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**AN INVESTIGATION OF SWIRLING FLOW MIXING FOR APPLICATION
IN AN MHD PULVERIZED COAL COMBUSTOR USING ISOTHERMAL
MODELING**

Interim Report

By
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May 1980

Work Performed Under Contract No. AC02-79ET10815

The University of Tennessee Space Institute
Energy Conversion Division
Tullahoma, Tennessee

U. S. DEPARTMENT OF ENERGY

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COMBUSTOR USING ISOTHERMAL MODELING

INTERIM REPORT

Based on

A Thesis

Presented for the

Master of Science Degree

The University of Tennessee, Knoxville

The University of Tennessee Space Institute
Energy Conversion Division

PREPARED FOR THE UNITED STATES
DEPARTMENT OF ENERGY

William H. Power

MAY 1980

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ABSTRACT

The purpose of this study was to investigate combustor reactant mixing with swirling oxidizer flow. The combustor configuration that was considered was designed to simulate a 4 lbm/sec mas flow pulverized coal combustor being tested in The University of Tennessee Space Institute MHD Facility. A one-fourth dimensionally scaled combustor model was developed for isothermal flow testing. A comparison was made of cold flow tests using 3 swirler designs with a base case oxidizer injector design of perforated plate which demonstrated acceptable performance in the 4 lbm/sec MHD combustor.

The three swirlers that were evaluated were designed to allow a wide range of swirl intensity to be investigated. The design criterion of the swirler was the swirl number which has been related to swirler geometry.

The results of the study showed that the swirlers that were tested fell short of the mixing characteristics displayed with the perforated plate base case oxidizer injector. Test data obtained with the cold flow model established that the actual swirl numbers of two of the swirlers were much lower than the design swirl numbers. Recirculation zones were defined for all configurations that were tested, and a comparison of velocity profiles was made for the configurations.

The test program proved to be a very useful phase of a large scale combustor development program. In general the test results were consistent with similar studies that were investigated by means of available literature. Although the oxidizer swirler configurations that were tested proved to be inadequate for the MHD combustor, the tests provided valuable information that can contribute to future swirler designs as well as other oxidizer and fuel injector concepts for direct application in an MHD pulverized coal combustor.

TABLE OF CONTENTS

CHAPTER		PAGE
I.	INTRODUCTION	1
II.	FUEL AND OXIDIZER MIXING	5
III.	ISOTHERMAL COMBUSTION MODELING	11
IV.	COMBUSTOR MODEL DESIGN	14
	Assembly Design.	14
	Oxidizer Swirler Design.	14
	Fuel Injector Design	27
	Baseline Oxidizer Injector	27
V.	EXPERIMENTAL INSTALLATION AND PROCEDURE.	34
	Installation	34
	Instrumentation.	34
	Test Procedure	36
VI.	EXPERIMENTAL RESULTS	40
	Defining of Recirculation Zone	40
	Dynamic Pressure Profiles.	54
	Swirl Number Correlation	54
	Combustor Oxidizer Injector Pressure Loss.	58
VII.	CONCLUSIONS.	62
	Swirlers	62
	Base Case.	63
VIII.	RECOMMENDATIONS.	66
	LIST OF REFERENCES.	69

LIST OF TABLES

TABLE	PAGE
1. Swirler Design Descriptions	23
2. Fuel Injector Comparison.	28
3. Operating Conditions.	37
4. Oxidizer Injector Recirculating Flow Parameters . .	55
5. A Comparison of Design and Measured Swirl Numbers	59

LIST OF FIGURES

FIGURE	PAGE
1. Typical Swirler Configuration.	6
2. Streamline Representation of Swirling Oxidizer Flow.	8
3. Cross-Sectional View of Scaled Combustor Simulator with Baseline Injector	15
4. Cross-Sectional View of Scaled Combustor Simulator with Swirler Injector.	16
5. Combustor Showing Downstream Section	17
6. Combustor Showing Upstream Section	18
7. Oxidizer Swirler Configurations.	19
8. Streamline Representation of Expanding Swirling Oxidizer Flow	21
9. Swirler TEI-0127-3	24
10. Swirler TEI-0128	25
11. Swirler TEI-0129	26
12. One-Fourth Scale Fuel Injector	29
13. Cumulative Heat Loss in the 4 Lbm/Sec Combustor System	31
14. One-Fourth Scale Base Line Oxidizer Injector with Fuel Injector Installed	32
15. Baseline Oxidizer Injector	33
16. Test Apparatus Installation.	35
17. Fuel Injector Flow Calibration Curve Using Water.	39
18. Swirler TEI-0127-3 Recirculation Zones	41
19. Swirler TEI-0128 Recirculation Zones	42

