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FLUIDIZED-BED WASTE-HEAT RECOVERY
SYSTEM DEVELOPMENT

Final Report

By
Keith D. Patch
William E. Cole

June 1988

Work Performed Under Contract No. FC07-81ID12302

For
U. S. Department of Energy
Idaho Operations Office
Idaho Falls, Idaho

By
Tecogen, Inc.
Waltham, Massachusetts

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ABSTRACT

A major energy loss in industry is the heat content of the flue gases from industrial process heaters. One effective way to utilize the energy, which is applicable to all processes, is to preheat the combustion air for the process heater. Although recuperators are available to preheat this air when the flue gases are clean, recuperators to recover the heat from dirty and corrosive flue gases do not exist.

The Fluidized-Bed Waste-Heat Recovery (FBWHR) system is designed to preheat this combustion air using the heat available in dirty flue gas streams. In this system, recirculating alumina particles are heated by the flue gas in a raining bed. The hot particles are then removed from the bed and placed in a fluidized bed where they are fluidized by the combustion air. Through this process, the combustion air is preheated. The cooled particles are then returned to the raining bed. Initial development of this concept is for the aluminum smelting industry.

In this final report, the design, development, fabrication, and installation of a full-scale FBWHR system is detailed.

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1. EXECUTIVE SUMMARY

In many industries, stack gases from high-temperature process heaters are discharged directly to the atmosphere at high temperatures, wasting a significant fraction of the thermal energy input. One effective way to utilize this energy, which is applicable to all processes, is to preheat the combustion air for the process heater. Although recuperators are available to preheat this air when the flue gases are clean, recuperators to recover the heat from dirty and corrosive flue gases do not exist. Hence, most of this heat is lost with the flue gases.

The variety of unrecuperated process heaters that exists suggests that any advanced heat exchanger system that is developed should be widely applicable. Such a waste-heat recovery system would be suitable for any combustion air preheating process, and would not be limited to only specific industries or a few particular operating arrangements. Furthermore, a widely applicable heat exchanger system should allow for future upgrades in thermal performance and mechanical durability. Advances are going to be made in related technological areas, and newly developed recuperators should be able to take advantage of this new technology. Widely applicable systems that keep pace with scientific advances have the potential of recovering the largest amount of energy from industrial waste heat sources.

The Fluidized-Bed Waste-Heat Recovery (FBWHR) system is such a widely applicable recuperator, designed to preheat combustion air using the heat available in hostile flue gas atmospheres. In this system, recirculating alumina particles are heated by the flue gas in a hot stage raining bed. The hot particles are transferred to a cold-stage fluidized bed and fluidized by the combustion air. Through this process, the combustion air is preheated. The cooled alumina is

then returned to the first bed. Initial development of this concept is for the aluminum smelting industry.

Accomplishments of this development program include:

- Characterization of aluminum smelting furnace operations;
- Coupon testing of several materials in the flue of an aluminum smelting furnace;
- Successful, long-duration field-testing of alumina bed material under actual temperature and atmospheric conditions;
- Field-testing of a 1/10-scale fluidized-bed fouling and corrosion test unit;
- Field-testing of coated metallic distributor plates under actual stress, temperature, and corrosion conditions;
- Testing of distributor plate cleaning systems capable of overcoming the fouling process;
- Successful, long-duration, on-line, field-testing of a distributor plate cleaning system;
- Analytical development and experimental verification of the distributor plate stability criteria, fluidization characterization, and heat transfer characteristics;
- Fabrication, startup, and testing of a laboratory test facility, where integrated dual fluidized-bed testing and subsystem development were completed;
- Completion of integrated hybrid fluidized bed/raining bed testing and subsystem development in the laboratory test facility; and
- Completion of the design fabrication, and partial installation of a full-scale FBWHR system on an aluminum smelting furnace.

In conclusion, the FBWHR system fulfills a real need: waste-heat recovery from high-temperature, dirty, corrosive, furnace exhaust gases. This program developed the technology to build the system. However, the economic viability of recuperator systems is predicated upon expensive energy. With energy prices dropping, payback periods stretch, and recuperation systems are less attractive. This is true for any recuperation system, as well as for the FBWHR system. The Department of Energy, therefore, has reoriented its heat exchanger research and development program away from recuperation and toward higher technology applications. In the future, when energy prices increase to higher levels, FBWHR system technology will be available.

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2. INTRODUCTION

In many industries, stack gases from high-temperature process heaters are discharged directly to the atmosphere at high temperatures, wasting a significant fraction of the thermal energy input. In many cases, the stack gas temperature is 2000°F or higher, corresponding to a flue gas loss of 55 percent or greater. The energy loss in the flue gas in any direct heating process is shown in Figure 2.1 as a function of the flue gas temperature. In 1977, direct heating processes in the U.S. industrial sector used about 11 percent of the national energy consumption of all types, or about 8.4×10^{15} Btu/hr;¹ the major fraction of this energy was provided by combustion of natural gas and fuel oil. If an average savings of 15 percent were achieved by widespread application of heat recovery systems to industrial process furnaces, national yearly savings in premium fuel consumption would be 1.26×10^{15} Btu/hr, sufficient to heat about 18 million homes every year.

The most universally applicable use of the energy content of the flue gas is to preheat the combustion air with a heat recovery regenerator or recuperator, as illustrated in Figure 2.2. The heat recovered results in a greater than one-to-one reduction in the fuel input to the furnace for a given heating duty; i.e., 1 Btu recovered results in more than 1 Btu reduction in the fuel input. The percent savings in fuel consumption by use of the heat recovery unit is presented in Figure 2.3 as a function of the furnace exit exhaust gas temperature and the heat transfer effectiveness of the regenerator or recuperator. The percent energy savings by use of a 60-percent effective regenerator ranges from 13 percent at a furnace exit temperature of 1000°F to 48 percent at a furnace exit temperature of

Figure 2.1 Energy Loss in Flue Gas in Any Direct Heating Process

Figure 2.2 Schematic of Regenerator/Recuperator Heat Recovery for Industrial Process Heater

Figure 2.3 Percent Savings in Fuel Consumption as Function of ϵ_{OA} with Furnace Exit Temperature as Parameter (Natural Gas, 10% Excess Air)

2400°F. The savings would be greater for a higher effectiveness recuperator.

However, universally applicable recuperators for recovery of this waste heat are not available because most of these gas streams are at high temperatures and contain dirty or corrosive contaminants. These severe environments create materials problems for existing recuperator designs. Thus, the key to developing a universally applicable recuperator is serviceability in this environment. The second most important criterion is for the unit to be low in cost to ensure that the heat exchanger can be amortized over a reasonable period of time and, hence, will be widely purchased by energy users. These two criteria are more important than high effectiveness. To demonstrate this, consider, for example, a furnace with 2000°F exhaust gases. A moderate-effectiveness recuperator (50 percent) will save a little over 30 percent of the energy input. A high-effectiveness recuperator (90 percent) will increase the energy savings to 45 percent - a 50-percent increase. But, for a typical counterflow recuperator design, this effectiveness increase would require a 400-percent increase in heat transfer surface area. Hence, two-thirds of the benefit was purchased for only one fourth of the recuperator heat transfer area. Looking at a 1,000,000-Btu/hr unrecuperated furnace firing rate, a moderate-effectiveness recuperator reduces the firing rate to 680,000 Btu/hr, and a high-effectiveness recuperator reduces the firing rate to 550,000 Btu/hr. Therefore, the need is to get heat recovery equipment installed on the furnace and then strive to increase its effectiveness (by increasing the recuperator heat transfer area, for example).

The objective of this program is to demonstrate the combustion air preheat capability and long service life of the Fluidized-Bed Waste-Heat Recovery (FBWHR) system, while meeting the above criteria. This system will have very wide applicability and, thus, it has the greatest potential for maximum energy savings through recovery of

currently wasted heat. Of particular importance is the ability of the FBWHR system to operate with dirty and corrosive flue gases, a major impediment to industrial heat recovery in the past. The proposed unit will also have high reliability, an adequate service life, and be low in cost, resulting in a short amortization period.

Figures 2.4 and 2.5 show the two related FBWHR system arrangements examined during the course of this research. The systems consist of two beds, one mounted above the other. The exhaust gases from the furnace at 2000°F pass through the upper hot-stage fluidized-bed or raining bed, and heat the particles. These heated particles are then transferred to the lower cold stage-fluidized-bed where they are used to preheat the incoming combustion air. The cooled particles are then returned to the upper bed for reheating. The preferred configuration was found to be the hybrid fluidized-bed/raining bed FBWHR system.

Initial evaluation of this concept was for the secondary aluminum industry. The secondary aluminum industry melts scrap aluminum and alloys it into a specified alloy, which is sold as ingots or liquid metal. There are currently 76 secondary aluminum plants in the United States.² Output from these plants was 1,954,000 tons in 1983. This was 29 percent of net domestic aluminum shipments of 6,653,800 tons. There are two sources of aluminum scrap material, new scrap and old scrap. New scrap is waste material created in the manufacturing process. It is directly related to production, and over 90 percent is recycled. Old scrap is that material fabricated into finished products and discarded by the ultimate purchasers. Only a small fraction of old scrap is recycled, and over 2 million tons are lost yearly to municipal and commercial waste disposal sites.

The problem with using standard metallic recuperators on aluminum smelting furnaces is the very corrosive flue gas. The primary

Figure 2.4 Dual Fluidized-Bed FBWHR System Concept

Figure 2.5 Hybrid Fluidized-Bed/Raining-Bed FBWHR System Concept

contaminant in scrap aluminum is magnesium. To remove the magnesium, chlorine is bubbled through the melt, forming magnesium chloride, which is easily skimmed off. Unfortunately, some of the chlorine escapes into the flue gases and up the stack resulting in a severe corrosion problem with standard metallic recuperators. Thus, no satisfactory recuperator is available to recover the waste heat from the over 2000°F flue gases produced by these furnaces. Hence, the development of the FBWHR system, which is described in this final report.

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3. SITE CHARACTERIZATION

3.1 INTRODUCTION

Vulcan Materials Co. (VMC), a major smelter of secondary aluminum, was the owner of the site where the initial FBWHR system development field work was done. Initial tests were performed in their Sandusky, Ohio plant, on furnace No. 2. At the VMC Sandusky Operations, the aluminum scrap material was received by either truck or rail. Much of this material was crushed to a uniform size, and any paint, oil, or dirt was removed in a rotary drier. The aluminum scrap was then smelted in a batch process using an open hearth reverberatory furnace. Figure 3.1 is a schematic of such a furnace, showing the aluminum being loaded into the open hearth. The charge floated on a pool of liquid aluminum and was gradually melted. Melting this charge took 12 hours. During melting, the aluminum was alloyed, and any unwanted contaminants were removed. The aluminum was then drawn off and cast into 30-lb ingots, 1000-lb sows, or shipped in molten form in insulated ladles to the customers' plants.

Heat for melting the aluminum was provided by burning natural gas. Unpreheated air was provided to the burners by a blower, at a maximum pressure of 8 oz/in². Three burners fired over the melt, each with a capacity of 8 MMBtu/hr, for a maximum firing rate of 24 MMBtu/hr. The flue gases were exhausted from the furnace into an induced draft hood at the same end of the furnace as the burners. Exhaust temperatures ranged from 1400°F to 2200°F over the production cycle.

Burner firing rate was controlled using two thermocouples; one measured the flue gas temperature and one the melt temperature. The desired temperature was set by the operator. During the charging cycle, the flue gas temperature was set at 2100°F and the melt temperature at 1400°F. During standby both settings were reduced.

Figure 3.1 Aluminum Smelting Furnace

If both of the thermocouples called for heat, the burners fired at a rate proportional to the difference between the set and measured melt temperature. If this difference exceeded 30°F, the burners fired at maximum capacity. The flue gas thermocouple provided over-temperature protection. If the measured flue gas temperature exceeded the set temperature, the burners were turned off. The operator was free to change the settings and, hence, the firing rate at any time. It was not uncommon for the operator to shut down the burners during the charging cycle.

The furnace pressure was maintained by an air jet damper located at the exit of the flue. Flue pressures were 0.12 in. w.c. during charging and 0.04 in. w.c. during standby operations. The 0.12-in. w.c. value was higher than desired by VMC, and could have been the result of an undersized flue.

During the charging cycle, chlorine was injected into the open hearth to remove unwanted magnesium. The chlorine combined with the magnesium to form magnesium chloride, which floated to the top of the melt and was skimmed off. (The chlorinating system was new at the time and still in the development stage.) In addition, fluxing compounds were manually added to the melt to combine with trace metal impurities which were also skimmed off. The combination of chlorine and potash compounds could enter the furnace and exhaust out the flue. The FBWHR system components that were exposed to the flue gas had to withstand these compounds.

