figure 32. Decreasing the fuel tube diameter to 3/8" produced more turbulent detached flames. However it was possible to maintain the flame front by

adjusting the UV Flame detector position. Again, increasing the wall temperature increased the firing rates capable of maintaining a stable flame for both the 5/8 and 3/8" gas tube scenarios.

Flame stability was plotted as a function of gas and air velocities as shown in Figure 33. The data presented here are for stoichiometric ratios of 1.1 to 1.22. They are also plotted as a function of momentum fluxes as shown in Figure 11. Note that the axes in Figure 34 are titled Air and Gas Energy. Both figures clearly depict flame stability regimes. The stabilizing effects of heating the furnace walls is also evident.

Heating the furnace walls tended to increase the flame standoff distance as shown in Figure 35 for both the 5/8 and 3/8" fuel tubes under both high and low operating conditions. Higher wall temperatures tended to broaden and increase the length of the detached flame as shown in Figure 36. Flame length measurements were made using a UV Flame detector mounted on the translating stage.

4.2.2 Conclusions

The direct relationship between furnace wall temperature and flame stability was expected. Hotter walls would provide sufficient thermal energy necessary to exceed the activation energy of the air/fuel mixture and provide a stable reaction compared to cold walls and the same flow conditions. However, the increase in flame standoff distance with increasing wall temperature was unexpected and counter-intuitive. In this situation, increasing the wall temperature for a stable flow situation was expected to reduce the standoff distance due to the increased thermal energy and diffusivities of the reactants. It is believed that while the kinetic energy of the reactants increase, the increase in viscosity and reduced density may actually reduce turbulent mixing resulting in longer standoff distances.

The presence of distinct flow regimes for both velocity and momentum flux warrants additional work. While these regimes may be obvious "after the fact", it is not readily apparent where exactly such regimes exist for a given burner geometry. It is interesting to observe that the attached flame data appears to be located where the two detached flame regimes merge. The ability to stabilize flames with either preheated air or hot walls is also readily apparent in the flow regime graphs. It is also important to note that the stability regimes seem to be about equally sensitive to fuel/air velocity ratios and momentum flux ratios.

The current experimental effort validates the novel hot furnace design. The full length quartz window was used for visualizing on photographing flame shapes and allowed greater flexibility adjusting the flame detectors for monitoring the detached flames. The translating stage was used extensively for measuring the flame standoff distances and flame length as well as for flame stabilization. Finally, the ability to easily control the wall temperature greatly increased the range of operating conditions that produced a stable flame. Initially, wall temperature was thought to be of minor importance!

The investigators realized rather early in the experimental stage that systematically controlling gas velocity and momentum independently was virtually impossible due to the interconnectedness of velocity and momentum. While it is easy to change the combustion air annulus to increase or decrease the gas velocity, it was very difficult to control the preheat air temperature and flow rate to maintain a constant momentum without drastically altering the stoichiometric ratio. Also, adjusting the fuel tube not only affected the fuel velocity, it also changed the velocity of the combustion air due to the change in the diameter of the annulus.

The difficulty of achieving attached flames and **stable** detached flames cannot be under estimated, especially when considering laboratory safety. The use of two UV Flame detectors were invaluable for attaining stable detached flames.

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Figure 1 NO_x Formation Pathways in Coal Combustion

Figure 2 Schematic Diagram of the Cold-Flow Experimental Setup

Figure 3 Detailed Schematic of Cold-Flow Primary and Annular Nozzles

Figure 4 Gas and particle velocities at the nozzle exit (a) axial mean velocity, (b) radial mean velocity, (c) axial fluctuating velocity, and (d) radial fluctuating velocity

(c)

(d)

- Gas, $U_{go}/U_{gi} = 0$
- △— Gas, $U_{go}/U_{gi} = 1$
- Gas, $U_{go}/U_{gi} = 1.8$
- Particle, $U_{go}/U_{gi} = 0$
- ▲— Particle, $U_{go}/U_{gi} = 1$
- Particle, $U_{go}/U_{gi} = 1.8$

Figure 4. Gas and particle velocities at the nozzle exit (c) axial fluctuating velocity, and (d) radial fluctuating velocity

Figure 5. Axial mean gas and particle velocities at (a) $x/d_i = 3$ and (b) $x/d_i = 10$

Figure 6. Radial particle velocity at (a) $x/d_i = 3$ and (b) $x/d_i = 10$

Figure 7. Radial mean gas velocities at (a) $x/d_i = 3$ and (b) $x/d_i = 10$

Figure 8. Axial evolution of the axial fluctuating gas and particle velocities along jet centerline

Figure 9. Directional angle, α of particle velocity vector at (a) $x/d_i=3$ and (b) $x/d_i=10$

Figure 10. Comparison between the directional angle, α of velocity vector of the gas and particle at (a) $x/d_i = 3$ and (b) $x/d_i = 10$.