FIGURE	PAGE
20. Swirler TEI-0129 Recirculation Zones	43
21. Baseline Case Recirculation Zones.	44
22. Primary Recirculation Zone Using Swirler TEI-0127-3	46
23. Primary Recirculation Zone Using Swirler TEI-0128	47
24. Primary Recirculation Zone Using Swirler TEI-0129	48
25. Outer Flow Field Recirculation Zone Using Baseline Injector TEI-0130 Operating in the Unstable Mode with Higher Recirculating Fuel Flow	49
26. Outer Flow Field Recirculation Zone Using Baseline Injector TEI-0130 Operating in the Unstable Mode with Decreased Recirculating Flow	50
27. Centerline Recirculation Zone Using the Baseline Injector TEI-0130	51
28. Recirculation Zone Boundaries for Centrally Located Zones.	52
29. Recirculation Boundaries for Outer Zones.	53
30. Recirculation Zone Boundaries for Different Swirl Numbers.	56
31. Swirler Flow Area vs. Pressure Coefficient for the Three Swirler Configurations	60
32. Normalized Combustor Pressure Drop vs. Oxidizer Mass Flow Rate.	61
33. Swirl Number vs. Pressure Coefficient for All Configurations Tested.	64
34. Combustor Development Program Flow Chart.	67

LIST OF SYMBOLS

A_D	Swirler discharge area
A_R	Flow restricting area
B	Swirl vane height
C_D	Particle drag coefficient
C_p	Pressure coefficient $\left(\frac{\Delta P}{.5\rho U^2} \right)$
d	Flow area diameter
D	Combustor diameter
D_p	Fuel particle diameter
G_x	Axial flux of axial momentum
G_ϕ	Axial flux of angular momentum
K	Resistance coefficient
$\dot{m}$	Mass flow rate
$\dot{m}_R$	Recirculating mass flow rate
p, P	Pressure
r_I	Inner radius of swirler annulus
R_O	Outer radius of swirler annulus
R_1	Inner radial location of swirl vane
Re_{IN}	Inlet oxidizer Reynolds number
Re_p	Particle Reynolds number
s	Swirl vane thickness
S	Swirl number
S_{CORR}	Design swirl number corrected to the combustor gas stream temperature variations
S_{ISO}	Design swirl number for isothermal model

T_{IN}	Combustor inlet temperature
T_{OUT}	Combustor exit temperature
U_D	Discharge axial velocity
U_p	Fuel particle velocity
v	The y-component of velocity or the radial component of velocity
Y	Expansion factor
z	Number of swirl vanes
α	Swirl vane angle
Δ	Differential
θ	Angle of oxidizer resultant velocity with respect to the combustor axis
μ	Oxidizer coefficient of viscosity
ρ	Density
ρ_1	Density at an upstream location
σ	Ratio of average tangential and radial velocity components
τ_B	Shear stress at the stream boundary
ψ	Swirler blockage factor

CHAPTER I

INTRODUCTION

The development of efficient combustion systems has become a major consideration for the basis of combustor design in recent years. Since society has been made aware of possible energy resource shortages, as well as environmental pollution from combustion products, the need and application of combustion technology has been established in all types of industry. The combustion of pulverized coal has long served as a means of generating superheated steam in conventional power plants. A greater usage of coal as fuel would essentially increase fuel availability in the United States. However, the difficulty in meeting Federal pollution standards with coal fired systems has made coal undesirable as a fuel.

The use of an open cycle magnetohydrodynamic (MHD) electrical generating system in combination with a conventional electrical generating system has appeared to offer significant increases in cycle efficiencies over conventional power plant schemes.¹ Also the coal fired MHD concept has demonstrated an acceptable pollutant emissions level. However, to maintain acceptable MHD cycle efficiency, 2700 K to 2900 K combustor exit temperatures are required.² To achieve the necessary combustion gas

temperature the combustor geometry must be optimized to allow adequate mixing and combustion with minimal heat loss.³ While utilizing pulverized coal, a combustor residence time in the range of 50 milliseconds is required.⁴

Tests conducted by The University of Tennessee Space Institute, Energy Conversion Division, have established the feasibility of electrical power generation with acceptable pollutant emissions utilizing pulverized coal fired combustors. Pulverized coal combustors using a variety of fuel and oxidizer injection techniques have been tested at The University of Tennessee Space Institute. The first coal combustor design burned coal mixed with potassium carbonate, and oxygen. A second combustor concept utilizing a kerosene fueled burner to supply vitiated, high temperature oxidizer to the coal combustor was developed to simulate the combustion of preheated air. This second combustor design burned a vitiated oxygen-nitrogen mixture, potassium carbonate, and coal. Both coal combustor designs were single stage combustion chambers with pulverized coal being injected at the combustor centerline and oxidizer being injected by various techniques at locations around the fuel injector.

Based on the gas stream temperature requirements for MHD, combustion efficiency must be optimized. By establishing the most efficient mixing technique, combustor

losses should be minimized.³ Swirling flow has proved very useful in practically all applications of combustion.

Three most favorable aspects of swirling flow in combustors are:

1. Reduction of effective combustion length as a result of a high entrainment rate.
2. Improved flame stability as a result of a toroidal recirculation zone.
3. Reduction of flame impingement due to the combustion taking place primarily on the axial centerline of the combustor.⁵

A project was initiated at The University of Tennessee Space Institute to allow the evaluation of various combustor mixing techniques for pulverized coal combustors. The initial phase of a combustor development program was established to allow screening of oxidizer and fuel injectors prior to large scale fabrication and testing of these components. The purpose of this document is to establish the resulting mixing effects of swirling oxidizer flow in a single stage pulverized coal combustor. The overall coal combustor configuration considered in this work was based on the configuration of the most recently tested 4, lbm/sec vitiated coal combustor (full scale combustor).