In order to match the recuperator to VMC's furnace, operational data such as air and fuel flow rates, flue gas temperature, and furnace turndown were required. For instance, the fluidized-bed distributor plate could be sized only after the flue gas flow rate was known. Further, to document the effect of a recuperator, it was necessary to record the furnace performance before and after the recuperator was installed. Unfortunately, the data needed for the design and baseline

energy consumption were not normally recorded by VMC. For that reason, a characterization of VMC's furnace was undertaken. The characterization was conducted by the following methods:

1. On-site inspections,
2. On-site small-scale experiments in the host furnace flue, and
3. Continuous data acquisition of the host furnace.

The following sections document the results of the furnace characterization.

3.2 FURNACE DATA ACQUISITION

An automatic data acquisition system was assembled to monitor the furnace operations. A photograph of the unit is shown in Figure 3.2. The system used a Hewlett-Packard Model 85A microcomputer and a Hewlett-Packard Model 3497A data acquisition module. The capabilities of this system included monitoring 40 channels of analog and 16 channels of digital data. The system was installed in Sandusky to monitor the operation of furnace No. 2. Specifically, air and fuel flow rates to each burner, flue gas temperature, and aluminum temperature were monitored continuously. These data were recorded on a magnetic tape for later analysis. Channel sampling frequency could be varied for each channel, limited only by tape capacity and the computer clock.

3.2.1 Furnace Operations

An overview of the furnace operation is shown in Figure 3.3. The burner air flow rate was plotted over a typical two-week period. The operation of the furnace was cyclic with wide variations in operating parameters. The flow rate varied from over 15,000 lb/hr during high fire to zero during low fire. (A zero shift in the instrumentation accounts for the finite reading at low fire in this figure.) Further, the flow rate often changed abruptly from high

Figure 3.2 The Automatic Data Acquisition Unit

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AIR FLOWRATE STARTING 5/5/82 ENDING 5/19/82

to low fire, indicating the burners were operating in an on-off mode. This type of cyclic on-off operation usually occurred for a week or more.

Smelting was sometimes suspended during the weekend. At about 72 hours, on Figure 3.3, a weekend was encountered. The burner flow rates were reduced; the burners fired only long enough to keep the remaining molten aluminum at about 1300°F.

The flue gas temperature corresponding to the two-week period of Figure 3.3 is shown in Figure 3.4. The maximum flue gas temperature during charging was about 2200°F. During the weekend holding period, the maximum flue gas temperature was 1350°F.

3.2.2 Furnace Flow Rate

Actual furnace operations are shown in Figure 3.5, an analogous plot to Figure 3.4. For the first 10 hours no scrap was being charged, and the burners mostly fired at a reduced rate. During scrap charging (10 hours and beyond on Figure 3.5) the burners fired almost continuously at high fire. Generally, the burner flow rate was constant during this period. The decline shown between 10 and 20 hours in Figure 3.5 was typical of the completion of a charging cycle. As the scrap remaining in the furnace was melted, the melt temperature rose. Since the flow rate was inversely proportional to the melt temperature, the flow rate declined.

Occasionally, a spiked decrease in the flow rate was encountered. This was usually due to manual override by the operator. For instance, it was necessary to periodically stir the melt in the furnace. To do this, the burners had to be manually turned off.

The fuel flow rate corresponding to Figure 3.5 is shown in Figure 3.6. The fuel flow rate and air flow rate were coupled by a ratio control system. Percent changes in the fuel flow rate were almost identical with percent changes in the air flow rate, resulting

Figure 3.4 Flue Gas Temperature for a Typical Two Week Period

Figure 3.5 Total Combustion Air Flow Rate

Figure 3.6 Total Fuel Flow Rate

in a more or less fixed air/fuel ratio as shown in Figure 3.7. An air/fuel ratio of 20 (by mass) corresponds to about 16 percent excess air.

The total flue gas flow rate was the sum of the combustion air flow rate, fuel flow rate, and furnace leakage. Neglecting leakage, the flue gas flow rate, based on the preceding plots, was about 4.8 lbm/sec during charging. The preceding data were gathered from burners that had recently been tuned. It was observed that over a period of time burner valve settings drifted, causing an increase in flow rate. Figures 3.8 and 3.9 show the air and fuel flow rates, respectively, for such conditions. The total flue gas flow rate was 5.4 lbm/sec, a 12-percent increase over the tuned burners.

3.2.3 Flue Gas Temperature

Figure 3.10 shows the flue gas temperature to be approximately constant at 2200°F during the charging cycle. During standby, when scrap charging was not occurring (0 to 10 hours of Figure 3.10), the measured flue gas temperature was below 500°F. However, the thermocouple measuring the flue gas temperature was located close to the air jet damper. When the burners were shut off, it was possible for the cold air from the damper to convectively cool the thermocouple, resulting in a temperature reading lower than the actual flue gas temperature.

The value of the flue gas temperature could not be correlated to burner flow rate alone. It was dependent on the difference between the heat entering the furnace from the burners (controlled by the operator and automatic controls) and the heat being absorbed by the scrap (controlled by the scrap feed rate). Wall losses could be ignored, being relatively constant. In Figure 3.11, after about 14 hours, the firing rate is decreasing in response to the completion of the scrap feed. However, the melt temperature was rising and hence

Figure 3.7 Air Fuel Ratio

Figure 3.8 Total Combustion Air Flow Rate for Untuned Burners

Figure 3.9 Total Fuel Flow Rate for Untuned Burners

Figure 3.10 Measured Flue Gas Temperature

Figure 3.11 Flue Gas Temperature Variation with Flow Rate

was absorbing less heat. Thus the flue gas temperature remained constant.

After completion of scrap melting, the operator frequently reduced the firing rate by reducing one or both of the temperature settings. Beyond 7 hours on Figure 3.12 the flue gas temperature setting was reduced to 1400°F and the melt temperature was set to some value above 1250°F. Now the burners fired only as necessary to satisfy the flue gas temperature setting.

The flow rate and temperature response to a step change in the valve setting can be observed in Figure 3.13. At about 0.225 hours, the air and fuel flow valves were closed by the controller. The air and fuel flow rates then simultaneously decreased in a step-wise fashion. At that point, the flue gas temperature began to decay. The proportional and step-wise controls described were typical of the VMC operation.

Also shown in Figure 3.13 are the fluctuations in the nominal air flow rate brought about by the control system. The fluctuations were minimal, about ± 95 lbm/hr or $\pm 1\frac{1}{2}$ percent.

3.3 FURNACE ENERGY USAGE ANALYSIS

An energy balance on the furnace, based on the data gathered by the data acquisition system, was calculated. The energy input to the furnace was known from the measured air and fuel flow rates. Three additional assumptions were used:

1. Stack losses. The flue gas flow rate was assumed to be equal to the sum of the air and fuel flow rates. Leakage was ignored. The flue gas temperature was measured; hence, the flue gas energy could be calculated.
2. Energy used to melt the aluminum scrap. This was composed of two parts: the sensible heat needed to raise the scrap

Figure 3.12 Flue Gas and Melt Temperature Variation with Flow Rate

Figure 3.13 Effect of Changes in Valve Setting on Flue Gas Temperature and Air and Fuel Flow Rates

from room temperature to melting temperature, and the latent heat of fusion of the scrap. Since the melt temperature and scrap feed were known, both components could be determined.

3. Wall losses. This was taken as the difference between the energy input and the sum of 1 and 2 above.

Figures 3.14 through 3.17 show the energy balance during the charging cycle for four typical days. Of the total energy input of 14 to 18 MMBtu/hr only 2.3 to 3.3 MMBtu/hr were needed to melt the scrap, for an overall efficiency of 13 to 23 percent. The efficiency was somewhat lower than the design value. This was possibly due to the difficulties VMC experienced with their chlorinating system. In order to complete the chlorinating process it was necessary at times to keep the burners at high fire longer than was necessary to melt the scrap.

Of the balance of the heat, 7.4 to 10.2 MMBtu/hr were exhausted in the flue gases. Thus, about 55 percent of the total heat input was lost in the exhaust gases.

3.4 FURNACE TURNDOWN

Furnace turndown is defined as the ratio of maximum to minimum furnace flow rates encountered during normal operation. It was necessary to know the furnace turndown and the energy contained in the turndown in order to determine the required turndown of the recuperator. The first step was to determine the amount of time the furnace spent at each flow rate; this is shown in Figure 3.18. The combustion air flow rate was plotted as a function of the percent of time spent at that flow rate. The data were averaged for two 19-day periods; one before the burners were tuned (solid line) and one after (dashed line). For each condition, about 40 percent of the time was spent at high fire. At a flow turndown of 2:1 (one half of the maximum flow rate) the cumulative amount of time spent was 55 percent for the tuned case and 50 percent for the untuned case.

3/29/82

BASELINE PERFORMANCE

35

WITH FBWHR SYSTEM

Figure 3.14 Heat Balance on Furnace #OH2 at Maximum Firing Rate
March 29, 1982

3/30/82

BASELINE PERFORMANCE **WITH FBWHR SYSTEM**

Figure 3.15 Heat Balance on Furnace #OH2 at Maximum Firing Rate
March 30, 1982

5/11/82

BASELINE PERFORMANCE

WITH FBWHR SYSTEM

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Figure 3.16 Heat Balance on Furnace #OH2 at Maximum Firing Rate
May 11, 1982

5/12/82

BASELINE PERFORMANCE

WITH FBWHR SYSTEM

Figure 3.17 Heat Balance on Furnace #OH2 at Maximum Firing Rate
May 12, 1982

Figure 3.18 Time Spent at Turndown

The flue gas energy content during turndown was of considerable importance since it determined the economy of the system. Flue gas energy content could be calculated from the flow rates of Figure 3.18 and from the measured flue gas temperatures. The results are shown in Figure 3.19. About 80 percent of the available flue gas energy was contained in the high fire flow rate. Well over 90 percent of the energy was represented at a turndown of 2:1. The recuperator, then, had to be capable of operating at half the design flow rate. Operation below this level would produce only a marginal increase in energy savings.

3.5 FINAL INSTALLATION SITE CHARACTERIZATION

3.5.1 Introduction

Vulcan Materials Company (VMC) shut down their aluminum smelting operation at Sandusky, Ohio, shortly after the initial FBWHR system site characterization and materials testing (see below) was completed. Due to the promising early results of the program, VMC expressed continued interest in the FBWHR System development program, and offered an alternate site for the full-scale demonstration. Subsequently, an Oak Creek, Wisconsin, VMC plant became the new FBWHR system installation site.

At the Oak Creek site, the aluminum scrap material is received either by truck or by rail. Much of this material is crushed to a uniform size, and any paint, oil, or dirt is removed in a rotary drier. The aluminum scrap is then smelted in a batch process using an open hearth reverberatory furnace. Figure 3.20 shows the plant layout and Figure 3.21 is a schematic of the furnace. Drawings of a typical furnace are included as Figures 3.22 and 3.23. The aluminum charge is loaded into the furnace, where it floats on a pool of liquid aluminum and is melted. During melting, the aluminum is alloyed, and any

Figure 3.19 Energy Turndown

Figure 3.20 Vulcan Materials Company, Oak Creek, Wisconsin Site Plan

Figure 3.21 Open Hearth Aluminum Reverberatory Furnace

Figure 3.22 VMC Furnace Plan View

Figure 3.23 VMC Furnace Elevation Views

unwanted contaminants are removed. The aluminum is then drawn off and cast into 30-lb ingots, 1000-lb sows, or shipped in molten form in insulated ladles to the customer's plants.

Heat for melting the aluminum is provided by burning natural gas or fuel oil. Unpreheated air is provided to the burners by a blower, at a maximum pressure of 8 oz/in². Two burners fire over the melt, each with a capacity of 10.5 MMBtu/hr, for a maximum firing rate of 21 MMBtu/hr. The flue gases are exhausted from the furnace into an induced draft hood at the same end of the furnace as the burners. Exhaust temperatures range from 1400°F to 2000°F over the production cycle.

Burner firing rate is maintained using two thermocouples and controllers; one measuring the flue gas temperature and one the melt temperature. The desired set point temperatures are set by the operator. During the charging cycle, the flue temperature controller is set at 2000°F and the melt temperature controller at 1500°F; during standby one of the settings is reduced. If both of the controllers call for heat, the burners will fire at their preset maximum set point. If either the measured flue gas temperature or the melt temperature exceeds its set point, the burners are turned to a low fire setting. The operator is free to change the settings and, hence, the firing rate at any time. Typical firing rates range between 16 MMBtu/hr at high fire and 3 MMBtu/hr at low fire.