- Gas, $U_{go}/U_{gi} = 0$
- △— Gas, $U_{go}/U_{gi} = 1$
- Gas, $U_{go}/U_{gi} = 1.8$

Figure 11. Axial fluctuating gas velocity at (a) $x/d_i = 3$ and (b) $x/d_i = 10$, and the radial fluctuating gas velocity at (c) $x/d_i = 3$ and (c) $x/d_i = 10$

- Particle, $U_{go}/U_{gi} = 0$
- ▲— Particle, $U_{go}/U_{gi} = 1$
- Particle, $U_{go}/U_{gi} = 1.8$

Figure 12. Axial fluctuating particle velocity at (a) $x/d_i = 3$, (b) $x/d_i = 10$, and the radial fluctuating particle velocity at (c) $x/d_i = 3$, (d) $x/d_i = 10$

(a)

(b)

- Gas, $U_{go}/U_{gi} = 0$
- △— Gas, $U_{go}/U_{gi} = 1$
- Gas, $U_{go}/U_{gi} = 1.8$

Figure 13. Reynolds stress of the gas-phase at (a) $x/d_i = 3$, (b) $x/d_i = 10$

r/d_i

- Particle, $U_{go}/U_{gi} = 0$
- ▲— Particle, $U_{go}/U_{gi} = 1$
- Particle, $U_{go}/U_{gi} = 1.8$

r/d_i

Figure 14. Particle Reynolds stress at (a) $x/d_i = 3$, and (b) $x/d_i = 10$

500 grids in axial direction (Shown part of the domain)
40 grids in radial direction

Figure 15. Computational grid used in Fluent 4.5

Figure 16. Comparison between the Fluent predictions and experimental data on the axial mean gas velocity along the jet centerline for all U_{go}/U_{gi}

Figure 17. Comparison between the Fluent predictions and experimental data on the turbulence kinetic energy along the jet centerline for all U_{go}/U_{gi}

□ Data, $U_{go}/U_{gi} = 0$
 — Fluent, $U_{go}/U_{gi} = 0$

(a)

△ Data, $U_{go}/U_{gi} = 1$
 — Fluent, $U_{go}/U_{gi} = 1$

(b)

○ Data, $U_{go}/U_{gi} = 1.8$
 — Fluent, $U_{go}/U_{gi} = 1.8$

(c)

Figure 18. Comparison between the Fluent prediction and experimental data on mean axial gas velocity at $x/d_i = 10$ for all velocity ratios

□ Data, $U_{go}/U_{gi} = 0$
— Fluent, $U_{go}/U_{gi} = 0$

(a)

△ Data, $VR = 1$
— Fluent, $VR = 1$

(b)

○ Data, $U_{go}/U_{gi} = 1.8$
— Fluent, $U_{go}/U_{gi} = 1.8$

(c)

Figure 19. Comparison between the Fluent prediction and experimental data on the gas turbulent kinetic energy at $x/d_i = 10$ for all velocity ratios

Figure 20. Comparison between Fluent prediction and experimental data on the radial mean gas velocity at (a) $x/d_i = 3$ and (b) $x/d_i = 10$

Figure 21. Comparison between the SSF and DSF predictions in the axial particle velocity of the 70 microns glass bead particle for the three U_{go}/U_{gi}

Figure 23. Comparison between the SSF and DSF predictions in the radial particle velocity of the 70 microns glass bead particle for the three U_{go}/U_{gi}

Figure 22. Comparison between the SSF and DSF predictions in the particle trajectories of the 70 microns glass bead particle for the three U_{go}/U_{gi}