Tests using an isothermal scaled combustor model (discussed in detail in Chapter IV) were conducted to evaluate

different oxidizer injector designs: one base case high velocity axial injector and three swirler injectors with varying amounts of oxidizer swirl. One fuel injector was used with all oxidizer injector configurations. The base case high velocity axial flow injector was used as a reference point in evaluating the swirlers. Water flow patterns were observed and photographed in the presence of oxidizer flow. A few tests were also made while injecting coal. However, only visual observations were made because of difficulties in maintaining uniform flow. Future tests are recommended to more quantitatively evaluate coal injectors. Oxidizer recirculation zones were also defined for each oxidizer injector configuration.

Overall, we hope this effort will provide a significant first step toward a cold flow capability at UTSI.

CHAPTER II

FUEL AND OXIDIZER MIXING

Mixing of oxidizer and fuel streams has been very efficiently done by utilizing swirling flow in all types of combustion systems.^{5,6} Figure 1 represents a typical swirler configuration for use in a combustor.

An attempt was made to evaluate the swirling concept in a coal combustor at The University of Tennessee Space Institute (UTSI) in 1972 and 1973. The combustor tests of these oxidizer swirl injectors demonstrated unsatisfactory performance.⁷ However, the aerodynamics of these configurations was not studied prior to the combustor tests.

In the designs used, in reference 7, at UTSI there was also a baffle introduced at times into the combustion chamber which gives an even higher tangential to axial flow velocity ratio than is obtained in the body of the combustor itself.

In general, by creating large velocity gradients in a mixing zone entrainment is enhanced. Mixing rates of two streams at a common boundary can be related to

Figure 1. Typical Swirler Configuration.

the conservation of momentum in a two-dimensional form of:⁸

$$\frac{\partial \rho u^2}{\partial x} + \frac{\partial \rho uv}{\partial y} = \frac{\partial \tau}{\partial y} - \frac{\partial p}{\partial x} .$$

The stream boundary shear stress is normally related to the velocity gradient in the following manner:⁸

$$\tau_B = \mu \frac{\partial u}{\partial y} .$$

The entrainment of one stream by the second stream results when the velocity gradient, and hence shear stresses, exist. The link between the two streams results from the conservation of momentum in the system.

Swirler configurations have been observed to display high rates of recirculation resulting from flow entrainment. However, swirling flow utilizes the effects of expanding flow as well as velocity gradients to establish strong recirculation.⁵ (Figure 2 is a representation of swirling flow using streamlines.) Therefore total pressure losses can be reduced when compared to a similar recirculation that relies solely on a velocity gradient.

The desirable recirculation associated with swirling flow in combustors is closely related to the concept of vortex breakdown. The general trends discussed by Sarpkaya establish that the vortex breakdown is caused

Figure 2. Streamline Representation of Swirling Oxidizer Flow.⁵

by either increased swirl⁹ or an adverse pressure gradient.¹⁰ Based on extensive experimental work with water flow, the reverse flow "bubble" was observed to move upstream as the swirl was increased.⁹ Similar trends were observed with the existence of an adverse pressure gradient resulting from a diverging flow area.⁵

Recent efforts to establish numerical predictions for vortex breakdown have been successful when considering inviscid flow.^{11,12} This is an indication that the recirculation resulting from swirling flow is not largely dependent on high velocity gradients.

The parameter utilized in swirler design is the swirl number. The swirl number is defined as the ratio of the axial flux of angular momentum to the axial flux of axial momentum divided by the discharge radius.⁵

$$S = \frac{G_{\phi}}{G_x R}$$

Syred and Beér have established that the recirculating flow is significantly increased with swirl numbers greater than 0.6.^{5,6}

Although the primary aerodynamic characteristics are established by the swirl number, there are secondary effects resulting from the properties of the fuel being injected on the swirler centerline. The entrainment rate of the fuel particles has been related to the particle

drag coefficient as follows:¹³

$$C_D = 0.32 + \frac{4.4}{\sqrt{Re_p}} + \frac{24}{Re_p} \quad , \quad .001 \leq Re_p \leq 6 \times 10^3 .$$

The particle Reynolds number, Re_p , is defined as¹³

$$Re_p = \frac{\rho D_p |u - u_p|}{\mu} .$$

When applying the above equations to this study, effects of changing the fuel simulating medium from coal to water resulted only in altering the drag coefficient of the fuel particle. By changing this parameter general flow patterns remained unchanged since the flow characteristics were dictated by the oxidizer. (The governing factors of the combustor aerodynamics for each configuration are established in Chapter VI.)

CHAPTER III

ISOTHERMAL COMBUSTION MODELING

Isothermal modeling has served as a useful tool to establish aerodynamic characteristics in combustors.⁶ The advantages of flow visualization testing are that data can be easily obtained and that tests can be conducted inexpensively. The difficulty in obtaining accurate combustion data has been a major obstacle in combustor development programs. Effective numerical solutions to dynamic combustion processes have been very limited in accuracy until recently¹⁴ and thus no single basis for combustor design has been available. All available resources (physical modeling, analytical modeling, and full scale testing) have had to be utilized for efficient combustion development programs to be carried out. Flow visualization studies have best served as an important initial phase of combustor development programs.⁶

Utilizing an isothermal model to establish improved mixing techniques brings to point the controlling conditions of the simulator. The combustor Reynolds number, referenced to inlet fluid properties, is utilized to control the oxidizer conditions.⁶ The Reynolds number is defined as follows:¹⁵

$$Re_{IN} = \frac{\rho U D}{\mu}$$

To apply the isothermal flow characteristics to the full scale combustor utilizing a swirler, a correction factor should be applied to the design swirl number. The relationship is as follows:⁵

$$S_{CORR} = S_{ISOTH} \left(\frac{T_{IN}}{T_{OUT}} \right),$$

where:

S_{CORR} = the design swirl number applied to the full scale combustor.