The furnace pressure is maintained by an air damper located at the exit of the flue. Flue pressure is 0.05 in. w.c. above atmospheric pressure.

During the charging cycle, chlorine is continuously injected into the molten aluminum with the aid of a molten metal pump to remove unwanted magnesium. The chlorine combines with the magnesium to form magnesium chloride, which floats to the top of the melt and is

skimmed off. In addition, fluxing salts are manually added to the melt to combine with trace metal impurities, which float to the surface and get skimmed off. The combination of chlorine and fluxing salts can enter the furnace and exhaust out the flue. The FBWHR system components that are exposed to the flue gas must withstand these corrosive constituents.

3.5.2 Final Furnace Design Specifications

In order to match the recuperator to VMC's host furnace, operational data such as air and fuel flow rates, flue gas temperature, and furnace turndown were required. For instance, the cold-stage fluidized bed's distributor plate could be sized only after the combustion air flow rate was known. For this reason, a characterization of VMC's furnace at Oak Creek, Wisconsin was undertaken.

Determining the pre-retrofit furnace design values was accomplished by several methods. On-site inspections of actual furnace operation, and examination of actual fuel usage and furnace production values revealed furnace flow rates and energy usage.

Table 3.1 enumerates the outcome of the site characterization of furnace No. 1 at the VMC Oak Creek, Wisconsin plant. Under furnace high-fire conditions, the burner firing rate was 16 MMBtu/hr, with an unpreheated combustion air mass flow rate of 13,520 lb/hr. The resultant flue gas flow rate was 14,190 lb/hr. Design low-fire conditions included a burner firing rate of 3 MMBtu/hr, and a combustion air flow rate of 2535 lb/hr. Flue gas mass flow rate under these conditions was 2661 lb/hr. Figure 3.24 shows the number of hours at these firing rates for a typical cycle.

These design mass flow rates and firing rates, coupled with the previously described thermal history of an aluminum smelting furnace, resulted in the pre-retrofit furnace flow sheet pictured in Figure 3.25.

TABLE 3.1
HOST SITE CHARACTERIZATION

HIGH FIRE CONDITIONS:

Burner Firing Rate	16 MMBtu/hr
Air Flow Rate	13,520 lb/hr
Flue Gas Flow Rate	14,190 lb/hr

LOW FIRE CONDITIONS:

Burner Firing Rate	3 MMBtu/hr
Air Flow Rate	2535 lb/hr
Flue Gas Flow Rate	2661 lb/hr

Figure 3.24 Furnace Operation at VMC Oak Creek Facility

Figure 3.25 Pre-Retrofit Furnace Flow Sheet
(16 MMBtu/hr at Full Fire)

(In this report, the thermal capacity of a material is simply the mass flow rate of a material multiplied by its heat capacity.) The 970 Btu/hr-°F fuel input to the furnace is combined with 3264 Btu/hr-°F of 120°F combustion air, providing heat to the furnace. Waste gas from the furnace leaves at 2000°F, and has a thermal capacity of 4234 Btu/hr-°F.

Work detailed at the Vulcan Material Company Sandusky Plant showed that the majority of an aluminum smelting furnace's energy could be recuperated by a waste-heat recovery system possessing only 2:1 turndown. Figure 3.26 summarizes those results, and indicates the percent of a furnace's total energy usage during normal operations. Over 90 percent of the potential energy savings on a furnace could be obtained during only a 2:1 turndown on the furnace.

In conclusion, the unrecuperated furnace was characterized at both high fire and low fire, indicating the maximum and minimum system design conditions. Furthermore, a practical FBWHR system turndown condition of 2:1 was indicated.

Figure 3.26 Energy Use at VMC Oak Creek Facility

4. MATERIALS TESTING

4.1 MATERIALS SELECTION AND SCREENING

The materials requirements for critical raining bed components are that it:

- Be resistant to the corrosive agents in the flue gases;
- Retain strength at high temperature;
- Be easily fabricated;
- Be readily available in a range of sizes;
- Be reasonable in cost; and
- Be maintenance free for periods of one year or more.

Bare metals are often not acceptable because of a lack of corrosion resistance. Alumina, an excellent corrosion-resistant material, may not be acceptable because of its poor thermal shock resistance. Silicon carbide may not be acceptable because of its high cost and fabrication risk. Other ceramics may be unacceptable because of their poor thermal shock resistance, poor corrosion resistance, or poor fabricability. Alloy-coating systems offer improved corrosion resistance, and were tested for potential use in hot stage bed components. Table 4.1 is a list of these metals, ceramics, and coatings.

4.2 PRELIMINARY MATERIALS TESTING

Preliminary exposure testing of alternate materials was conducted in the furnace flue at VMC-Sandusky. The materials exposed were silicon carbide and pure alumina. Figure 4.1 is a photograph of the materials holder with samples in place.

The samples were examined after one week of exposure. One of the alumina samples had cracked in half, which was probably due to thermal shock. The densities of all the samples were unchanged indicating no major chemical reactions had occurred. However, longer

TABLE 4.1
COUPON TESTING PLAN

- Alloys
 - Haynes Alloy 188 (Improved Corrosion Resistance, Higher Strength)
 - Inconel 601 (Improved Corrosion Resistance)
 - Inconel 625 (Improved Corrosion Resistance)
 - Cabot Alloy 214 (Self-Coating)
 - 316 Stainless Steel (Low Cost)
- Coating Manufacturers for Alloys
 - Alon Processing, Inc.
 - Aremco Products, Inc.
 - Coatings Technology Co.
- Ceramics
 - Silicon Carbide
 - Alumina
 - Mullite

Figure 4.1 Materials Holder

exposure times would be needed for meaningful results. Additional tests were planned in conjunction with Oak Ridge National Laboratories including a wider variety of samples.

4.3 ALLOY AND CERAMIC COUPON TESTING

4.3.1 Materials Temperature Measurements

The alloys listed in Table 4.1 were immersed in VMC's No. 2 furnace flue in Sandusky for exposure testing, using a test holder shown in Figure 4.2. Coupon temperature measurements were taken during the testing with thermocouples welded to each coupon.

Because corrosion is a strong function of temperature, it is necessary to maintain the coupon temperatures at about the same temperature as the fluidized-bed distributor plate. Higher materials temperatures greatly accelerate the corrosion process. In a well-insulated unit, the distributor plate may be 100° to 200°F lower than the inlet flue gases. Figure 4.3 shows the measured coupon temperature (solid line) and the flue gas temperature (dashed line) as a function of time. The coupon temperature was about 50°F lower than the flue gas temperature, or about 50° to 150°F higher than the distributor plate temperature. Hence, the coupon tests would be expected to yield slightly conservative corrosion rates.

4.3.2 Materials Exposure Test Results

Samples of the exposed materials were examined by cutting off small pieces of the materials and taking photomicrographs of the cut pieces. Photomicrographs of the aluminum-diffusion-coated Inconel 625, exposed for 911 hours, are shown in Figure 4.4. The top figure shows that approximately 50 percent of the coating depth was damaged; the middle figure shows that approximately 30 percent of the coating depth was damaged; and the lower figure shows that approximately 20 percent was damaged. No damage to the substrate material occurred. For comparison, a photomicrograph of an unexposed sample is shown in Figure 4.5.

Figure 4.2 Material Coupon Test Holder

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Figure 4.3 Material Sample and Flue Gas Temperature

Worst: Approximately 50% of Coating's Life Used

Average: Approximately 30% of Coating's Life Used

Best: Approximately 20% of Coating's Life Used

Figure 4.4 Photomicrographs of Exposed Coated Inconel

Figure 4.5 Photomicrograph of Unexposed Coated Inconel

The coated Inconel sample, along with a simultaneously exposed uncoated Inconel sample, was again inspected after 2250 hours of exposure in VMC's flue. A typical photomicrograph of the coated sample is shown in Figure 4.6. The coating continued to be protective, although a small amount of corrosion (about 0.0065-inch deep) had occurred on the substrate. Assuming that the thickness of hot stage bed materials should be 0.125 inch, the damage represented a 5 percent reduction in thickness.

The coated Inconel sample could be compared to the uncoated Inconel sample. A photomicrograph is shown in Figure 4.7. The corrosion depth on the uncoated sample was about 0.03 inch, an increase in damage over the coated sample by a factor of 4.6.

Figure 4.8 shows a photomicrograph of coated Inconel 601 exposed for 3230 hours. Attack occurred on one side of the sample only. The damage progressed through about one half of the thickness or about 0.018 inch. The precise reason for the damage on one side of the sample and not the other side is unknown. Other photomicrographs of the different areas of the same material specimen showed different amounts of grain boundary attack. In some instances, the attack went through the entire thickness of the material. It is noted that the coating may have been more protective in some areas than in others.

The damage to the sample may not have been due solely to high temperature corrosion. In addition to the 3230 hours of exposure during furnace operation, this sample was also in the flue for 100 hours when the furnace was not operating. During this period, the sample was at ambient temperatures, which could have led to condensation of water vapor onto the sample surface. This water may have contained hydrochloric acid or other corrosive constituents that have damaged the sample, increasing the corrosion rate.

Figure 4.6 Coated Inconel 625 Exposed for 2250 Hours

795-N

Figure 4.7 Uncoated Inconel 625 Exposed for 2250 Hours

794-N

Figure 4.8 Coated Inconel 625 Exposed for 3230 Hours

A photomicrograph of the coated Inconel sample after 4410 hours of exposure is shown in Figure 4.9. In this specimen, damage occurred equally on each side ranging from a depth of 0.012 inch to 0.018 inch. A sample of coated Inconel 625 exposed for 8030 hours of furnace operation had damage throughout its thickness, or for 0.018 inches on each side.

A summary of the corrosion depth of the Inconel 625 (with the Coatings Technology, Inc., coating) that occurred to the samples retrieved at 911, 2250, 3230, 4410, and 8030 hours is shown in Figure 4.10. For the first 911 hours, no corrosion to the substrate material occurred. Between 911 and 2250 hours the Inconel was damaged through 0.007 inch of its thickness. This damage continued as the test progressed. At 3230 hours the damage was between 0.002 and 0.018 inch (the entire sample thickness). At 4410 hours photomicrographs show the damage to be between 0.012 and 0.018 inch. At 8030 hours photomicrographs show the damage to be 0.018 inch, the entire sample thickness. The damage increased with time, with a considerable amount of scatter in the data beyond 3230 hours.

For comparison, the corrosion rate of uncoated Inconel 625 is also shown in Figure 4.10. At 2250 hours the uncoated Inconel's corrosion was about two to 4-1/2 times greater than that of the coated Inconel. At 3230 hours the corrosion of the uncoated Inconel was about two to 20 times greater than the coated Inconel. Surprisingly, at 8030 hours the uncoated Inconel's corrosion was 1/3 to 1/4 of the attack at either 2250 or 3230 hours.

From the foregoing results, it was possible to specify the distributor plate's original thickness needed to allow a plate life of one year using the uncoated Inconel. The following assumptions were made:

- Grain boundary attack reduced the plate's effective thickness by the depth of the attack. Hence, grain boundary damage

Figure 4.9 Coated Inconel 625 Exposed for 4000 Hours

Figure 4.10 Corrosion History of Inconel 625 Exposed in Flue (/D=Aluminum Coating by Coatings Technology)

to 0.02 inch of the plate would reduce the effective thickness by 0.04 inch.

- The minimum effective thickness required to support the plate stress was 1/8 inch.
- Grain boundary damage was linear with time. From the preceding test results, a conservative estimate for this damage was 0.11 inch/year.

Based on the above, the distributor plate's original thickness had to be about 0.35 inch so that, at the end of one year, the remaining effective thickness would be greater than 1/8 inch.

It should be noted that the above estimate is conservative:

- Coupon corrosion rates were accelerated because they were tested at temperatures higher than design. Figure 4.3 shows the coupon sample temperature to be 50°F lower than the flue gas temperature, while the design value is 100° to 200°F below the flue gas temperature.
- The optimum alloy-coating system has not yet been identified. An improvement in the corrosion resistance of either the alloy or coating was possible, which would lead to a prediction of improved distributor plate life.

4.3.3 Additional Material Exposure Test Results

Additional metallic and ceramic materials coupons were retrieved from the VMC-Sandusky flue, as listed in Table 4.2. These materials included an additional coated Inconel 625, coated Inconel 601, coated Haynes 188, Alloy 214, sintered-alpha SiC, sintered (NC-430) SiC, siliconized (Refel) SiC, alumina, and mullite. Additionally, exposure results for the coated distributor plates from the 1/10-scale fouling unit (see Section 6.6) have been included.

