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# TECHNICAL PROGRESS REPORT

## January 1994 - March 1994



For:

**U.S. Department of Energy**
**Morgantown Energy Technology Center**

Under:

**DOE Contract No. DE-AC21-89MC26288**

## Sonic Enhanced Ash Agglomeration and Sulfur Capture

By:

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## PREFACE

This 19th Quarterly Technical Progress Report presents the results of work accomplished during the period January 3, 1994 through March 27, 1994 under Contract No. DE-AC21-88MC26288 entitled "Sonic Enhanced Ash Agglomeration and Sulfur Capture." The fundamental studies conducted by West Virginia University and Penn State University are provided in Subsections 2.2 and 2.3.

During this period, an isokinetic sampling system containing three high efficiency cyclones were designed and fabricated to overcome problems in particle size distribution analysis sampling. These were used during the tests of a Pittsburgh Seacoal during the end of the period. This test was conducted at steady state over an eight-hour period. Analysis of the samples have not been completed and the results will be reported during the next period.

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# SECTION 1.0

# INTRODUCTION

## 1.1 PROJECT DESCRIPTION AND WORK STATUS

A major concern with the utilization of coal in directly fired gas turbines is the control of particulate emissions and reduction of sulfur dioxide, and alkali vapor from combustion of coal, upstream of the gas turbine. Much research and development has been sponsored on methods for particulate emissions control and the direct injection of calcium-based sorbents to reduce $SO_2$ emission levels. The results of this research and development indicate that both acoustic agglomeration of particulates and direct injection of sorbents have the potential to become a significant emissions control strategy.

The Sonic Enhanced Ash Agglomeration and Sulfur Capture program focuses upon the application of an MTCI proprietary invention (Patent No. 5,197,399) for simultaneously enhancing sulfur capture and particulate agglomeration of the combustor effluent. This application can be adapted as either a "hot flue gas cleanup" subsystem for the current concepts for combustor islands or as an alternative primary pulse combustor island in which slagging, sulfur capture, particulate agglomeration and control, and alkali gettering as well as $NO_x$ control processes become an integral part of the pulse combustion process.

The goal of the program is to support the DOE mission in developing coal-fired combustion gas turbines. In particular, the MTCI proprietary process for bimodal ash agglomeration and simultaneous sulfur capture will be evaluated and developed. The technology embodiment of the invention provides for the use of standard grind, moderately beneficiated coal and WEM for firing the gas turbine with efficient sulfur capture and particulate emission control upstream of the turbine. The process also accommodates injection of alkali gettering material if necessary. This is aimed at utilization of relatively inexpensive coal fuels, thus realizing the primary benefit being sought by direct firing of coal in such gas turbine systems. The proposed technology provides for practical, reliable, and capital (and O&M) cost-effective means of protection for the gas turbine from impurities in the coal combustor effluent.

## 1.2 PROGRAM OBJECTIVES

The major objective of the Phase I test program is to confirm the feasibility of the MTCI bimodal particle size approach to enhance particulate control by acoustic ash agglomeration. An ancillary objective of the Phase I effort is to demonstrate and confirm the feasibility of an acoustic field to enhance sulfur capture by increasing sorbent reactivity. Phase I tests are designed to cover the frequency range between 50 and 1400 Hz, establish monomodal baseline performance as a benchmark from which to measure the degree of enhancement expected from the bimodal approach, and, finally, to confirm the effectiveness of low-frequency fields over high-frequency fields for realistic particulate streams.

The program will demonstrate the effectiveness of a unique approach which uses a bimodal distribution composed of large sorbent particles and fine fly ash particles to enhance ash agglomeration and sulfur capture at conditions found in direct coal-fired turbines. Under the impact of high-intensity sound waves, sorbent reactivity and utilization, it is theorized, will increase while agglomerates of fly ash and sorbents are formed which are readily collected in commercial cyclones. The work will extend the concept from the demonstration of feasibility (Phase I), through proof-of-concept (Phase II) to the construction (Phase III) of a coal-fired pulsed combustor with in-furnace sorbent injection. For Phase I, Pennsylvania State University will conduct studies for enhanced sulfur capture in The Combustion Laboratory and agglomeration tests in the High Intensity Acoustic Laboratory.

## 1.3 SUMMARY STATUS FOR THE PERIOD

During this period, an isokinetic sampling system containing three high efficiency cyclones were designed and fabricated to overcome problems in particle size distribution analysis sampling. These were used during the tests of a Pittsburgh Seacoal during the end of the period. This test was conducted at steady state over an eight-hour period. Analysis of the samples have not been completed and the results will be reported during the next period.