S_{ISOTH} = the design swirl number applied to the isothermal simulator.

T_{IN} = the design combustor inlet temperature for the full scale combustor.

T_{OUT} = the design combustor outlet temperature for the full scale combustor.

It was decided prior to the conducting of the tests that the model fuel flow would be operated over a flow range. The desired flow range for tests would allow a general observation to be made with:

1. Similar oxidizer-fuel mass flow ratios for the model and the full scale combustor.
2. Similar oxidizer-fuel momentum ratios for the model and the full scale combustor.

3. Similar fuel particle drag coefficients for the model and the full scale combustor.

Prior to component evaluation it was decided that if any test conditions contributed to significant isothermal aerodynamic changes, this would be considered in design recommendations made for the full scale combustor.

CHAPTER IV

COMBUSTOR MODEL DESIGN

Assembly Design

The combustor simulator was constructed of readily available materials. The characteristic diameter of the model was $1/4$ of the internal diameter of the 4 lbm/sec combustor in the UTSI Research and Development facility. The downstream test section was constructed of transparent acrylic tubing (3.5" I.D.) to allow visualization of oxidizer and fuel mixing. The fuel disperser was installed concentrically with the oxidizer injector. The model was designed to allow the axial location of fuel injection to be varied. The upstream portion of the simulator was designed to operate at higher than atmospheric pressure. The combustor simulator accepted any one of the swirlers or the baseline oxidizer injector (Figures 3 through 6).

Oxidizer Swirler Design

The two general types of swirler configurations used are the axial flow type and the radial inflow type (Figure 7). Both configurations were considered for the MHD combustor. The radial inflow design was decided upon for this series of tests. The reasons for selecting the radial inflow swirler type are as follows:

Figure 3. Cross-Sectional View of Scaled Combustor Simulator with Baseline Injector

Figure 4. Cross-Sectional View of Scaled Combustor Simulator with Swirler Injector.

Figure 5. Combustor Simulator Showing Downstream Section.

Figure 6. Combustor Simulator Showing Upstream Section.

Figure 7. Oxidizer Swirler Configurations.

1. The radial inflow design appeared more likely adaptable to ceramic construction (for the full scale combustor) requiring little or no water cooling.
2. The radial inflow design was more adaptable to the existing combustor geometry allowing more discharge flow area per unit of discharge diameter.

It was believed that by maintaining smaller outer annular diameters of the swirler discharge, recirculation could be enhanced with the increased radial expansion of the oxidizer flow (Figure 8).

Three swirlers were constructed to allow testing of a wide range of swirl numbers. Beér and Chigier established the relationship of the swirl number to swirler geometry:⁶

$$S = \frac{G_{\phi}}{G_x R_o} ,$$

where:

$$G_x = 2\pi \int_0^{R_o} \rho u^2 r dr ,$$

$$G_{\phi} = \sigma \left[\frac{\dot{m}^2}{\rho 2\pi B} \right] ,$$

Figure 8. Streamline Representation of Expanding Swirling Oxidizer Flow.

$$\sigma = \frac{1}{1 - \psi} \left[\frac{\tan \alpha}{1 + \tan \alpha \tan(\frac{\pi}{z})} \right] ,$$

$$\psi = \frac{zs}{2\pi R_1 \cos \alpha} .$$

In the above equations:

$\dot{m}$ = mass flow rate,

B = the swirl vane height,

s = the swirl vane thickness,

R_1 = inner radial location of the swirl vane,

z = the number of swirl vanes,

R_o - the outer radius of the annular discharge area,

α = the included angle between the swirl vane
centerline and the radial line contacting the
tip of the trailing edge of the swirl vane.

The design swirl number range that was tested was achieved by varying swirl vane angle and discharge annulus flow area. The design flow area was chosen to allow a design pressure drop to be maintained. Combustor oxidizer pressure drop was predicted by:¹⁶

$$\dot{m} = 0.525 Y d^2 \sqrt{\frac{\Delta P \rho_1}{K}}$$

Table 1 gives a summary of characteristics of the swirlers tested. Figures 9-11 show the corresponding vane angles and discharge annulus dimensions for each swirler.

TABLE 1
SWIRLER DESIGN DESCRIPTIONS

Part Number	Inlet Area (In ²)	Discharge Area (In ²)	Effective Diameter (In)	Swirl No.	$\frac{\Delta P^*}{P_c}$
TE1-0127	1.44	1.46	1.35	1.40	.015
TE1-0128	2.30	.920	1.08	2.45	.020
TE1-0129	1.725	2.10	1.48	3.28	.010

* $\frac{\Delta P}{P_c}$ value shown is calculated value for a
4 lbm/sec combustor operating at 75 psia.

Design Swirl No. = 1.40

Figure 9. Swirler TEI-0127-3.