TABLE 4.2
MATERIALS AND EXPOSURE PERIODS IN
ALUMINUM REMELT FURNACE FLUE GASES

Material	Coating and Coating Supplier	Exposure Period (h)
Alloys		
Inconel 625	---	2250, 3230, 8030
Inconel 625/A	/A = Silica by Aremco	3110
Inconel 625/D	/D = Aluminum by Coatings Technology	900,* 911, 1000,* 2250, 3230, 4410, 8030
Inconel 601/B	/B = Aluminum by Alon	1200
Haynes 188/B	/B = Aluminum by Alon	1200, 4030
Alloy 214	---	1200, 4030
Ceramic Materials		
CVD SiC	---	8000
Sintered SiC	---	4300
NC-430 SiC	---	4300
Refel SiC	---	4300
Alumina	---	4300
Mullite	---	4300

* = Coated distributor plate, not coupon sample

Measured corrosion rates of these materials are given in Table 4.3. Projected annual corrosion rates range from a low of 0 inch/year for alumina exposed for 4300 hours, and CVD silicon carbide exposed for 8000 hours, to a maximum projected annual corrosion rate of 0.33 inch/year for an Inconel 625/D distributor plate exposed for 1000 hours. A graphical summary of the corrosion data is presented in Figure 4.11.

Photomicrographics of the additional metallic samples showed scale formation and intergranular attack comparable to that discussed above for the uncoated Inconel 625 and Inconel 625/D.

A photomicrograph of the CVD silicon carbide, the best of the silicon carbides, is shown in Figure 4.12. No corrosion can be observed. A thin slag layer is, however, present on the surface. The alumina also showed no corrosion damage; however, significant cracking was present. This was expected because of the historically poor thermal shock properties of alumina. The mullite exhibited corrosion behavior that was intermediate between the CVD silicon carbide and the less pure silicon carbides.

4.4 MATERIAL EXPOSURE TEST CONCLUSIONS

The following conclusions can be drawn from the preceding test results:

- Coated alloys are viable materials for use as distributor plates;
- Coated alloys exhibited variable results, indicating that the coating process and/or coating/substrate combination are not yet optimized: additional testing to establish the coating reliability is justified;
- Aluminum-diffusion-coated Inconel 625 must be 0.35-inch thick for a one-year life, assuming operation of the alloy is at a full, 2000°F temperature;
- CVD silicon carbide furnace exhaust corrosion resistance is unexcelled; however, it is not commercially available; and

TABLE 4.3
CORROSION RATES OF MATERIALS EXPOSED TO
ALUMINUM REMELT FURNACE FLUE GASES^d

Material	Total Exposure Time (h)	Minimum-to-Maximum Depth of attack ^a (in.)	Projected Corrosion Rate (in./yr)
Alloys			
Inconel 625	2250	0.030	0.12
	3230	0.041	0.11
	8030	0.010	0.01
	911	0.001	0.010
Inconel 625/D	2250	0.007	0.03
	3230	0.002-0.018 ^b	0.005-0.05
	4410	0.012-0.018 ^b	0.02 ^b
	8030	0.018 ^b	b
	900*	0.030	0.29
	1000*	0.038	0.33
Inconel 625/A	3110	0.022	0.06
Inconel 601/B	1200	0.028	0.20
Haynes 188/B	1200	0.008	0.06
	4030	0.016	0.04
Alloy 214	1200	0.040	0.29
	4030	0.050	0.11
Ceramic Materials			
CVD SiC	8000	0	0
Sintered- α SiC	4300	0.012	0.02
NC-430 SiC	4300	0.010	0.02
Refel SiC	4300	0.010	0.02
Alumina	4300	0	0
Mullite	4300	0.003	0.01

^aDepth of grain boundary attack in alloys.
Depth of reaction zone in ceramic materials.

^bDepth of grain boundary attack equal to one-half specimen thickness.

^cNot measured.

^dTable 4.2 for Material Coating Identification

*Coated distributor plate, not coupon sample.

Figure 4.11 Corrosion History of Materials Exposed in Flue

Figure 4.12 Silicon Carbide Exposed to Host Furnace Flue Gases
~ 8000 Hours

- Alumina's susceptibility to thermal shock cracking eliminates it as a distributor plate material.

5. FBWHR SYSTEM CONCEPTUAL DESIGN

This section describes the conceptual design of the FBWHR system for the Sandusky, Ohio, installation site. The operating conditions necessary for successful furnace operation were determined and limitations established in Chapter 3. In this chapter, the system sizing and media selection are presented as well as the concept for off-design operation.

5.1 SYSTEM CAPACITY

The FBWHR system flow rate and temperature specifications could be accurately predicted based on the measured furnace data by using proven analytical procedures. The analytical procedures determined the effect the recuperator would have on the furnace. For instance, the effect of preheating the combustion air was to reduce the burner fuel and air requirements needed to produce a given furnace heat flux input. The analytical procedures were conducted by first constructing an energy balance around the furnace-recuperator combination and, second, by knowing the FBWHR system characteristics. The results, in terms of the furnace total mass and volume flow rate change, are given in functional form as:

$$\frac{\Delta \dot{m}}{\dot{m}} = 1 - \frac{f(T_A)}{f(T_{A'})}$$

$$\frac{\Delta \dot{v}}{\dot{v}} = \frac{f(T_A)}{f(T_{A'})} \frac{T_{A'}}{T_A} - 1$$

where:

$\frac{\Delta \dot{m}}{\dot{m}}, \frac{\Delta \dot{v}}{\dot{v}}$ = change in burner mass and volume flow rate, respectively

T_A = preheat combustion air temperature

$T_{A'}$ = unpreheated combustion air temperature

The results are given in Figure 5.1. As the preheated air temperature was increased, the total burner air mass flow rate and, hence, fuel mass flow rate was reduced. At the design preheat temperature of 1000°F, a 30-percent reduction in fuel mass flow rate occurred. This resulted in a design flow rate of 3.4 lbm/sec. Despite the reduction in mass flow rate, the volume flow rate increased about 84 percent. In order for the FBWHR system to pass this flow and remain within the fluidized-bed design criteria, a bed cross-sectional area of 20 ft² was required.

The heat balance for the VMC-Sandusky aluminum furnace with the recuperator was shown in Figures 3.14 through 3.17. At the maximum firing rate, the FBWHR system recovered 2.4 MMBtu/hr from the exhaust gases and used it to preheat the combustion air. This reduced the required fuel input from 18 to 12.6 MMBtu/hr, a 30-percent reduction. The specific fuel consumption for the unrecuperated furnace was between 2200 and 4000 Btu/lbm. The recuperator reduced this to 1550 to 2800 Btu/lbm.

The recuperator flow rates and temperatures at the design point of a dual-fluidized bed FBWHR system are shown in Figure 5.2. The combustion air was preheated from 100°F to 1000°F at a flow rate of 146,000 scfh. The heat provided to this combustion air was taken from the exhaust gases where the temperature was reduced from 2000°F to 1250°F at a flow rate of 160,000 scfh.

The amount of flue gas exiting the furnace can increase over the design value due either to air leakage into the furnace or off-design burner flow rates. Off-design flow rates increases as great as 15 percent have been observed. Generally, the FBWHR system was not greatly affected by these off-design flows. However, it was possible for these flows to periodically increase to higher values or for air leakage into the furnace to occur so that the flue gas flow rate could become substantially higher than the design value. Further, the amount of this excess flow would vary with furnace operation and could not be

Figure 5.1 Effect of Recuperator on Furnace Flow Rate

Figure 5.2 Recuperator Flow Rates and Temperatures

accurately predicted. To accommodate this increased flow, a flue gas bypass was provided on the FBWHR system, as shown in Figure 5.2, that automatically diverted the excess flow. Additionally, a flue gas bypass allows access to the system while the furnace is running.

5.2 RECUPERATOR TURNDOWN

Recuperator turndown was defined as the ratio of the highest to the lowest flue gas flow rate that could be passed through the recuperator without adversely affecting its performance. Turndown was important because it determined the amount of available flue gas energy that could be recovered. If the recuperator turndown was greater than or equal to the furnace turndown, all of the available flue gas energy would be recovered. However, since most of the flue gas energy was contained in the high-fire part of the cycle, it was possible for the recuperator turndown to be less than the furnace turndown and still capture a substantial part of the available energy. As shown in Section 3.4, a turndown of 2:1 captured over 95 percent of the available flue gas energy.

In the FBWHR system, off-design operation could be handled by one of two methods. As the overall flow rate was decreased from the maximum, the fluidizing velocity could be allowed to decrease through a constant bed area, or the bed area could be decreased to maintain a constant fluidizing velocity.

5.2.1 Fluidizing Velocity Turndown

In this fixed geometry concept, the percent open area of the distributor plates was such that the minimum pressure loss requirement across the distributor plate was satisfied at the required minimum flue gas flow rate.

The major advantage of this concept was that it could supply adequate turndown capability; yet, it was inexpensive to fabricate, easy to operate, and required little maintenance compared to other concepts.

The major problem in using the fluidizing velocity to control turn-down was the high-pressure loss needed at maximum firing conditions, even when assuming a minimal velocity at low fire. The lowest velocity possible to maintain fluidization was the minimum fluidization velocity, which occurred when the drag force on the particles, caused by the gas passing upwards through the bed, was about equal to the weight of the particles.

Although it was theoretically possible to operate the FBWHR system at the minimum fluidization velocity, practical considerations required that the superficial velocity be appreciably increased above this minimum. The primary consideration was to impart sufficient mobility to the particles to assure a smooth flow of solids away from the feed zone and along the bed. Laboratory tests simulating the FBWHR system operating conditions indicated that a velocity of three times the minimum fluidization velocity was required.

Using the criteria just explained, the minimum superficial velocity for a given temperature could be established and, therefore, the maximum superficial velocity that was acceptable could be calculated. If, for a given temperature and particle size, the minimum superficial velocity was 3 ft/sec and a turndown of 5 was necessary, the maximum superficial velocity would be 15 ft/sec.

Because the minimum pressure loss established the basis for the design, there was a great deal of emphasis on this area in laboratory testing. From this testing, detailed below, it was determined that the plate pressure loss had to be at least one half of the bed pressure loss to maintain stability. For a 2-inch bed of the media selected, a distributor plate pressure difference of 2.0 in. w.c. was needed at low load conditions. Below this point, the bed would destabilize and results could not be predicted.

A 2-inch bed depth was selected for a production unit. Although the bed height could be decreased to reduce the overall pressure loss, there was a point where the bed height itself set the stability criteria. Laboratory tests suggested that operating at less than 2-inch bed depth in a production unit could be possible.

The major limitation to this concept was the relatively high pressure loss required at high fire. Since pressure loss was proportional to the square of the flow rate, a turndown of 5:1 would require a plate pressure loss, at high-fire, 25 times greater than the minimum required. The total pressure loss (plate plus media) at high-fire is plotted against turndown in Figure 5.3. A turndown of 2:1 required a pressure loss of about $9\frac{1}{2}$ -in. w.c.; anything above that would be prohibitive.

5.2.2 Bed Area Turndown

The turndown of the system could be increased without a further increase in the system pressure loss by varying the bed area. This could be accomplished by separating the fluidized bed into two or more modules. As the flow rate of gases decreased, one or more modules would be shut off with the gases all going through the remaining modules, thereby reducing the flow area. It was, therefore, possible to operate at a high turndown with only a small variation (factor of two) in the superficial velocity.

If the temperature of the flue gases was constant at both high fire and low fire, the turndown of the system would vary directly with the area ratio. However, as is the case with many furnaces, the VMC furnace had a reduced flue gas temperature at low-fire. In fact, the flue gas temperature dropped over 600°F. The turndown capability was reduced due to the temperature effects on the pressure loss and the minimum superficial velocity. The net effect of the number of equally sized modules versus turndown, corrected for temperature, is shown in Figure 5.4. For example, an area ratio of 5 to 1 would yield

Figure 5.3 Effect of Turndown on Total Pressure Loss

Figure 5.4 Turndown with Variable Bed Area

a turndown of about 3 to 1; whereas, without temperature correction, the turndown expected would be 5 to 1. If module sizes were chosen by a geometric progression, as shown in Figure 5.5, it would be possible to achieve the same turndown with fewer modules. This is also illustrated in Figure 5.4.

There were a number of problems with using the modular concept. The complexity of the system was such that further development would have been needed to ensure that the maintenance level and costs were acceptable. Also, the gases in the nonoperating modules were cooled, allowing possibly corrosive agents to condense, which could have resulted in serious materials problems.

5.2.3 Conclusions

Turndown for the FBWHR system was effected by using the fluidized velocity option with a single bed. As mentioned in an earlier section, 95 percent of the potentially recoverable energy could be saved by a turndown of 2 to 1. This level of turndown could be reached with the fluidizing velocity variation.