Figure 24. 2M Furnace Details

Figure 25. Furnace Construction Photos

Figure 26. Three-Zone Heater Wiring Diagram

Figure 27. Translating Stage

Figure 28. 2M Furnace Process Flow Diagram

Figure 29. Burner Details

Schematic of Water Cooled and Water Quench Sampling Probe

Figure 30. Combustion Gas Sampling Probe

Figure 32. Detached Flame

Figure 31. Attached Flame

Figure 33. Flame Stability Regime as a function of Velocity

Figure 34. Flame Stability Regime as a function of Momentum Flux

Figure 35. Flame detachment vs. Wall temperature

Figure 36. Flame Length vs. Wall Temperature

Table 1 Literature values for the of fluid time scale, τ_f

Author	Flow configuration	Le	Ue	$\tau_f = L_e / U_e$
Hardalupas <i>et al.</i> (1989)	Pipe flow	d_{pipe}	U_{to}	d_{pipe}/U_{to} U_{to} = the average gas velocity
	Pipe flow	-	-	Lagrangian integral time scale (Vames and Hannatty, 1988 and Tennekes and Lumley, 1972) $\frac{\epsilon_f}{v_f'^2}$ $\epsilon_f = 0.037(u_f^*)d_{pipe}$ u_f^* = frictional gas velocity ϵ_f = gas dissipation rate
Hestroni (1989)	Pipe flow	$0.1 d_{pipe}$	u_f^*	$0.1 d_{pipe}/u_f^*$
Agarwal <i>et al.</i> (1994)	Jet flow	-	-	$1/f_p$ f_p = preferred mode of frequency of the jet
	Jet flow	Jet width	$U_{f,cl}$	Jet width/ $U_{f,d}$ $U_{f,d}$ = the axial mean gas velocity in the jet centerline
Crowe <i>et al.</i> (1985)	Jet flow	Half jet width	$U_{f,cl}$	0.5 Jet width/ $U_{f,cl}$
Hardalupas <i>et al.</i> (1989)		-	-	Lagrangian integral time scale (Vames and Hannatty, 1988 and Tennekes and Lumley, 1972) $\frac{\epsilon_f}{v_f'^2}$ $\epsilon_f = 0.021632 U_{f,cl} x$
Longmire and Eaton (1992)	Jet flow	Distance between the vortical structures (L)	$0.5 U_{f,cl}$	$L/(0.5 U_{f,cl})$
Prevost <i>et al.</i> (1996)	Jet flow	$0.1 x$ Half jet width	v_f'	$0.1x/v_f'$ x = axial distance from the jet nozzle

Table 2 Operating flow conditions

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Gas-phase						
Central jet						
Centerline velocity (m/s)	11.6	11.6	11.6	11.6	11.6	11.6
Mass flow rate (kg/s)	0.0017	0.0017	0.0017	0.0017	0.0017	0.0017
density (kg/m ³)	1.2	1.2	1.2	1.2	1.2	1.2
Annular jet						
Maximum velocity (m/s)	0	11.65	17.9	0	11.65	17.9
Mass Flow Rate (kg/s)	0	0.0068	0.011	0	0.0068	0.011
Density (kg/m ³)	1.2	1.2	1.2	1.2	1.2	1.2
Velocity ratio	0	1	1.8	0	1	1.8
Particle-Phase						
Central Jet						
Centerline velocity (m/s)	-	-	-	8.95	8.95	8.95
Mass flow rate (kg/s)	-	-	-	0.0009	0.0009	0.0009
Mass loading	-	-	-	0.5	0.5	0.5
Density (kg/m ³)	-	-	-	2,500	2,500	2,500

Table 3. Particle St of 70 micron glass bead particles at different axial locations

Location	St		
	$U_o/U_i = 0$	$U_o/U_i = 1$	$U_o/U_i = 1.8$
x/di			
0	20	20	20
1	43	20	21
3	33	16	16
5	20	13	14
10	11	8.8	12

Table 4 Burner Sleeve Sizes

Primary Annulus			
Nominal	OD	wall	ID
Tubing	in	in	in
3/16	0.1875	0.035	0.118
1/4	0.25	0.035	0.180
3/8	0.375	0.035	0.305
1/2	0.5	0.035	0.430
5/8	0.625	0.035	0.555
Secondary Annulus			
Nominal	OD	wall	ID
Size	in	in	in
5/8 tube	0.625	0.035	0.555
3/4 tube	0.75	0.035	0.680
1" 5s pipe	1.315	0.065	1.185
1 1/4" pipe	1.66	0.18	1.300
1 1/2" pipe	1.91	0.135	1.640
Furnace tube	1.89	0.1	1.690
2" pipe	2.375	0.154	2.067

Table 5. Combustion Air Velocities

Secondary	Primary Tube						Temp - K	471	589	644	700	755	811
Tube	Option						Temp-°F	388.4	600	700	800	900	1000
Option		Nominal Size	OD	wall	ID	Tube Area	Area Gas	Vel.	Vel.	Vel.	Vel.	Vel.	Vel.
			in		in	ft^2	Ft^2	fps	fps	fps	fps	fps	fps
1	5	5/8 tube	0.625	0.035	0.555	0.0017	0.0000	0.00	0.00	0.00	0.00	0.00	0.00
2	5	3/4 tube	0.75	0.035	0.680	0.0025	0.0004	864.66	1080.3	1182.2	1284.2	1386.1	1488.0
3	5	1" 5s pipe	1.315	0.065	1.185	0.0077	0.0055	61.23	76.5	83.7	90.9	98.2	105.4
4	5	1 1/4" pipe	1.66	0.18	1.300	0.0092	0.0071	47.8	59.7	65.3	70.9	76.6	82.2
5	5	1 1/2" pipe	1.91	0.135	1.640	0.0147	0.0125	27.0	33.7	36.9	40.1	43.3	46.5
6	5	Furnace tube	1.89	0.1	1.690	0.0156	0.0134	25.2	31.5	34.4	37.4	40.4	43.3
7	5	2" pipe	2.375	0.154	2.067	0.0233	0.0212	16.0	20.0	21.9	23.7	25.6	27.5

Table 6. Natural Gas Experimental Scope

Air Temperature	Wall Temperature	Gas Tube Size	Air Pipe Dimensions
Cold (25°C)	Cold	3/8" Tube	1.18" ID
Pre heated (190-245°)	500-900°C	5/8" Tube	1.69" ID