Design Swirl No. = 2.45

Figure 10. Swirler TEI-0128.

Design Swirl No. = 3.28

Figure 11. Swirler TEI-0129.

Fuel Injector Design

The fuel injector was designed with three major considerations:

- 1) to simulate the geometry of the full scale fuel injector
- 2) to provide fuel spray patterns that were the same for both water and pulverized coal as the test fluids
- 3) to maintain acceptable fuel injector reliability (eliminate plugging) when using pulverized coal in the sub-scale injector.

The model fuel injector was developed by a cut and try procedure, wherein both water and dense phase pulverized coal were used as test fluids. The data analyzed were both visual, in-situ observations, and photographic records of the spray patterns created. From the observations, the spray patterns, their angle of spreading, and general configurations were acceptably close. The mean particle diameter in the water spray was estimated from photographs to be somewhat longer than the pulverized coal which was pulverized to mass mean diameter of 50 microns. The density of the water particles was 1 gram/cm^3 and the density of the coal particles is about 0.65 gm/cm^3 . The exact quality of the similitude between water sprays and dense phase coal sprays was left for future research. It is known that fundamental differences will exist in the combustion of sprays of liquids and sprays of solid particulate, but the observations of this research

study support the use of water as a test fluid for establishing the general trends in the mixing patterns within the combustor simulator.

The one-fourth scale fuel injector was modeled after the full scale fuel injector that was used in most tests. The full scale injector consisted of an annulus with four helical vanes. The reduced scale injector incorporated four circular holes drilled at the same angle as the discharge vane angle of the full scale injector.

Table 2 gives a comparison of full scale and reduced scale fuel injectors. The same fuel injector was used during all tests (Figure 12).

Baseline Oxidizer Injector

The one-fourth scale oxidizer injector consisted of an identical orifice pattern to the existing full scale injector, with orifice diameters being reduced by a factor of four. The full scale oxidizer injector that was modeled demonstrated the highest level of combustor performance of any 4 lbm/sec combustor configuration.¹⁷ The measured cumulative heat loss for two full scale oxidizer injector

TABLE 2
FUEL INJECTOR COMPARISON

Specification	Full Scale Fuel Injector	One-Fourth Scale Fuel Injector
Flow Area	.1327 in ²	.0123 in ²
Equivalent Diameter	.2055 in	.0625 in
Discharge Angle	34.8°	35°
L/D	> 3.0	2.44
Hole Annular Radius	.281 in	.087

Figure 12. One-Fourth Scale Fuel Injector.

configurations is shown in Figure 13. We believe the higher heat flux to the walls is due to higher combustion temperatures, which, in turn, implies better mixing and combustion was occurring with the water cooled injector plate. The higher heat fluxes also corresponded to improved MHD generator performance. Figures 14 and 15 show the one-fourth scale base line oxidizer injector.

Figure 13. Cumulative Heat Loss in the 4 Lbm/Sec Combustor System.¹³

Figure 14. One-Fourth Scale Base Line Oxidizer Injector
with Fuel Injector Installed.

Figure 15. Baseline Oxidizer Injector.

CHAPTER V

EXPERIMENTAL INSTALLATION AND PROCEDURE

Installation

The installation utilized for the combustor aerodynamic study was intentionally made as simple as possible. The test components that were not subjected to large pressure loadings were constructed of Plexiglas and Teflon. Components that were loaded due to pressure differentials were fabricated from carbon steel. Since the combustor simulator operated at atmospheric pressure, the maximum expected air pressure was approximately 15 psig at an upstream location of the oxidizer injector. For most tests an acrylic tube was used to prevent ambient air entrainment. Oxidizer flow conditions were measured with a 0.75-inch diameter square-edged orifice (Figure 16).

Instrumentation

Instrumentation consisted of the following:

- 1 Orifice upstream static pressure gage
- 1 Orifice differential pressure gage
- 1 Combustor upstream static pressure gage
- 1 Fuel supply pressure gage

Also, a 0.125-inch diameter pitot probe with a 0-9 in. WC manometer was used to study recirculation zones.

Figure 16. Test Apparatus Installation.

Test Procedure

For the evaluation of the configurations discussed in this report two types of data were obtained. First, oxidizer recirculation zones were defined by probing and determining dynamic pressure profiles without fuel or water flow. Secondly, flow patterns of a fuel simulating medium were observed (with oxidizer flow present) and documented by photography.

The oxidizer flow rate was established prior to recording data. The optimum oxidizer flow rate was predetermined to be 0.25 lbm/sec to simulate the full scale combustor inlet Reynolds number.

Table 3 shows a comparison of oxidizer conditions of the full scale combustor and the isothermal model.

When testing to define recirculation zones only oxidizer flow was maintained. When observing visual fuel flow patterns, the most successful results were obtained by flowing water through the fuel injector. Also some attempts were made to utilize pulverized coal, as well as other solid particulate matter, with more desirable properties than coal. The major difficulty encountered with solid fuel was maintaining a steady predetermined flow rate. Still photography was not as revealing when solid fuel particles were used because coal coated the plexiglas.