# SECTION 2.0

# TECHNICAL DISCUSSION OF THE WORK ACCOMPLISHED DURING THE REPORTING PERIOD

## 2.1 TASK 1: SHAKEDOWN TESTING

SCREENING TESTS

Tests performed on the bimodal system during the previous reporting period (December 1993) showed high sulfur capture (95%) and low flue gas solids loading after the second cyclone (as low as 32 ppmw). It was, however, impossible to take representative (uncaked and dry) solids samples for particle size distribution analysis. For this reason a new isokinetic sampling system was designed, fabricated and installed after pressure letdown valve. Figure 1 shows the schematic diagram of the sampling system. Three high efficiency cyclones are used to collect samples for size analysis of particles and paper filter is used to collect particles for mass balance.

Pittsburgh pulverized grade 5 Seacoal feed test was performed on the bimodal unit. Eight hours of steady-state condition was sustained. Ash samples from the bottom of the agglomeration chamber, primary cyclone catch, secondary cyclone catch and three high efficiency cyclones were collected after the test and sent for Bahco size analyses. Ash samples from the primary cyclone catch and agglomeration chamber catch were also sent for ultimate analysis. Separate isokinetic samples from the 6-inch pipe to the baghouse and from the pipe downstream of the three high efficiency cyclones were collected. Both flue gas streams show nearly identical solids loading (42 and 40 ppmw). Detailed data will be presented after size and chemical analyses results are received.

During this reporting period, 715 pounds of calcined and classified Anville lime were also prepared in the MTCI PAFBC system for bimodal tests.