The modular bed concept was not used in this program because its higher turndown capability was not needed. Also, the effort to develop the concept would have been considerable because of corrosion problems and design complexity.

5.3 SYSTEM MEDIA SELECTION

Proper selection of the circulating heat transfer media for the FBWHR system was critical to assure optimum performance and minimum maintenance. Since the media were continually circulated from contact with the hot flue gases to the cool combustion air bed, high thermal shock resistance and low attrition of particles were of paramount importance. Equally important was the ability of the material to withstand the corrosive environment of the flue gas. Table 5.1 delineates the

	<u>No. OF BEDS</u>	<u>AREA RATIO</u>
2	2:1
3	4:1
4	8:1
5	16:1

Figure 5.5 Geometrically Varying Bed Area

TABLE 5.1
MEDIA SELECTION CRITERIA

High Corrosion Resistance
Compatibility with Process Furnace Exhaust
High Thermal Shock Resistance
Minimal Attrition Rate
Moderate Cost
Availability in Uniform Sizes
High Thermal Capacity

properties required of the heat transfer media. Although no material was ideal, the best material was alumina. Alumina is a hard ceramic that is readily available in a large variety of sizes, shapes, and purities.

The size of particles used in the FBWHR system influenced both the size of the recuperator and the pressure loss requirements, which in turn affected the cost of building and operating the system. From the heat exchanger characterization studies, a bed depth of approximately 30 particle diameters was needed for the fluidizing gas to reach thermal equilibrium with the particles. This, combined with the stability criteria, showed that the distributor plate total pressure loss of the fluidized bed increased linearly with the particle diameter, as shown in Figure 5.6, resulting in an increasing operating cost. However, a larger particle size required a higher fluidizing velocity which, with a constant mass flow rate of flue gas, resulted in a smaller distributor plate area, as shown in Figure 5.7. This resulted in a decreasing fabrication cost.

The optimum particle size could be found by noting that the slope of the pressure curve was constant, while that of the area curve was not. Increasing the particle diameter yielded a diminishing decrement in area yet required a constantly increasing pressure drop. For this reason, it was not useful to use a particle size greater than 800 μm .

A lower bound on the particle size was established based not only on the large recuperator size required but also on particle weepage through the distributor plate holes. To determine the minimum particle size, tests were performed with 660- μm particles and perforated plates with various hole sizes ranging from 1/16 to 1/8 in. in diameter. The tests indicated that while the bed was fluidized, particle weepage was insignificant. However, when the bed was stagnant, a hole-size to particle-diameter ratio greater than 2.5 caused an unacceptably high weepage. Since hole size was limited to a minimum of 0.062 inch

Figure 5.6 FBWHR System Size and Pressure Drop Dependence

Figure 5.7 Distributor Plate Area vs Particle Diameter

because of fabrication and fouling considerations, the smallest particle size that could be used was 630 μm . Thus, the media size was limited to the range from 630 to 800 μm .

In the laboratory studies, nonspherical alumina of 660 μm diameter and 95-percent purity was used. This selection was made essentially on the basis of economics. This low-grade material was readily available as sand blasting grain. There was some evidence that this material would not perform over a long period of time because of the impurities, cracks, and porosity within the particles. To assess the potential of this problem, photomicrographs of some alumina particles were taken by Oak Ridge National Laboratories. Figure 5.8 is a typical cross section of the particles showing the various defects mentioned. If these alumina particles broke down in the recuperator, high-purity spherical alumina pellets such as those illustrated in Figure 5.9 could be used. The spherical alumina had fewer defects and, therefore, a diminished attrition potential.

Figure 5.8 Polished Section of Alumina Particles Used in FBWHR System

Figure 5.9 Polished Sections of Spherical Alumina Particles from Norton Company

6. COMPONENT CHARACTERIZATION

Design of any fluidized-bed system requires experimental verification of the correlations being used because the extremely complex nature of the bed behavior limits the accuracy of existing correlations. The analytical design tools needed to characterize the bed were developed and experimentally verified. These design tools were developed from the extensive fluidized-bed technology base available at Tecogen. This chapter describes the test facility, the fluidization characteristics of the heat transfer media, the heat transfer characteristics of the fluidized bed, lift line test results, particle feed system test results, and field test results of a 1/10-scale fluidized bed test unit.

6.1 EXPERIMENTAL EQUIPMENT

The laboratory test unit used in these studies is shown schematically in Figure 6.1. Figure 6.2 shows the actual installation. The heat exchanger was 1-ft wide x 4-ft long, with a total height of 10 ft.

Vitiated air (products of combustion with high excess air levels) was used as the fluidizing medium. (The flow circuit is shown in Figure 6.3.) Air was drawn from the room by a 1200-scfm forced draft fan and metered through an orifice plate. After passing through the orifice, the flow was split into two streams - one supplying air to the gas burner and the other bypassing directly into the plenum. Because the burner had a turndown ratio of 50:1, this bypass air line was used only when temperatures lower than 400°F were necessary. After exiting the fluidized bed, the gas passed through the disengagement section before being vented to the atmosphere. The shell of the fluidized bed was insulated with 2 inches of binderless insulation and outfitted with removable walls on three sides, allowing flexibility in changing feed systems, overflow weirs, and distributor plates. A New York blower Model 144LS with a 1000°F temperature rating was used as an induced draft fan. The test material was fed into the fluidized bed

Figure 6.1 Fluidized Bed Heat Exchanger

Figure 6.2 Fluidized-Bed Heat Exchanger - Solids Inlet View

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through a Ducon Perma-Flow rotating air lock. The solids flowed through the bed and emptied over a weir at the far end.

Type K thermocouples (TC's) were used to measure all temperatures. Shielded thermocouples were used in the bed, and aspiration thermocouples were used to measure all gas temperatures. Thermocouples were located in the following positions:

- One TC at the inlet to the gas burner
- Three aspiration TC's measuring gas temperature below the distributor plate
- Three bare TC's welded to the distributor plate
- Three TC's attached to walls of the air plenum
- Ten shielded TC's in the bed measuring its temperature
- Eight aspiration TC's measuring exiting gas temperature above the bed surface
- Three TC's on the freeboard wall (above the bed)
- Two aspiration TC's measuring gas temperature at the top of the freeboard
- One TC measuring gas temperature entering the exhaust fan

In total, 40 thermocouples were available for measurement and were relocated as needed for the various tests.

Pressure measurements included gas and air orifice pressure losses, pressure loss of the distributor/bed combination, and absolute pressures throughout the system. Dampers on both the burner fan and exhaust fan allowed the velocity to be accurately controlled.

6.2 FLUIDIZED BED COMPONENT

6.2.1 Heat Transfer Media and Fluidization Characteristics

Proper selection of the circulation heat transfer media for the FBWHR system was critical to assure maximum performance and minimum maintenance. Since the media were constantly recirculated from the hot flue gases to the cool combustion air, high thermal shock resistance and low particle attrition were of paramount importance. Equally important was the ability of the material to withstand the corrosive environment of the flue gas. The material needed the following characteristics:

- High Corrosion Resistance
- Minimal Attrition Rate
- Moderate Cost
- Availability in Uniform Sizes
- High Thermal Capacity
- High Temperature Capability
- Compability with Process Furnace Exhaust
- High Thermal Shock Resistance

The material best suited to this application was alumina, a hard ceramic compatible with the aluminum furnace exhaust environment. It was available in several grades ranging from nonspherical low-purity commercial grade to a spherical high-purity medium. For this laboratory test program, nonspherical alumina ranging between 500 and 1000 microns in diameter and of approximately 95-percent purity was selected. This material is commonly available as blasting grain. If attrition, due to the jagged particles or chemical attack of the impurities, became a problem, high purity or spherical particles could be substituted.

When gas was passed vertically through a bed of alumina, it exerted a drag force on the particles. The pressure loss through the bed was proportional to the velocity. When the velocity was increased,

the drag increased, which eventually offset the weight of the particles. This velocity was the minimum fluidization velocity (V_{mf}).

Because the particles were not spherical, the minimum fluidization velocity could not be accurately calculated and any calculations were experimentally verified. The best correlation for the minimum fluidization velocity was that of Wen and Yu (1962):³

$$Re_{mf} = \frac{\rho_f V_{mf} d_p}{\mu} = (33.7^2 + 0.0408 Ar)^{\frac{1}{2}} - 33.7$$

where:

$$Ar = \frac{\rho_f (\rho_s - \rho_f) g d_p^2}{\mu^2}$$

This correlation was tested over a wide temperature range for the 24-mesh alumina blasting grain. The correlation between theoretical and experimental results was excellent, as shown in Table 6.1.

Although it was theoretically possible to operate the FBWHR system at the minimum fluidization velocity, practical considerations required that the superficial velocity be appreciably increased. The primary consideration was imparting sufficient mobility to the particles to assure a smooth flow of solids away from the feed zone and along the bed. Tests simulating the FBWHR system operating conditions indicated that a velocity of three times minimum fluidization velocity was required.

The minimum fluidization velocity varied strongly with particle diameter. Figure 6.4 illustrates this variation for 1000°F - a typical temperature of a fluidized bed. The theoretical correlation of Wen and Yu was used to construct this curve. Also shown in this figure is the minimum operating velocity (three times minimum fluidization velocity) for this range of particle sizes. These curves therefore defined the maximum particle diameter for a specified superficial velocity.

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TABLE G.1
COMPARISON BETWEEN MEASURED AND CALCULATED
MINIMUM FLUIDIZATION VELOCITIES

Temperature (°F)	Measured V_{mf} (ft/sec)	Calculated V_{mf} (ft/sec)
80	1.4	1.42
250	1.2	1.28
425	1.02	1.16
710	1.10	1.00
890	1.0	0.92
1160	0.90	0.85

Figure 6.4 Minimum Fluidization Velocity for Alumina Particles

When the gas velocity was increased above the minimum fluidization velocity, the bed expanded to maintain the drag on each particle equal to its weight. Thus, the void fraction increased with velocity. This is shown in Figure 6.5. The theoretical curve is that of Wen and Yu (1962), and the experimental points are for 24-mesh blasing grain. These results were obtained by placing a known weight of alumina in the laboratory unit and measuring the depth of the bed for different velocities. This depth was measured by mounting a scale on the edge of the distributor plate and recording the lowest level that the fluctuating bed surface attained. It can be seen that there was very good agreement between the experimental and theoretical results.

The maximum superficial velocity was determined by the loss of media. For spherical particles the terminal velocity, or velocity at which the particles would be entrained by the gas, was 15 to 30 times the minimum fluidization velocity, depending on the particle size. However, significant amounts of bed material could be entrained at velocities less than terminal, reducing the operable velocity range. Generally, velocities should be less than one-half of terminal velocity to minimize entrainment. This, therefore, resulted in an operating range of 3 to 7 or 3 to 15 times the minimum fluidization velocity, depending on the upper limit. Tests were not performed to quantify the clutriation, although no noticeable entrainment was observed during the laboratory testing.

6.2.2 Heat Transfer

A fluidized bed could be modelled as a crossflow heat exchanger. Ideally, if both the gas and particles flowed through the heat exchanger without mixing, the thermal effectiveness of the fluidized bed would be

$$\epsilon_p = 1 - \exp(-C_g/C_s)$$

where:

- C_s = thermal capacity of solids
- C_g = thermal capacity of gases.

Figure 6.5 Voidage of Alumina Fluidized Bed

This is graphed in Figure 6.6. However, in a fluidized bed the movement of the particles caused the particles to mix laterally with adjacent material.

A model for the heat transfer was formulated to predict the temperature profile within the bed based on this concept. When the back mixing of the particles was modelled as an effective thermal conductivity, an energy balance on a differential element (shown in Figure 6.7) within the bed yielded

$$\frac{d^2T}{dx^2} - \frac{C_s}{kwh} \frac{dT}{dx} - \frac{C_g}{kwhL} (T - T_g) = 0$$

where:

- T = bed temperature (°F)
- x = distance along bed (ft)
- C_s = thermal capacity of solids (Btu/hr-°F)
- C_g = thermal capacity of gases (Btu/hr-°F)
- T_g = gas temperature (°F)
- k = conductivity of bed (Btu/hr-ft-°F)
- w = width (ft)
- h = height (ft)
- L = length (ft)

This equation was solved using appropriate boundary conditions to give the temperature profile. A family of curves is shown in Figure 6.8 for $C_g/C_s = 1.1$ and various values of the effective conductivity, k_* (kwh/C_sL). When mixing was low, k_* was zero and the temperature profile was that of an ideal crossflow heat exchanger. When k_* was high, the temperature profile became uniform. The degree to which the profile flattened out could be altered by varying the geometric proportions of the bed. A long narrow bed would decrease the effect of having a high conductivity.