The fuel flow simulation was operated over a range of flow to study the effects of different fuel flows. When using water the particle Reynolds number of coal

TABLE 3
OPERATING CONDITIONS

Specification	4 lbm/sec Combustor	Cold Flow Combustor
Oxidizer Type	Vitiated O ₂ and N ₂ mixture (85% O ₂)	Air
Oxidized Inlet Temperature	1367 °K	520°K
Combustor Pressure	75 psia	14.7 psia
Oxidizer Flow	2.8 lbm/sec	.25 lbm/sec (nominal)
Inlet Oxidizer Re	8.6 x 10 ⁴	8.6 x 10 ⁴

could not be achieved since the mean water droplet diameter was not known for the apparatus used. When simulating fuel flow with water, flow patterns were not affected by varying fuel flows. Once this was established, fuel flows were set in a range to reproduce the O/F and momentum ratio of the oxidizer to fuel in the full scale combustor. The flows were set by measuring static pressure at an upstream point of the fuel injector (Figure 17).

When using a solid fuel a small chamber containing the fuel was pressurized to 30 psig. A ball valve located downstream of the fuel chamber was opened, which allowed the fuel to flow for 2-3 seconds. Pulverized coal proved to be the best solid particulate that was tried. Based on a measured chamber volume and an average bulk density of pulverized coal of 40 lbm/ft^3 , the mass flow ratio of the oxidizer and the coal was of the correct order of magnitude to give a similar O/F ratio to that of the full scale combustor.

Figure 17. Fuel Injector Flow Calibration Curve Using Water.

CHAPTER VI

EXPERIMENTAL RESULTS

Defining of Recirculation Zone

One of the most revealing results of the cold flow tests was that of determining recirculation zones. In general, all oxidizer swirlers set up a primary reverse flow zone located about the combustor centerline. Also a smaller (secondary) zone was detected at the outer diameter and beginning at the upstream end of the combustor while utilizing all swirler configurations (Figures 18 through 20).

The baseline case that was tested (high axial velocity oxidizer injector) established a very small recirculation zone at the upstream end and on centerline location of the combustor. The most intense recirculation zone was probed near the outer diameter of the flow region with the high velocity injector. This reverse flow zone was, by far, the most intense of any configuration (Figure 21).

With each swirler tested, a significant difference in recirculating flow profiles was established. These profiles can be related to relative degrees of oxidizer swirl intensity.⁵ Figures 18 through 20 show recirculation profiles. The tests conducted with fuel flow did not

$$\dot{m}_{AIR} = .247 \text{ lbm/sec}$$

$$Re = 8.889 \times 10^4$$

Design Swirl No. = 1.4

Shaded areas represent recirculation zones.

Figure 18. Swirler TEI-0127-3 Recirculation Zones.

$$\dot{m}_{AIR} = .235 \text{ lbm/sec}$$

$$Re = 8.4932 \times 10^4$$

$$\text{Design Swirl No.} = 2.45$$

Shaded areas represent recirculation zones.

Figure 19. Swirler TEI-0128 Recirculation Zones.

$$\dot{m}_{AIR} = .243$$
$$Re = 8.7452 \times 10^7$$

Design Swirl No. = 3.28

Shaded areas represent recirculation zones.

Figure 20. Swirler TEI-0129 Recirculation Zones.

High Velocity Injector TEI-0130

$$\dot{m}_{\text{AIR}} = .226$$

$$\text{Re} = 8.1333 \times 10^4$$

Shaded areas represent recirculation zones.

Figure 21. Baseline Case Recirculation Zones.

establish fuel flow in the upstream direction in the primary recirculation zone of any of the swirlers. Based on the observations made, only high speed motion picture photography could detect this upstream flow of fuel particles if it exists with the swirlers. However, the dispersion pattern of fuel closely adhered to the shape of the primary recirculation zones of the swirlers (Figures 22 through 24). Note that these photographs were taken without the Plexiglas enclosure to allow observations of flow at the fuel injector only.

Observations of the recirculation near the combustor outer diameter were easily seen with all configurations. This flow agreed with the dynamic pressure profiles as well. The swirler configurations showed very little fuel flowing upstream in the secondary recirculation zone. The high velocity injector configuration showed very intense fuel recirculation in the outer zone. Also at maximum air and fuel flows (simulating fuel with water) the flow was unstable with the base case injector. The two extremes of outer recirculation are shown in Figures 25 and 26. (Note relative axial locations of the fuel cone with the acrylic tube.) Figure 27 shows the central recirculation zone utilizing the high velocity injector.

Figures 28 and 29 show the recirculation zone boundaries for all of the configurations tested.

Figure 22. Primary Recirculation Zone Using Swirler TEI-0127-3.

Figure 23. Primary Recirculation Zone Using Swirler TEI-0128.

Figure 24. Primary Recirculation Zone Using Swirler TEI-0129.

Figure 25. Outer Flow Field Recirculation Zone Using Baseline Injector TEI-0130 Operating in the Unstable Mode with Higher Recirculating Fuel Flow.

Figure 26. Outer Flow Field Recirculation Zone Using Baseline Injector TEI-0130 Operating in the Unstable Mode with Decreased Recirculating Flow.

Figure 27. Centerline Recirculation Zone Using the Baseline Injector TEI-0130.

Figure 28. Recirculation Zone Boundaries for Centrally Located Zones.

Figure 29. Recirculation Boundaries for Outer Zones.