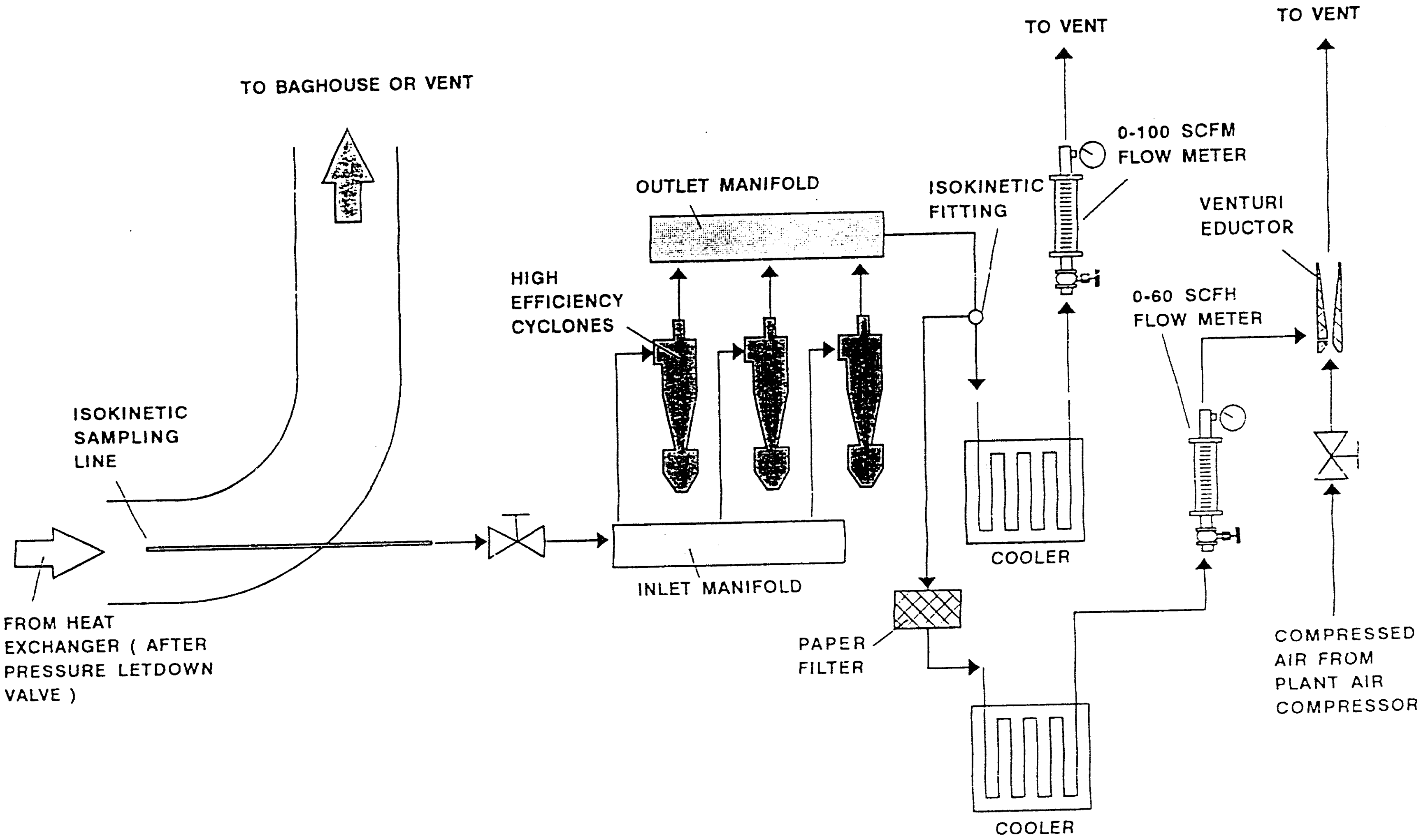


FIGURE 1: FLUE GAS SOLIDS SAMPLING SYSTEM

## 2.2 AEROVALVE TEST (WEST VIRGINIA UNIVERSITY)

### 2.2.1 AEROVALVE DESIGN

EXPERIMENTAL PERFORMANCE OF THE VORTEX AEROVALVE

Performance tests of the vortex aerovalve presented in the October 1993 report were completed during the month of January. This valve will be designated AV5C1-N4, where AV is for aerovalve, 5 in the series number of the valve (series 3 and 4 were tested in the spring of 1993), C1 is the current configuration, and N4 is the number of throats. Future valves will be referenced using this designation. A slight modification was made to the valve in the form of a larger forward flow inlet radius $r_i$. The maximum value of $r_i$ yielded by the formulas $2\pi r_i s = Ns^2A_r$ and $\pi r_i^2 = Ns^2A_R$ was desired from previous design considerations so the inlet diameter was increased from 1 inch to 1 3/16 inch.

PROCEDURE

Two sets of performance tests were conducted: one on the venturi-type aerovalve (for comparison purposes) and one on the vortex aerovalve. The performance tests were conducted using water. Figure 2 is a diagram of the vortex aerovalve and venturi aerovalve test configurations. Figures 3 and 4 show the test setup and a close-up of the vortex aerovalve. Figure 5 is a diagram of the water receiver detailing its use. The venturi aerovalve (0.2 in$^2$ throat area, 0.36 in$^2$ exit area, $A_R$ = 1.8) was placed in forward flow on the water receiver and a pressure heat of 14 inches was measured and recorded. The water receiver level was measured using a yardstick attached to the receiver. The mass flow rate through the valve was measured by closing the drain valve on the water receiver, noting the water height and, after ten seconds, measuring and recording the water height. The same test was repeated for the valve in reverse flow.

The vortex aerovalve was tested in the same manner. The combined throat area of the four venturi passages was 0.95 in$^2$ and the exit area was 1.552 in$^2$, giving an area ratio $A_R$ = 1.63. For the forward flow test the valve was placed on the receiver and a pressure head $\Delta h_F$ of 8 inches was measured and recorded. The pressure head $\Delta h_F$ was measured from the top of the throat exit plane to the