Figure 6.6 Ideal Crossflow Heat Exchanger

Figure 6.7 Temperature Profile Model

Figure 6.8 Calculated Temperature Profile in Fluidized Bed

When evaluating the complicated solution of this differential equation at the solids outlet of the heat exchanger, two simplifications could be made. The first simplification, for large k_* , was to that for an ideal mixed flow heat exchanger:

$$\epsilon_p = 1 - \frac{1}{\frac{C_g}{C_s} + 1}.$$

The second, for small k_* , was when a reduction to that of an ideal crossflow heat exchanger could be made:

$$\epsilon_p = 1 - \exp(-C_g/C_s).$$

Laboratory testing showed that this model accurately determined the temperature profile within the bed. Figure 6.9 is a comparison of theoretical and measured temperature profiles showing excellent agreement. Testing also showed that the thermal conductivity was dependent on the superficial velocity varying from almost 0 at minimum fluidization to over 5000 Btu/hr/ft/°F at 10 ft/sec. This is illustrated in Figure 6.10.

Comparison of Figures 6.8 and 6.9 reveals the similarity between a shallow fluidized bed and an ideal crossflow heat exchanger. This suggests that applying a constant crossflow efficiency factor (F) to the ideal performance model, as in:

$$\epsilon_p = F [1 - \exp(-C_g/C_s)],$$

would yield results in close agreement with the more complex performance equation. For the FBWHR system design conditions, corresponding to k_* of approximately 0.1, F would be between 0.91 and 0.95. Laboratory results and performance calculations by these two methods differed by less than 5 percent, providing validation of this simplified nonideal performance model.

Figure 6.9 Experimental Temperature Profile Compared to Curve Fit of Theory to Data

Figure 6.10 Conductivity of Fluidized Bed as a Function of Gas Velocity

Radiation heat losses were also considered in the final analysis of the FBWHR system. If the radiation term was included in the differential equation, it became nonlinear and unsolvable explicitly. As an approximation, it was possible to superimpose a separate analysis of the radiation losses on the temperature profile. This would give an upper bound on the losses due to radiation. If a heat balance was performed on the element of bed shown in Figure 6.11, it was found that:

$$dq = (\dot{m}C_p)_s dT = \sigma \epsilon_o (T^4 - T_w^4) dA.$$

Integrating this equation allows a comparison of ideal and radiation-corrected temperature profiles as shown in Figure 6.12 for a typical unit. It can be seen that the radiation losses resulted in a decrease in the exit temperature.

Figure 6.13 shows the influence of wall temperature and average residence time on loss of heat from the bed. In a typical FBWHR installation, the residence time of the solids was one to five minutes. This, along with only a moderate amount of insulation, would decrease the radiative energy loss to negligible levels.

6.3 LIFT LINE COMPONENT

6.3.1 Theory

In the design of industrial vertical pneumatic conveying systems, choosing the correct velocity at which to transport the solids was critical. Too low a velocity would result in an unacceptable, unstable, slugging flow (choking); too high a velocity would result in excessive gas requirements and high pressure losses. The general relationship between velocity and pressure loss per unit length ($\Delta P/L$), in a dilute-phase vertical riser, is shown schematically in Figure 6.14. When no solids were in the lift line, there was the familiar quadratic relationship between pressure loss and superficial velocity (curve $w = 0$). However,

Figure 6.11 Radiation Loss Model

Figure 6.12 Radiation Heat Losses From a Fluidized Bed

Figure 6.13 Effect of Average Residence Time and Wall Temperature on Radiative Energy Losses From a Fluidized Bed Preheater

Figure 6.14 Pressure Drop-Velocity Characteristics of a Vertical Pneumatic Lift Line

when solids were added to the base of the lift line, this curve was modified into the "U" shape shown in curve w_1 . At high gas velocities, the frictional resistance predominated, and the pressure loss characteristic closely followed the empty tube characteristic, slightly displaced by the limited particle inventory in the lift line. At low gas velocities, however, the particle inventory in the lift line was increased and provided a large static load, thereby increasing the pressure loss. This accounted for the U shape of the curve. At higher solids throughput, the characteristic curve was displaced upward and to the right, shown in Figure 6.14 as curve w_2 . Each of these curves had a minimum pressure loss, shown as point D in curve w_1 . Operation to the right of this point was stable. Operation to the left was unstable - that is, the flow would slip into choking. Choking occurred when the smooth flow of gas and particles broke down into slugs with the attendant oscillating transport. Thus, two critical points were required to characterize the lift line: the choking velocity and the minimum pressure loss.

6.3.2 Lift Line Test Results

Experiments were performed using the apparatus shown in Figure 6.15 to determine the characteristics of the pneumatic lift line. In these tests, the particle flow rate was controlled by a calibrated rotary feeder. The particles dropped from the feeder into an L-valve. Air injected into the L-valve pushed the particles into the lift line, which lifted the material a vertical distance of 22 feet into a cyclone. The lift air was vented outside and the alumina was recirculated.

In the experiments, the solids flow rate was held constant and the lift velocity was varied from a maximum of 100 ft/sec to the choking limit. The pressure loss of the lift line was correlated with the lift velocity and solids flow rate. The results for five different solids flow rates are shown in Figure 6.16. The terminal velocity of the

Figure 6.15 Lift Line and L-Valve Feed Control Test System Configuration

Figure 6.16 Experimental Results of Lift Line Characterization

alumina particles is also shown for reference. Because small changes in the superficial velocity near the terminal velocity yielded large changes in the pressure loss, so much so that operation in this area was highly unstable, it was not possible to collect data at this point. The characteristics of the lift line exhibited in Figure 6.16 fit the general relationship shown in Figure 6.14 and indicated that the lift line must operate at an inlet velocity of approximately 50 ft/sec for stable operation. The lift line pressure loss at a nominal solids flow rate of 0.71 lb/sec was approximately 20 in. w.c.

When the lift line was incorporated into the FBWHR system, the particles that were ejected into the lift line were at 800°F. This altered the lift line characteristics because the air expanded as it was heated by the particles and the kinematic viscosity decreased. The net result was a slight decrease in the air requirements for lifting the alumina and a negligible change in the lift line pressure loss.

6.4 PARTICLE FEED SYSTEM SUBCOMPONENT

One area critical to the FBWHR system was the feed system, which fed hot particles from the upper bed to the lower bed, from the lower bed to the lift line, and from the lift line to the upper bed. Besides feeding particles, this system also had to minimize gas leakage across a pressure difference of up to 23 in. w.c. Presented below are a description of the feeder, results of preliminary experiments to verify its performance, and results of experiments to quantify the gas leakage.

6.4.1 Feeder Description

In the upper hot stage bed the heat transfer media were heated by flue gases from the furnace to temperatures of up to 1700°F. These particles were then removed from the upper bed, transferred to the lower bed, and injected into the lower bed. Flow rate of the media had to be easily controllable and variable from 0 to 7 lb/sec. The design specifications for the hot particle feed system are shown in Table 6.2.

TABLE 6.2
HOT PARTICLE FEED SYSTEM SPECIFICATIONS

Media Temperature	- Up to 1700°F
Media Flow Rate	- Variable, 0 to 7 lb/sec
Gas Leakage	- Less than 0.05 lb/sec
Media Temperature Loss	- Less than 50°F
Pressure Difference	- 6 to 23 in. w.c.
High Reliability	
Low Maintenance	
Ease of Control	

The requirement for minimum gas leakage was particularly hard to satisfy in the dual-fluidized-bed FBWHR system because of the very high pressure differential between the beds. Because the furnace had to be maintained at atmospheric pressure, pressure in the hot bed freeboard would be 2 to 10 in. of water subatmospheric. However, because of burner supply pressure requirements, the pressure in the cold bed freeboard would vary from 4 to 13 in. of water above atmospheric pressure. Thus, the pressure difference could be between 6 to 23 in. w.c.

An L-valve was the feed device capable of meeting the specifications of Table 6.2. This is illustrated in Figure 6.17. The particles would leave the upper fluidized bed over a weir, fall into a downcomer, and form a packed bed. This bed would be sufficiently deep to minimize the air leakage to the flue. A small quantity of air, called aeration air, was injected into the downcomer at the bottom of the L-valve to control the particle removal rate from the downcomer and into the lower bed. Without aeration air, the particles would build up in the downcomer. At high aeration air flow rates, the particle removal rate would be high, and the inventory of particles in the downcomer would be reduced. Thus, particle flow rate was easily controlled with the aeration air.

Figure 6.18 is a schematic of the laboratory particle circulatory system and system pressures. The circulation rate of the alumina particles was controlled by a rotary feeder feeding into the upper fluidized bed. The particle flow rate into the lower fluidized bed and into the lift line matched this rate exactly. The feed mechanism also sealed against the leakage of air from the lift line into the lower fluidized bed and between the two beds. A high leakage rate between various components would decrease the system performance and, in extreme cases, stop the circulation of the particles, rendering the system ineffective.

Figure 6.17 Media Circulation System

Figure 6.18 Particle Circulatory System

To accomplish this sealing, one L-valve was used between the two beds, one L-valve was used between the lower bed and lift line, and a rotary feeder was used between the upper lift line and upper bed.

Thus the L-valve met all the criteria for the hot particle feed system. What made this alternative particularly interesting was the lack of any mechanical moving parts, which maximized reliability and minimized maintenance requirements. Finally, the hybrid fluidized bed/raining bed FBWHR system could use the same particle feed system, eliminating the need for any further development work.

6.4.2 Particle Feed Testing

Laboratory tests were conducted to verify the operation of the L-valves. The tests were to determine particle flow rate dependence upon the aeration air flow rate, and to investigate alternative configurations. Figure 6.19 shows the experimental equipment. A description of the facility was given in Section 6.1. The L-valve was installed between the rotary valve and the distributor plate. This allowed accurate control of the particle flow into the L-valve. Five taps were provided for aeration air, which was obtained from the building compressed air supply. The L-valve was tested with both hot and cold fluidized beds.

Test results are shown in Figure 6.20 for a 2-7/16-in.-diameter horizontal tube (feed angle 0°) into the fluidized bed. Test results are given for three different aeration air tap locations. In all cases, a minimum aeration air flow rate had to be achieved before the particles started flowing. Once this minimum flow was achieved, the particle flow rate increased with increasing aeration. As the aeration air moved through the packed bed it exerted a drag force on the particles. The particles would not start moving until a critical drag force was achieved. For a high particle flow rate tap location (3) seemed to be the best, but for control, the linear characteristics of location (1) were better.

Figure 6.19 Schematic Showing L-Valve at 20° from Horizontal

Figure 6.20 Performance of a Horizontal L-Valve

Figure 6.21 shows similar results for a feed angle of 20°. In this case, the minimum aeration air flow rate decreased since gravity was assisting in dragging the particles into the bed. In the tests with aeration at locations (1) and (2), a maximum particle flow was observed.

The design particle flow rate could be achieved by using a layer feed or multiple feeds into the bed. At these flows the aeration flow constituted approximately 1 percent of the combustion air flow.

6.4.3 Feeder Sealing

As previously explained, the particle feed system fed the hot particles while simultaneously preventing excessive gas leakage. For an L-valve, a packed column of particles was used to accomplish this seal. Gas flow rate through a packed bed was most accurately calculated from the Ergun Equation: ⁴

$$\frac{\Delta P g_c}{L} = \frac{150 (1 - \epsilon_o)^2 \mu V_r}{\phi^2 \epsilon_o^3 d_p^2} + \frac{1.75 (1 - \epsilon_o) V_r^2 \rho_f}{\phi \epsilon_o^3 d_p}$$

where:

- ΔP = pressure across packed bed
- L = height of particles in tube
- g_c = gravitational constant
- ϵ_o = porosity of particles (fixed bed)
- V_r = gas velocity relative to the particles
- μ = viscosity
- ϕ = particle shape factor
- d_p = particle diameter
- ρ_f = gas density.

All the parameters in this equation are either design or physical constants, except the gas leakage rate. Thus the leakage rate can be calculated with this correlation.

Figure 6.21 Particle Feed Test with a 20° Entrance Angle

Experimental tests were conducted to verify the Ergun Equation for nonspherical alumina over the size range of interest. These experiments were performed in a Plexiglas apparatus, as shown in Figure 6.22. The box simulated the lower bed freeboard, and the tube simulated the downcomer. The box was completely sealed, and the tube was open at the upper end. The tube was filled with alumina particles to a given height to simulate the packed bed. The box was pressurized using a nitrogen bottle to set up a pressure differential across the packed bed. Since the box was sealed, the nitrogen flow rate into the box was equal to the leakage through the packed bed. Nitrogen flow rate, box pressure, and packed bed height were measured during the testing. Leakage tests were performed for various sizes of alumina particle diameters ranging from 0.031 in. to 0.111 in.