Dynamic Pressure Profiles

By utilizing the pitot probe and an inclined manometer, a crude dynamic pressure profile was obtained by traversing the combustor duct. Dynamic pressure measurements were taken for both upstream and downstream flow directions. Although the pitot probe was not designed for measurements in flow of the type considered, it is believed that these data are meaningful. This belief was based on the favorable comparison of recirculation zones located with the pitot probe and the flow patterns observed with fuel being injected. Table 4 is a comparison of configurations tested.

It is emphasized that these data should be used with caution. These data are indicative of only one axial location and are not indicative of recirculation upstream of that axial location at which the data were acquired.

Swirl Number Correlation

Based on general trends established by Syred and Beér, the relative primary recirculation zone shape is strongly related to the relative swirl number. Comparing Figure 30 to the recirculation zones shown in Figure 28, it appears that swirler TEI-0128 does not agree with the expected trends for the design swirl numbers. Based on measured swirler static pressure losses and oxidizer flow

TABLE 4
OXIDIZER INJECTOR RECIRCULATING FLOW PARAMETERS

Injector No. and Type	Axial Location of Velocity Measurement	Recirculation Velocity Range	$\dot{m}_R/\dot{m}$
TEI-0127-3 (Swirler)	2 Inches	49.0-81.0 (Ft/sec)	.511
TEI-0128 (Swirler)	2 Inches	104.0-115.0 (Ft/sec)	.113
TEI-0129 (Swirler)	2 Inches	20.9-36.2 (Ft/sec)	.192
TEI-0130 (High Velocity)	2 Inches	105.0-162.0 (Ft/sec)	2.863

Figure 30. Recirculation Zone Boundaries for Different Swirl Numbers.⁵

rates, actual swirl numbers were calculated utilizing the following equations:

1. Momentum equation¹⁸ $\Delta P A_R = \dot{m} \Delta V$

2. Conservation of mass equation¹⁴ $\dot{m} = \rho U_D A_D$

3. Swirl number equation⁶ $S = \frac{2}{3} \left[\frac{1 - \left(\frac{r_I}{R_O} \right)^3}{1 - \left(\frac{r_I}{R_O} \right)^2} \right] \tan \theta$

where

ΔP = Swirler static pressure drop

A_R = Restricting flow area of swirler

$\dot{m}$ = Mass flow rate through swirler

ΔV = Change in oxidizer velocity from just upstream of the swirler inlet to the discharge of the swirler

ρ = Oxidizer density at the swirler discharge

U_D = Oxidizer mean axial velocity component at swirler discharge

A_D = Swirler discharge area

r_I = Swirler annulus inner radius

R_O = Swirler annulus outer radius

θ = Angle of oxidizer resultant velocity with respect to the combustor axis

The swirl numbers calculated from experimental data are shown in Table 5. The experimentally determined swirl number assumes that only axial and tangential velocity components are present at the swirler discharge. Note that if the experimental values are related to corresponding recirculation zone locations, a reasonable trend is established.

The reasons for the large variations from design swirl numbers are not fully understood. However, in the case of swirler TEI-0128, the reason is attributed to the small swirler discharge area (refer to Figure 31). Also a possible explanation for swirler TEI-0129 can be related to vane geometry. Possibly flow separation occurred due to low inlet area L/D values or due to square leading edges of the vanes.

Combustor Oxidizer Injector Pressure Loss

The static pressure losses were measured for each configuration. This parameter was considered mainly because increased pressure loadings at upstream locations result from higher oxidizer injector pressure losses. The oxidizer injector pressure loss is an important factor to be considered for the commercial scale combustor design (Figure 32).

TABLE 5
A COMPARISON OF DESIGN AND
MEASURED SWIRL NUMBERS

Swirler	Design S	Experimentally Determined S
TEI-0127-3	1.4	1.48
TEI-0128	2.45	.65
TEI-0129	3.28	1.74

Inlet Area	Discharge Area
○	● Swirler TEI-0127-3 , S = 1.48
△	▲ Swirler TEI-0128 , S = .65
◇	◆ Swirler TEI-0129 , S = 1.74

For $8 \times 10^5 \leq Re \leq 9 \times 10^5$

Figure 31. Swirler Flow Area vs. Pressure Coefficient for the Three Swirler Configurations.

Figure 32. Normalized Combustor Pressure Drop vs. Oxidizer Mass Flow Rate.

CHAPTER VII

CONCLUSIONS

As a result of an isothermal model of the pulverized coal fired MHD combustor, significant advances were made toward the understanding of the aerodynamic behavior of the combustion reactants, particularly the introduction of O_2/N_2 , and in providing a base for future work. The primary configurations that were studied induced swirling flow at the combustor inlet. The reason for studying the concept of swirlers was due to the well established effects that swirling flow has demonstrated in conventional combustors.

Swirlers

Considering the three swirlers that were tested, the following conclusions were made:

1. All three swirlers induced a primary recirculation zone symmetrically located about the combustor axis.
2. All three swirlers induced a secondary recirculation zone at the upstream end of the combustor and near the outer diameter of the flow field of the combustor.
3. Two of the swirlers demonstrated swirl characteristics that fell short of each expected design swirl number.