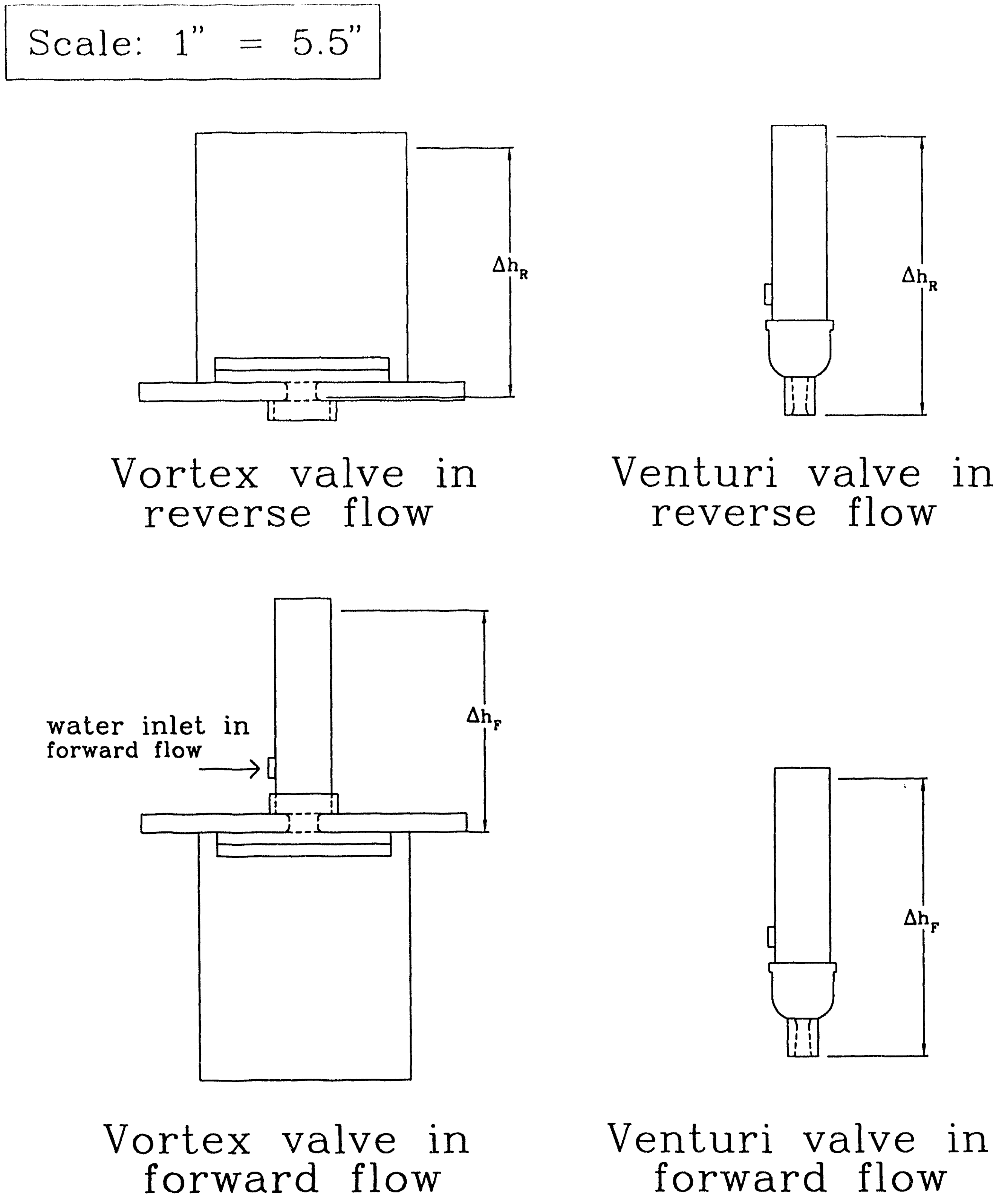


**FIGURE 2: VORTEX AND VENTURI AEROVALVE EXPERIMENTAL CONFIGURATION**

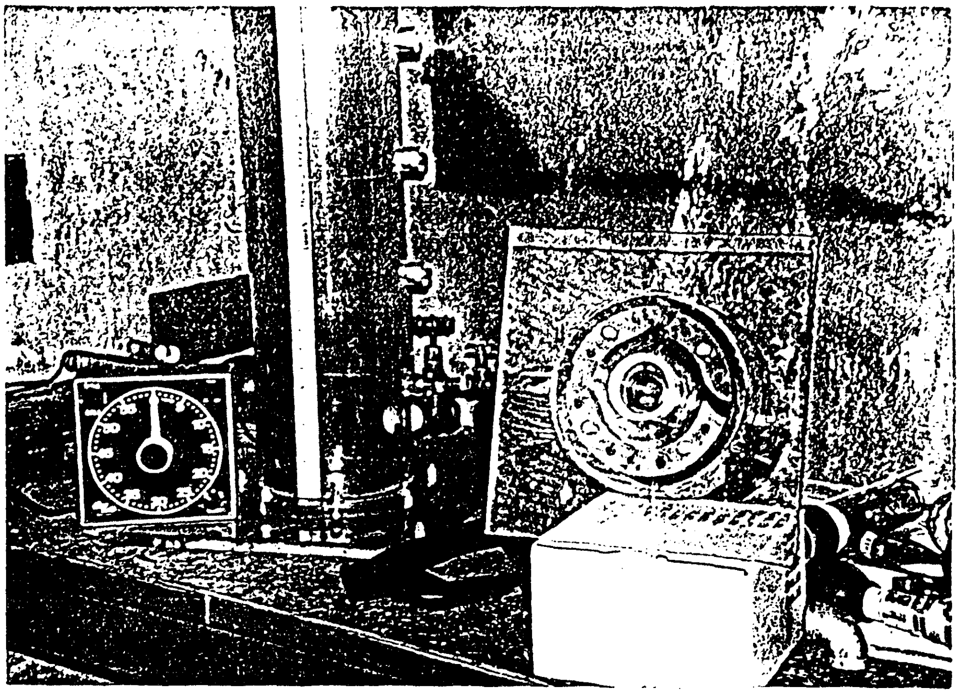

FIGURE 3: EXPERIMENTAL APPARATUS SHOWING WATER RECEIVER, CLOCK AND VORTEX AEROVALVE

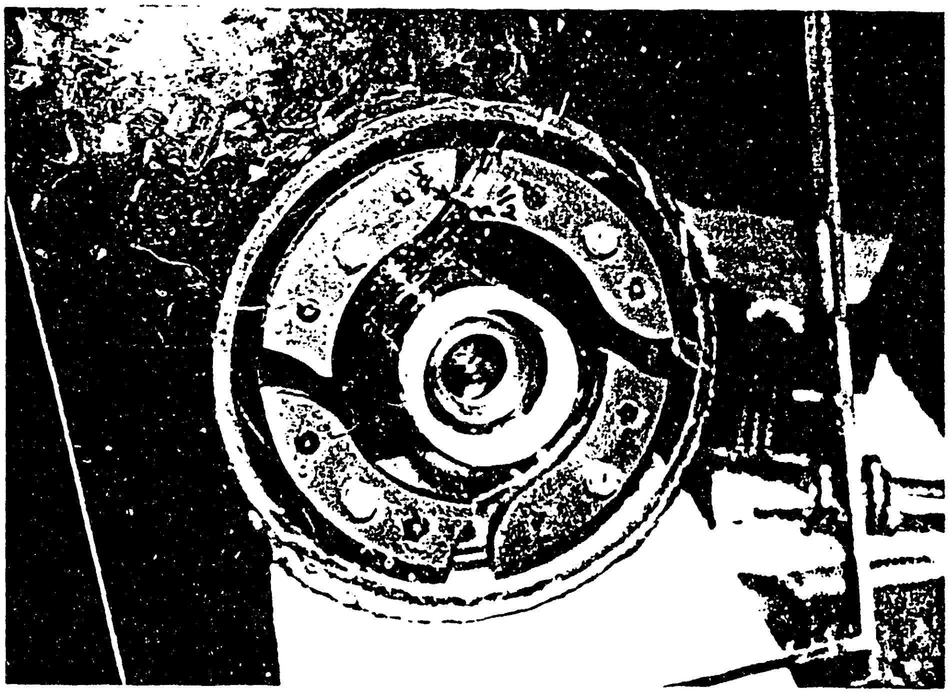

FIGURE 4: CLOSE-UP OF VORTEX AEROVALVE SHOWING INLET AND VORTEX CHAMBER

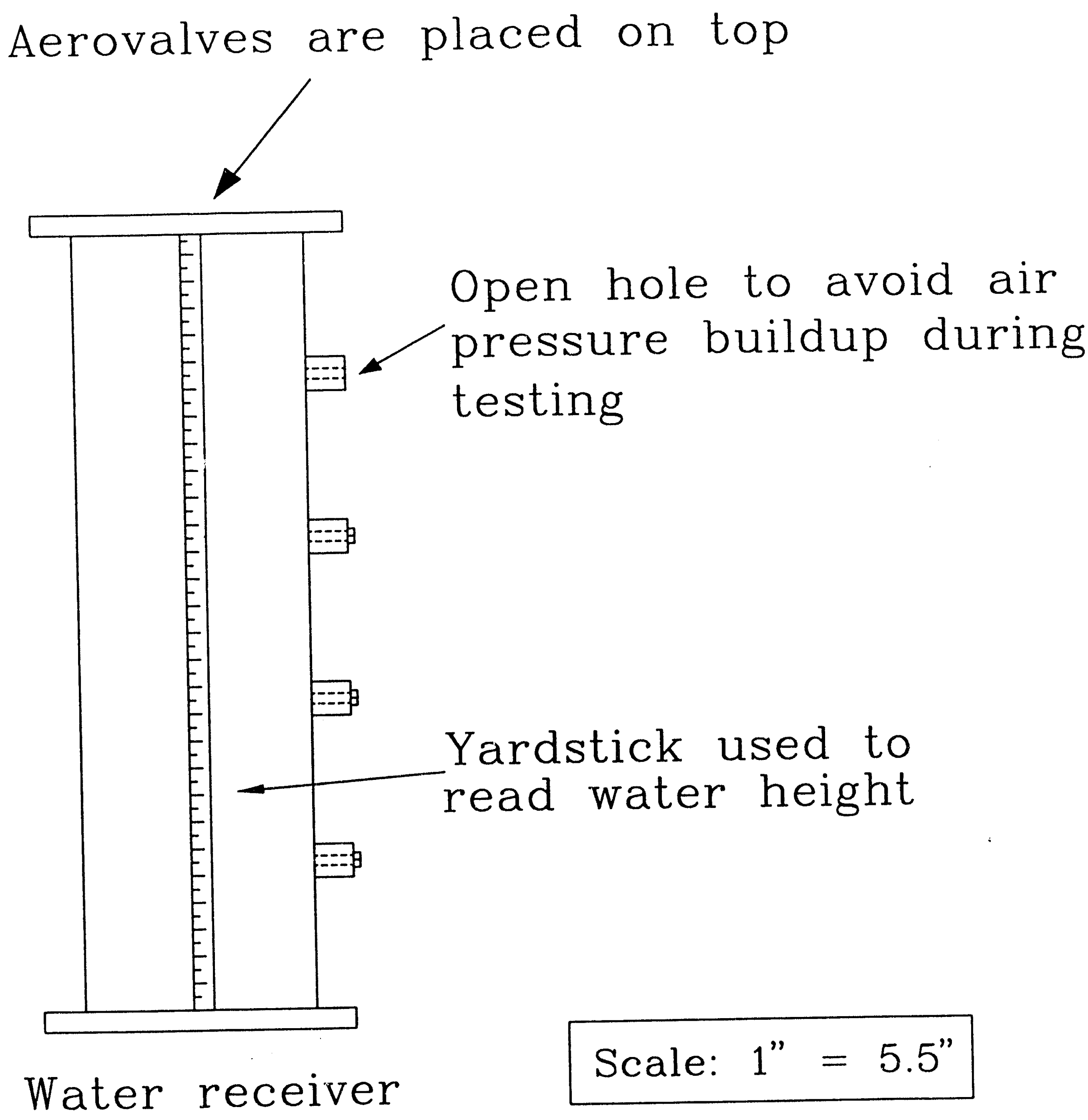


**FIGURE 5: WATER RECEIVER DETAIL**

free surface (see Figure 2). The mass flow rate was determined as in the venturi test. The same procedure was followed to measure the reverse flow rate but the pressure heat height $\Delta h_R$ was measured from a point 1/8" deep in the exit throat to the reservoir free surface (see Figure 2).

## RESULTS

The results of the performance test of the venturi aerovalve were:

For a 14-inch pressure head:

Forward flow:
5-inch water height change in 10 seconds = 0.5 in/s

Reverse flow:
3-inch water height change in 10 seconds = 0.3 in/s