The experimental results and analytical predictions are shown in Figure 6.23 for 24-mesh alumina. As anticipated, the gas leakage increased with the pressure gradient. The correlation between theoretical and experimental results was best at low values of the pressure gradient; the deviation increased as the pressure gradient was increased. This was because of the experimental difficulty of maintaining a uniform pressure difference at high leakage rates.

Test results for different particle diameters are shown in Figures 6.23 through 6.26. As predicted by the theory, leakage rates increased as the particle size increased. The correlation between theoretical and experimental results was reasonable for all tests.

These tests were conducted for a static bed. In the actual FBWHR system the particles would be moving downward, dragging the gas along with them and decreasing the leakage rate. Proper design of the feed system would minimize the leakage rate. For example, for a design leakage velocity of 0.5 ft/sec and a pressure difference of 21 in. w.c., a packed bed 2.2 feet in height was required.

Figure 6.22 Schematic of Sealing Experiment

Figure 6.23 Leakage Test Using 24-Mesh Particles

Figure 6.24 Leakage Test Using 8-Mesh Particles

Figure 6.25 Leakage Test Using 12-Mesh Particles

Figure 6.26 Leakage Test Using 16-Mesh Particles

6.4.4 Feeder Control

The flow rate of particles through the L-valves was controlled by adjusting the aeration air flow. A pressure switch was used to sense the level of the particle inventory in the standpipe. The pressure difference increased with inventory height. When the inventory rose above a specified height, the aeration flow was turned on. When it fell below a different height, the aeration flow was turned off. This on-off system, illustrated in Figure 6.27, caused a pulsatile flow of particles out of the L-valve. Occasionally this pulsatile flow caused particle surges that led to lift line choking. The pulsatile flow also caused large fluctuations in the lift line flow.

To solve this problem, a high-low control system was developed. In this system, the aeration flow varied between a high and low flow rate and was not completely shut off. This system is shown in Figure 6.28. The high-low controller minimized both the surge of solids and the fluctuations of the lift line air flow. Figure 6.29 illustrates the operating characteristics of the two controllers. Although the average solids flow and lift line air flow were the same for both systems, the fluctuations in the high-low control were much lower.

6.5 DISTRIBUTOR PLATE SUBCOMPONENT

One critical component in any fluidized-bed system is the distributor plate. The purpose of this plate is to stabilize the bed by inducing a flow-dependent pressure loss. From systems requirements, this pressure loss must be minimized, and the plate must survive in the environment. This was a particularly severe environmental problem for the upper plate in the dual-fluidized-bed FBWHR system. This section describes the stability criteria and configuration constraints of a distributor plate.

Figure 6.27 L-Valve On-Off Feed Control System

Figure 6.28 L-Valve High-Low Feed Control System

Figure 6.29 Comparison of On-Off and High-Low L-Valve Controllers

6.5.1 Bed Stability

One of the concerns in the design of a FBWHR system was the parasitic power required to overcome the pressure loss of the heat exchanger. To minimize this, a shallow bed was used, the minimum depth of which was determined by mechanical constraints such as fastening or warping of the distributor plate. Once the bed depth was determined, the minimum pressure loss for the distributor plate was fixed by bed stability constraints. However, this minimum pressure loss was not previously addressed for shallow bed applications.

The purpose of a distributor plate in a fluidized bed is twofold. First, it supports the bed material when the gas flow is turned off. Second, and more important, its pressure loss forces the gas flow to be distributed evenly over the area of the bed. The degree to which the distribution is even is dependent on the relative pressure losses of the distributor plate and the bed itself. Several investigations (Hiby,⁵ 1964; Davidson and Harrison,⁶ 1971; and Siegel,⁷ 1976) investigated this problem for deep beds (e.g., when the depth of the bed was larger than its diameter). However, very little has been done in shallow fluidized beds.

Siegel (1976) analytically developed a criterion for stability of porous distributor plates; i.e., plates with a linear pressure loss characteristic. This analysis was modified to apply to the case of a quadratic pressure loss characteristic typical of perforated plates. Results of this analysis indicated that the minimum pressure loss varied with material properties and operating conditions. This variation with superficial velocity, V , is shown in Figure 6.30. The minimum pressure loss ratio varied from 0.3 to 0.5 over the potential operating range of the preheater. For a 3-in. expanded bed depth, total pressure loss across the preheater could therefore be as little as 6.0 in. w.c.

Figure 6.30 Pressure Drop Ratio as a Function of Velocity for Alumina

Experiments were performed to determine the nature of the instability itself. This was accomplished by maintaining constant test conditions (velocity and temperature) and observing the fluidization of the bed as the bed depth was increased. As the depth of the bed increased, the pressure loss increased and the ratio ($\Delta P_D / \Delta P_B$) decreased, pushing the system into the unstable regime. Figure 6.31 shows experimental results. In Tests I and III,* the bed should have been in stable fluidization, according to the theory. Observations revealed that all areas of the bed were evenly fluidized and pressure loss readings were steady. However, in the other tests, as the depth of the bed was increased the bed became unstable, and channeling occurred. As the bed depth increased, the fluctuations of pressure became more severe and some segments of the bed became defluidized and stagnant. This was as predicted by the theoretical analysis.

6.5.2 Plate Configuration

Perforated plates were selected for the distributor plates in the FBWHR system. Though other configurations could have been selected (bubble caps, slotted plates) their features were not required for this system. Perforated plates had the advantages of ease of fabricability and specification.

As stated in the previous section, the minimum pressure loss across the plate was related to the bed pressure loss. This relationship was needed to maintain a stable bed. If the pressure loss was known, both the open area and plate area could be calculated for a given mass flow rate. The percent open area was the open area divided by the plate area. The percent open area is given in Figure 6.32 as a function of total pressure loss and plate area. Figure 6.33 shows the relationship between the percent open area and the hole spacing, assuming 1/10-in. hole diameter and straight rows of holes.

*A test consisted of all the points arranged vertically in Figure 6.31. Test I had two points, Test II had six points, etc.

Figure 6.31 Experimental Data Range for Pressure Drop Ratio

Figure 6.32 Variation of Open Area with Distributor Plate Pressure Loss and Cross Section

Figure 6.33 Variation of Hole Center Spacing with Open Area for 1/10-in. Hole Diameter

It was desirable to make the hole diameter as large as possible in order to:

- Minimize fouling problems
- Minimize fabrication difficulties

However, the maximum hole diameter was limited due to particles falling through the hole. A hole diameter equal to about $2\frac{1}{2}$ -particle diameters was found to be adequate. A $2\frac{1}{2}$ -particle-diameter rule and 800- μ m particles would require the hole diameters to be not much larger than 1/16-in.

6.6 FIELD TEST RESULTS OF 1/10-SCALE TEST UNIT

Field testing of a 1/10-scale hot stage fluidized bed, the most critical component of a dual-fluidized-bed FBWHR system, was conducted at VMC's plant in Sandusky, Ohio. The objectives of these tests were to investigate bed media corrosion, on-line distributor plate cleaning, and distributor plate corrosion/stress.

6.6.1 Field Test Unit Description

The 1/10-scale test unit was installed in VMC's plant in Sandusky, Ohio, sidestream to furnace No. 2. Figure 6.34 is a schematic of this arrangement, and Figure 6.35 is a photograph of the installed unit. Ten percent of the 3500 scfm of flue gases were pulled by the blower through a 1-ft² opening in the flue, diluted to 2000°F in the inlet duct, and then travelled through the distributor plate. The gases exited the freeboard, were diluted to 500°F with ambient air to protect the blower, and then returned to the top of the flue.

A 2-inch, nonflowing bed of 24 mesh, +99% pure alumina was fluidized on top of the distributor plate. Since the bed medium was not flowing, its temperature would be higher than that expected in the full-scale FBWHR system, which would use a moving bed. To simulate the correct media temperature (and hence corrosion rates), dilution air

Figure 6.34 1/10-Scale Fouling Test Unit

Figure 8.2 Perspective View of FBWHR System

Figure 6.35 1/10 Scale Fouling Unit Installed Sidestream to VMC's No. 2 Furnace

was forced through an opening in the freeboard, cooling the freeboard walls and, by radiative transfer, the alumina media. The dilution air was adjusted until the desired alumina temperature was achieved. Since the dilution air was brought in downstream of the alumina it could not alter the flue gas constituents in contact with the alumina and did not directly affect the corrosion process.

Temperatures, pressures, and flow rates recorded during a typical test run are shown in Figure 6.36. The flue gases extracted at 270 scfm exited the flue at 2020°F and entered the plenum at 1900°F. Heat losses in the plenum and freeboard amounted to 3.3 percent of the total flue gas heat content, resulting in the flue gas exiting the freeboard at 1840°F. The distributor plate temperature of 1820°F was close to that of the exiting gas temperature. Heat losses in the uninsulated duct were designed to be high, at 21 percent, to minimize the required dilution air into the fan. The distributor plate pressure loss was 5.5 in. w.c., and the orifice pressure loss was 0.7 in. w.c.

6.6.2 Bed Media Test Results

The temperatures of the bed media and flue gas during operation of the 1/10-scale fouling unit for a typical one-hour period are shown in Figure 6.37. At high fire (2030°F flue gas temperature) the media temperature was 1631°F, or about equal to its design maximum temperature. As the flue gas temperature dropped, the bed media temperature also dropped to 860°F, about equal to its design minimum temperature. Hence, the bed media, even though not flowing, experienced a temperature range equal to that expected in the full-scale system.

The bed media were sampled at the times indicated in Table 6.3 and sent to the Oak Ridge National Laboratories for analysis. Unexposed and exposed (1248 hours) particles are shown in Figures 6.38 and 6.39, respectively. The results of this analysis were that after 1248 hours of bed media exposure:

A-9297

Figure 6.36 Typical 1/10-Scale Operating Conditions

A-9301

Figure 6.37 Bed Media Temperature

A-9273

TABLE 6.3
BED MEDIA SAMPLES

Sample	Hours of Exposure
1	144
2	336
3	624
4	816
5	1248

2000 μm

10X

50 μm

400X

Figure 6.38 Unexposed Alumina Fluidization Media

Figure 6.39 Alumina Fluidization Media Exposed 1248 Hours

- The media showed no change in either porosity, size, or shape;
- The media picked up a 2- to 3-micron-thick coating. No degradation of the media by the coating was observed; and
- The surface layer was shown by electron microprobe analysis to contain Ca, Mo, Si, Fe, and Cr, and by x-ray diffraction analysis to contain crystalline CaMoO_4 and probably glass, as indicated by broad peaks in the low angle region of the x-ray diffraction pattern.

Conclusions from this test were that the alumina bed media could withstand the flue gases of a secondary aluminum furnace. In particular, after over 1000 hours of exposure at varied temperatures the alumina particles remained intact with no attrition, agglomeration, or corrosion problems.

6.6.3 Distributor Plate Fouling and Cleaning Test Results

The fouling test unit was operated for a total of 1000 hours during an initial period. During that time four fouling tests were conducted, as depicted in Figure 6.40. The first, second, and fourth tests were to characterize the distributor plate fouling in terms of both identifying the fouling constituents and evaluating the time required to foul. In addition, the first test evaluated off-line cleaning systems. The second and third test runs were conducted to evaluate the on-line cleaning systems.

6.6.3.1 The Distributor Plate ΔP

Distributor plate fouling could be identified by observing the distributor plate pressure loss. Because the flow rate was automatically held fixed by the control system, a decrease in the distributor plate open area, due to hole fouling, resulted in an increase in distributor plate pressure loss. This is shown by the governing flow equation:

Figure 6.40 Initial Field Test Unit Experience

$$\dot{m} = C A \sqrt{\Delta P}$$

where:

$\dot{m}$ = flow rate

ΔP = distributor plate pressure drop

C = constant.

For a fixed $\dot{m}$, a decrease in A results in an increase in ΔP .

If the distributor plate ΔP continued to increase, the blower capability would eventually be exceeded (22 in. w.c.). Additional fouling would thus also decrease the flow rate. If fouling continued unabated, the distributor plate would become completely plugged, resulting in zero flow rate and a distributor plate ΔP equal to the blower's maximum capability.