4. For a swirler to induce the recirculation of the magnitude observed with the base case, either a larger combustor diameter will be required to reduce oxidizer velocities or a higher pressure drop will result from the necessity of increasing the swirl number (Figure 33).
5. The swirlers tested did not offer improved mixing when compared to the base case.
6. To obtain a satisfactory swirler design, the swirler inlet to outlet area ratio should be no greater than 1.0.
7. Swirl vane configurations appear to be an important factor in swirler design for establishing a satisfactory swirl number.

Base Case

Considering the base case of the high velocity oxidizer injector, the following conclusions can be made:

1. Extremely high velocity gradients were observed.
2. The recirculation zone primarily existed near the outer walls of the flow field and axially located from the oxidizer injector plate to about 1.8 combustor diameters downstream.
3. Primary mixing of the oxidizer and fuel components occurs very near the discharge of

Figure 33. Swirl Number vs. Pressure Coefficient for All Configurations Tested.

the oxidizer jet at the downstream face of the injector.

Based on the data acquired during this test series, mixing lengths of the various configurations could not be defined. However recirculation zones were located.

Based on the consistency of the data obtained during tests with the cold flow combustor model, the usefulness of this experimental approach appears promising.

CHAPTER VIII

RECOMMENDATIONS

The usefulness of a cold flow aerodynamic combustor model is to serve as an integral part of a combustor development program. By using an isothermal model, as well as other intermediate stages of testing, the chance of success of full scale tests is increased (Figure 34). In an effort to apply the work of this report to a combustor development program, the following recommendations are made:

1. The investigation of oxidizer swirlers should be continued by studying recirculation flow characteristics with higher swirl numbers in the cold flow model.
2. The additional phase of studying effects of unrestricted flow area fuel injectors should be initiated utilizing the existing cold flow model.
3. An investigation should be made to utilize a Laser-Doppler velocimeter with the cold flow model to obtain more accurate turbulent flow data.
4. A study should be initiated to optimize the high velocity oxidizer injector design.

Figure 34. Combustor Development Program Flow Chart.

5. The investigation of requirements needed to establish an atmospheric liquid fuel fired combustor (utilizing as much hardware as possible of the cold flow model) to allow the study of flame stability should begin.
6. Provided that swirling flow oxidizer techniques are strongly considered for a full scale combustor, a numerical solution of at least the isothermal flow field should be developed.

By initiating these recommendations, a positive approach to establishing a successful MHD combustor with possible alternate configurations should result.

LIST OF REFERENCES

1. Dicks, J. B. "Review of Central Power Magnetohydrodynamics," AIAA Paper 75-264.
2. Chapman, J. N. "On the Economic Optimization of the Magnetohydrodynamic-Steam Power Plant," Dissertation presented for the Doctor of Philosophy degree, The University of Tennessee, Knoxville, December 1977.
3. Heywood, J. B., and G. J. Womack. Open Cycle MHD Power Generation, Pergamon Press, Ltd., 1969.
4. Postlethwaite, A. W., and M. M. Sluyter. "Heat Transfer Problems Associated with an MHD Power Generation System -- An Overview," ASME Paper 77-HT-62.
5. Syred, N., and J. M. Beér. "Combustion in Swirling Flow: A Review," Combustion and Flame 23, 143-201, 1974.
6. Beér, J. M., and N. A. Chigier. Combustion Aerodynamics, Halsted Press Division, John Wiley & Sons, Inc., New York, 1972.
7. Shanklin, R. V., et al, "The UTSI Coal Burning MHD Program," Proc. 13th Symp. on Engineering Aspects of MHD, Stanford University, Stanford, Calif., March 1973.
8. Liepmann, H. W., and A. Roshko. Elements of Gasdynamics, John Wiley & Sons, Inc., New York, 1967.
9. Sarpkaya, Turgut. "On Stationary and Travelling Vortex Breakdowns," Fluid Mechanics, Vol. 45, Part 3, pp. 545-559, 1971.
10. Sarpkaya, Turgut. "Effect of the Adverse Pressure Gradient on Vortex Breakdown," AIAA Journal, Vol. 12, No. 5, May 1974.
11. Bossel, Hartmut H. "Swirling Flows in Streamtubes of Variable Cross Section," AIAA Journal, Vol. 11, No. 8, August 1973.
12. Bossel, Hartmut H. "Vortex Breakdown Flowfield," The Physics of Fluids, Vol. 12, No. 3 March 1969.

13. Prudmkov, A. G., M. S. Vaynskiy, and V. N. Sagalovich. Mixing and Combustion Progress in Jet Engines, Wright-Patterson AFB, Foreign Technology Division, Dayton, Ohio, 1973.
14. Lilley, D. G. "Modeling of Combustor Swirl Flow," ACTA Astronautica, Vol. 1, pp. 1129-1147, September 1971.
15. Hill, P. G., and C. R. Peterson. Mechanics and Thermodynamics of Propulsion, Addison-Wesley Publishing Co., Inc., Reading, Massachusetts, 1970.
16. Engineering Division, Crane Co. "Flow of Fluids through Valves, Fittings, and Pipes," Crane Technical Paper No. 410, Crane Industrial Products Group, Chicago, 1957.
17. Carlson, P. K. M.S. Thesis in Engineering Science, The University of Tennessee, Knoxville, 1978.
18. Zucker, Robert D. Fundamentals of Gas Dynamics, Matrix Publishers, Inc., Champaign, Illinois, 1977.