The diodicity is the direct ratio of forward to reverse water height change velocity. This is because the mass flow ratio equals the height change ratio, and

$$\text{Venturi diodicity } D = 0.5/0.3 = 1.66$$

The results of the performance test of the vortex aerovalve were:

Forward flow:
7.5-inch water height change in 10 seconds = 0.75 in/s

Reverse flow:
3.0-inch water height change in 10 seconds = 0.3 in/s

The diodicity is again the direct ratio of the forward to reverse water height change velocities:

$$\text{Vortex diodicity } D_v = 0.75/0.3 = 2.5$$

## CONCLUSIONS

An initial computer estimate limited the diodicity for this configuration to $D_v \leq 3.5$. The diodicity of the vortex valve AV5C1-N4 was experimentally determined to be 2.5, which is 29% lower. To reduce friction in the forward flow, the sharp corners in the venturi passages and vortex chamber will be rounded off by filleting them with silicon rubber. The results of this modification as well as Reynolds number considerations and efficiency factors will be presented in the next report.

## FRICTION IN THE VORTEX AEROVALVE

It is of interest to calculate the resistance to forward flow produced by both the venturi and vortex aerovalves and compare the results. For the vortex aerovalve using the results from the water tests in the last report:

Water receiver cross-sectional area with $r_{wr}$ = 4 inches is

$$A_{wr} = 50.26 \text{ in}^2$$

Forward flow volumetric flow rate is

$$Q_{vF} = V_{wr}A_{wr} = (0.75 \text{ in/s})(50.26 \text{ in}^2) = 37.7 \text{ in}^3/\text{s}$$

Forward flow exit velocity $V_F$ based on $Q_{vF}$ and the vortex valve exit area $A_e$ is

$$V_F = (37.7 \text{ in}^3/\text{s})/(1.6 \text{ in}^2) = 23.56 \text{ in/s} = 2 \text{ ft/s}$$

Dynamic pressure $q_F$ is

$$\tfrac{1}{2}\rho V_F^2 = (0.5)(2 \text{ slug/ft}^3)(2 \text{ ft/s})^2 = 4 \text{ psf} = 0.769 \text{ inches } H_2O$$

The percent of the pressure head lost is then 1 - (0.769"/8") = 0.9 = 90%. The same calculation using the venturi aerovalve results shows that the percent of pressure head lost is 1 - (6.5"/14") = 0.54 = 54%, or 1.66 times less than in the vortex aerovalve.

The effect of friction on the diodicity of the vortex valve can be examined using the following procedure. First, the pressure drop in forward flow is written as and the reverse flow pressure drop as

$$9\Delta p_F = \frac{1}{2}\rho V_F^2\left(1 + f_{vF}\right) \tag{1}$$

By the definition of diodicity these two pressure drops are equal ($\Delta p_F = \Delta p_F$), equating and solving for $f_{vR}$,

$$f_{vR} = \left[\frac{V_F}{V_R}\right]^2 (1 + f_{vF}) - 1 \tag{2}$$

Note the diodicity is defined as $D_v = \dot{m}_F/\dot{m}_R = V_F/V_R$ because both velocities have been defined at the same exit area $A_e$, so

$$\Delta p_R = \frac{1}{2}\rho V_R^2\left(1 + f_{vR}\right) \tag{3}$$

$$f_{vR} = D_v^2 (1 + f_{vF}) - 1 \tag{4}$$

The value of $f_{vF}$ can be directly calculated from Equation 1. For example, using the test data from the February report, $\Delta p_F$ = 8" $H_2O$ = $\frac{1}{2}\rho V_F^2(1 + f_{vF})$ and solving for $f_{vF}$ = 8/0.769 - 1 = 9.4. The reverse flow friction coefficient from Equation 4 is $f_{vR} = (2.5)^2(1+9.4) - 1 = 64$. The forward flow friction coefficient $f$ of the venturi aerovalve AV3C1 using the test data from the February report was found as in Equation 1 with $\Delta p_F$ = 14" $H_2O$ = $\frac{1}{2}\rho V_F^2(1 + f)$, and solving for $f$ = 14/6.5 - 1 = 1.15. A comparison shows the vortex aerovalve forward flow friction coefficient is 9.4/1.15 = 8.1 times higher than the venturi coefficient; therefore friction is significant in the vortex aerovalve forward flow.

To get the same pressure loss in forward flow using the vortex aerovalve (with sharp edge 2-D venturi passages) as in the simple venturi valve, the passage areas must be increased by a factor of $\sqrt{(f_{vF}/f)} \approx 3$, or the diameter by a factor of $\sqrt{3} = 1.7$. The reasoning is that since $\Delta p \approx fV^2$, as area increases threefold, V drops threefold and $\Delta p$ drops (threefold)$^2$.

Another method to calculate the required area increase is to note that the ratio of vortex-to-venturi exit areas is 1.6/0.36 = 4.44 and the ratio of vortex-to-venturi volumetric flow rates at an 8 inch pressure head is $Q_{F,V}/Q_F$ = 0.75/(0.5·(8"/14")) = 2.625. Thus to make the $\Delta p_F$ for the vortex valve the same as the venturi, all areas must be increased by a factor of 4.44/2.625, or 1.7. This means that a vortex valve with a total throat area of $Ns^2$ = 1.7 in$^2$ has the same friction loss as a venturi valve with a 1.128 inch diameter throat which has $A_{th}$ = 1.0 in$^2$. The friction coefficient obtained here from the water test may be higher than would be found in an air test. To verify this the Reynolds numbers are compared.

## REYNOLDS NUMBER CONSIDERATIONS

To compare the results obtained using water to what they would be using air the Reynolds number of the forward flow in the vortex aerovalve must be considered. The Reynolds number is defined as

$$Re = \frac{\rho V_{F,th} D_H}{\mu} \tag{5}$$

and is here calculated at the throat. The velocity $V_{F,th}$ is simply the previously calculated exit velocity times the area ratio, $V_{F,th}$ = (2 ft/s)(1.6) = 3.2 ft/s. The quantity $D_H$ is the hydraulic radius, defined as 4 times the throat area divided by the throat perimeter or $D_H$ = (4)(0.2375 in$^2$) / (1.95 in) = 0.5 in = 1/24 ft. The water density is taken as 2 slug/ft$^3$ and the absolute viscosity $\mu$ = 2.1 x 10$^{-5}$ lb·s/ft$^2$. The Reynolds number using Equation 5 is Re = (2)(3.2)(1/24) / (2.1 x 10$^{-5}$) = 12698. This number is greater than 2000, which indicates turbulent flow in the vortex aerovalve throat.

To obtain the same Reynolds number in air as with water, the velocity must be increased as follows. Using the values for air of $\rho$ = 0.002377 slug/ft$^3$ and $\mu$ = 3.76 x $10^{-7}$ lb·s/ft$^2$ and solving for the velocity from Equation 5 the throat velocity is $V_{F,th}$ = (12698)(3.76 x $10^{-7}$ lb·s/ft$^2$) / (1/24 ft)(0.002377 slug/ft$^3$) = 48 ft/s. The exit velocity $V_{F,e}$ = 48/1.6 = 30 ft/s. The dynamic pressure based on this exit velocity is $(0.5)(0.002377)(30)^2$ = 1.07 psf. This is equivalent to 1.07/5.2 = 0.205 inches of water. In the usual air tests on venturi type aerovalves the test $\Delta p_F$ was as high as 60 inches of water, corresponding to an exit dynamic pressure of (0.46)(60) = 27.6 inches of water, which is 134 ( $\approx V^2$) times greater than 0.205 inches of water. Thus the air test Reynolds number ($\approx$ V) are 11.6 times higher than in water.

## CONCLUSIONS

The following conclusions based on the analysis above have been made:

- The throat Reynolds number of the water test is 12698 and much lower than it would be in the usual air test, at least by a factor of 100,

- Comparing friction coefficients for smooth pipe flow, the friction coefficient in an air test is likely to be 1.7 times lower than in the water test,

- In pulse combustors the effect of high friction in the vortex valve can be readily compensated for by a slight increase in cross-sectional areas.

When using air to test the performance of the vortex aerovalve it is expected that the friction coefficients will be lower because of the higher Reynolds number. The performance of the vortex valve using air and the effect of rounding the sharp corners of the venturi passages will be presented during the next period.

## 2.3 FUNDAMENTAL SORBENT STUDIES (PENN STATE UNIVERSITY)

The objective of Task 1 is to determine the physical and chemical changes occurring in calcium-based sorbents when they are subjected to high heating rates for short residence times. Specifically, the aim is to determine if a flash calcination phenomenon occurs and, if so, whether it produces a highly reactive calcine.

The objective of Task 2 is to explore the concept of bimodal acoustic agglomeration of fly ash and sorbent particles. In the first stage, the acoustic frequency and sound pressure level in an entrained-flow reactor (EFR) will be optimized for a range of fly ash and sorbent mass loadings, particle sizes and reactor temperatures. The focus of the second stage will be to identify experimentally the mechanisms that control the acoustically induced bimodal agglomeration and cohesion of fly ash and sorbent particles.

In Task 3, the data generated in Tasks 1 and 2 will be incorporated into a model to predict sulfur capture and the extent of bimodal acoustic agglomeration under pulse combustion conditions. As experimental data become available from Tasks 1 and 2, progress on Task 3 will be reported.

### 2.3.1 TASK 1: FUNDAMENTAL STUDY OF SORBENT BEHAVIOR

Sulfation studies in the entrained-flow reactor were to be conducted at 1000° and 1100°C with the application of an acoustic field. The tests were not started as planned due to equipment design problems. During a preliminary test the horns for the sound drivers sintered when the reactor was heated to 1100°C. It is believed that the horns sintered for two reasons: exposure to radiant heat from the preheater and convective heat when the sound drivers were activated. The ports for the drivers are water-cooled; however, it appears that they are not cooled sufficiently to counter the heat produced by the reactor preheater.

One possible solution is to shield the drivers from the radiative section of the reactor by inserting a screen within the driver ports. This screen will shield the horns but not interfere with the acoustics. It is proposed that stainless steel mesh be inserted within the port to shield the horns. One problem is that the stainless mesh screen will heat up with time even though the port is water-cooled. Another procedural change is that the sound driver will be attached after the reactor has been preheated. In either case the goal is to extend the life of the horns. A series of trial runs using different mesh screens will be conducted at ambient temperature to determine if any interference of the acoustic field occurs. It is necessary to address this problem for both Tasks 1 and 2.

### 2.3.2 Task 3: Sulfur Capture Model

The sulfation model that has been developed is now being incorporated into the PGCG-2 combustion code with the following assumptions:

1. Since $SO_2$ pollutant concentrations are very low, only light loading of sorbent particles is necessary. Therefore, sorbent particles can be introduced after the overall convergence of the main combustion code. The underlying assumption is that the sources/sinks of mass, momentum and energy which will be introduced due to the present of sorbent particles into the system, will not affect the gas phase significantly.

2. Changes in the sulfur pollutant concentration are considered to have negligible impact on the sulfur-free gas composition. This is a reasonable assumption since the concentration of $SO_2$ comprises less than 0.5% of the gas stream.

3. Sulfur release from the coal is assumed to occur at a rate proportional to the coal weight loss for the combustion model which is solved prior to solving the sorbent model. Local instantaneous equilibrium is assumed for the homogeneous gas phase chemistry for the conversion of sulfur to sulfur dioxide. These assumptions enable the sorbent model to be decoupled from the main code.

4. Sulfation is considered to be irreversible.

## SECTION 3.0

## PLANS FOR NEXT PERIOD

- Complete data reduction for the tests already performed.
- Modify the slipstream solids sampling system downstream of the pressure letdown valve to obtain representative, uncaked solids sample for size analysis.
- Perform additional tests with coal and coal sorbent feed.

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