The distributor plate fouling and orifice ΔP for a typical test run are shown in Figure 6.41. For the first 60 hours, the distributor plate ΔP increased continuously. After 60 hours, the continuing fouling resulted in a total system pressure loss greater than the blower's capability, hence the flow rate (orifice ΔP) decreased. After about 160 hours, the distributor plate was completely plugged, as evidenced by zero flow rate (ΔP orifice = 0) and a distributor plate ΔP greater than 20 in. w.c.

The flue gas and plenum gas temperatures during this test are shown in Figure 6.42. The plenum gas temperature was close to the flue gas temperature until the flow rate began to decrease (beyond 60 hours). As the flow rate continued to decrease, the temperature difference between the plenum gas and flue gas increased. At 160 hours fouling was complete, there was little gas flow, and the temperature difference was large.

Figure 6.41 Typical Distributor Plate Fouling Characteristics
Without On-Line Cleaning - Test II

Figure 6.42 Fouling Test Unit Temperature - Test II

6.6.3.2 Fouling Constituent Characterization

During Tests I and II, the distributor plate was allowed to foul completely, resulting in the complete plugging of the distributor plate. The unit was then partially disassembled and samples of the fouling material were taken. The samples were sent to Oak Ridge National Laboratories and Herron Testing Laboratories, Inc., Cleveland, Ohio, for an x-ray diffraction scan and a spark source mass spectroscopy analysis. The results are given in Table 6.4.

The major deposit constituent in Test I was aluminum, and in Test II, magnesium. Each could be traced to the chlorinating process. In Test I, the aluminum was probably a result of the chlorine combining with aluminum, volatilizing, and then reacting with water to form Al_2O_3 . In the second test, magnesium probably formed from entrained MgCl , which also reacted with water to form MgO . The source of the other constituents in each test run could be traced to the fluxes added to the melt, such as sodium and potassium, and the remaining constituents could be traced to either the melt itself or the furnace wall material.

6.6.3.3 Effect of Temperature on Fouling

Temperature could affect fouling because at high temperatures low vapor pressure flue gas constituents were unable to condense on the distributor plate, or if already condensed, would vaporize off the plate. As an example, KCl , CaSO_4 , and NaCl fluxes vaporized between 1400° and 2600°F . If these were binders holding the deposit, then heating to this range would also remove the deposit.

The temperature effect on fouling was verified by observing the plate ΔP and the flue gas temperature during a 10-hour period, as shown in Figure 6.43. The ΔP changed at the same time the flue gas temperature did and in the opposite direction. Between 4 and 5 hours the temperature increased from 700° to 1900°F . At the same time, the plate ΔP decreased from 10 in. w.c. to 5 in. w.c., presumably because

TABLE 6.4
DEPOSIT ANALYSIS

Material	Deposit from Test		Assumed Compound	Possible Source
	Run No. I	Run No. II		
Aluminum	Major	Minor	Al_2O_3	Volatilized AlCl
Magnesium	1%	Major	MgO^*	Entrained MgCl
Iron	Minor	2%	Fe_2O_3	Duct Corrosion
Sodium	2%	Minor	NaCl^*	Flux
Calcium	1%	4%	$\text{Ca}(\text{ClO}_4)_2^*$	Refractory
Potassium	1%	2%	KCl^*	Flux
Silicon	0.3%	2%	SiO_2	Alloying Material
Chromium	0.8%	2%	CrO	
Zirconia	0.4%	2%	ZrO_2	

*Indicates Water Soluble

Figure 6.43 Time Variation of Fouling Characteristics

the fouling particulates vaporized off the plate. Conversely, when the temperature decreased the ΔP increased. From 5 to 8 hours the flue gas temperature dropped back to 700°F and the plate pressure drop increased to 15 in. w.c., presumably because flue gas constituents were able to condense on the plate.

This test suggested that fouling could be minimized by stopping the blower whenever the flue gas temperature went below 1000°F, thus avoiding the condensing part of the observed cycle. This, however, would not eliminate fouling because some flue gas constituents condensed at temperatures above the maximum flue gas temperature.

6.6.3.4 Effect of the Chlorinating System on Fouling

The previously mentioned fouling tests were conducted while VMC was using a continuous chlorinating system that was a relatively new design. In this design, chlorine was injected into the batch continuously during the charging cycle, as contrasted to the old-style system in which the chlorinating was done during a holding period at the completion of the charging cycle. The continuous chlorinating system eliminated this holding period, decreasing the total furnace fuel consumption. Because of these fuel savings, industry trends have been to use the continuous chlorinating system. Hence, these fouling test results were representative of what would be expected in any furnace. However, due to the recent introduction of this system, one concern was how sensitive the fouling phenomenon was to the chlorinating system.

Fortunately, VMC switched from continuous to batch chlorinating on the No. 2 furnace for about two months. During that time, Test IV was conducted, in which no on-line cleaning was used. The purpose of this test was to compare the fouling characteristics with the batch chlorinating system in place to those obtained in Tests I and II, where the continuous system was used.

The results of Test IV were:

- The distributor plate fouled completely. The appearance of the fouled plate and the fouling material was identical to that of the previous fouling test results.
- The time for the fouling to occur was 350 hours (Figure 6.40). This was of the same order as Tests I and II (125 to 350 hours).

The fouling material for this run was sampled, analyzed, and was comparable to the previous samples.

These preliminary results indicated that fouling would occur and was only secondarily dependent upon the chlorinating system.

6.6.3.5 General Cleaning System Effect on Fouling

Several methods were tested to clean the distributor plate while the unit was off-line* and also on-line. A summary of the methods used off-line is shown in Table 6.5. The air lance, steam, and water wash were all effective in cleaning the plate. In the case of the water wash and steam cleaning, water soluble constituents of the fouling deposit, such as MgO, NaCl, KCl, and $\text{Ca}(\text{ClO}_4)_2$, were easily washed away.

The wire brush and the mechanical rapper were not effective in cleaning the plate off-line.

The air lance was also used on-line to prevent the plate from fouling during operation. The air lance was used once a day for about three minutes. The flow rate through the air lance was less than 0.01 percent of the total flue gas flow rate through the test unit and resulted in a negligible gas temperature drop. The effect of the air lance on plate ΔP is shown in Figure 6.44. While the distributor plate ΔP increased each day, its trend was reversed by the air lance;

*No flue gases flowing through the test unit.

TABLE 6.5
ALTERNATIVE CLEANING TECHNIQUES

Air Lance	Removes Deposit Continuous Use Inhibits Fouling
Moderate Pressure Steam	Removes Deposit
Water Wash	Removes Deposit
Mechanical Rapper	No Effect
Wire Brush	No Effect

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Figure 6.44 Distributor Plate Pressure Loss - Test III

therefore, the ΔP never rose much above the design value during the entire 80-hour test. For instance, from 21 to 25 hours the ΔP increased from 6 in. w.c. to 7.6 in. w.c. The air lance was then used and the ΔP decreased back down to 6 in. w.c. This could be compared to the tests without on-line cleaning (Figure 6.41) where, during the same time span, the ΔP increased by a factor of 2 over the design value.

The plenum gas and flue gas temperatures during the air lance cleaning are shown in Figure 6.45. The plenum gas temperature never went much below the flue gas temperature, confirming the air lance's effectiveness.

6.6.3.6 Distributor Plate On-Line Cleaning-Duration Testing

Initial test results showed that the distributor plate fouled while a fluidized bed was operated with a secondary aluminum smelter; the effect of such fouling was to degrade the performance of the fluidized bed. Several cleaning systems were tested that showed the distributor plate could be cleaned after the plate had fouled. However, for practical operation of an FBWHR system it was necessary to keep the plate from fouling without the need for periodic shutdowns. A successful demonstration of such a long endurance on-line cleaning system was conducted during a final test period.

An air lance, shown in Figure 6.46, was used on-line to prevent the plate from fouling during operation. The air lance consisted of a 1/16- or 1/8-in.-diameter nozzle at the end of a 1/4-in.-diameter stainless steel tube. The tube fit through the side of the plenum wall so that the nozzle was positioned in the plenum directly under the distributor plate. The opposite end of the tubing was connected to a high-pressure air supply. The distributor plate was cleaned by the high velocity air, generated by the nozzle, as it impacted the distributor plate. The impact force could be controlled by varying the air pressure, altering nozzle geometry, or positioning the nozzle at varying distances

A-8072

Figure 6.45 Fouling Test Unit Temperatures - Test III

A-9102

Figure 6.46 Air Lance Cleaning System

from the plate. The entire distributor plate could be cleaned by manually translating and rotating a single air lance. The air lance was used once a day on the 1/10-scale test unit for a three-minute period. The flow rate through the air lance was less than 0.01 percent of the total flue gas flow rate through the test unit, resulting in a negligible loss of gas temperature.

The ability of the air lance to keep the plate clean during on-line testing, as compared to other tests without on-line cleaning, is depicted in Figure 6.47. During Tests I and IV, which had no on-line cleaning, fouling occurred between 125 to 360 hours. With on-line cleaning, the fluidized bed was operated for 900 hours with no indication of fouling. This was 3 to 8 times longer than had been possible without on-line cleaning. Further, the test was stopped solely due to unrelated test unit repairs. Since there was no indication of fouling prior to shutdown, it is likely that the on-line duration cleaning test could have continued for much longer than 900 hours.

Several separate measurements were used to assess the ability of the air lance to keep the distributor plate from fouling. These indications of gas flow were:

- A gradual increase in the distributor plate pressure loss to 22 in. w.c., the blower's maximum capacity at the no-flow condition;
- A gradual decrease in the orifice plate pressure loss to 0 in. w.c., indicating no flow existed;
- An increase in the difference between the test unit operating temperature and the furnace flue temperature, confirming that no makeup flue gas was entering the test unit, and the fluidized bed was beginning to cool down; and
- Visual inspections of the distributor plate holes.

Figure 6.47 Field Test Unit Experience

A comparison of the distributor plate pressure loss and orifice pressure loss (a measure of total flow rate through the plate) with and without on-line cleaning is given in Figures 6.48 and 6.49, respectively. Without on-line cleaning, the plate pressure loss continuously increased, indicating a buildup of material in the plate holes. After 120 hours, the plate pressure loss had quadrupled to 20 in. w.c., which was the blower's maximum capacity. The flow rate during this test remained constant for the first 60 hours. Beyond this, the flow rate continuously decreased as the blower's air flow delivery was curtailed due to the plate's high pressure loss. After 170 hours the flow rate was zero, indicating that complete distributor plate fouling had occurred.

With the use of on-line cleaning (Figure 6.48), neither an increase in plate pressure loss nor a decrease in flow rate was observed after 900 hours of testing. In fact, the opposite occurred for the first 200 hours: the distributor plate pressure loss decreased while the orifice pressure loss increased. These trends, possibly the result of the smoothing of plate holes or the partial blockage of the orifice, ceased after 200 hours of operation. The above indicates that the air lance was maintaining the plate free of any fouling deposits.

Comparisons of the plate and flue gas temperatures for the tests with and without on-line cleaning are given in Figures 6.50 and 6.51, respectively. Without on-line cleaning, the temperature difference between the plenum gas (similar to the plate temperature) and the flue gas increased because of the decreasing flow. At 160 hours fouling was complete (see Figure 6.51); there was little gas flow and the temperature difference was large. However, with on-line cleaning (Figure 6.50) the plate temperature closely followed changes in the flue gas temperature, again indicating no reduction in flow was occurring and, hence, the plate remained unfouled.

Figure 6.48 Typical Distributor Plate Characteristics With On-Line Cleaning - Test V

Figure 6.49 Typical Distributor Plate Fouling Characteristics Without On-Line Cleaning - Test II

Figure 6.50 Typical Distributor Plate and Flue Temperatures with On-Line Cleaning - Test V

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Figure 6.51 Fouling Test Unit Temperature - Test II

6.6.3.7 Conclusions

The following conclusions could be drawn from the test results:

- The distributor plate could be kept sufficiently clean for at least 900 hours continuously so that operation of the FBWHR system was unaffected by the flue gas constituents;
- The desired plate pressure loss and flow rate could be maintained, thus assuring stable operation and design performance of the FBWHR system; and
- The air lance was a proven cleaning system: it worked, it required little energy to operate, and it had a negligible impact on flue gas energy content.

6.6.4 Scaled Distributor Plate Test Results

The distributor plate used in the 1/10-scale fouling unit was a 38-in. x 12-in. x 1/8-in., coated Inconel 625 plate, shown in Figure 6.52. The Inconel 625 plate was aluminum diffusion coated to enhance its corrosion resistance. The plate length, width, and perforation patterns were fixed by flow and heat transfer design requirements.

Since the distributor plate was stressed as a result of the pressure loss imposed on it by the flowing flue gases, an important design consideration was the maximum allowable stress that could be tolerated and yield an acceptable plate life. This was determined by:

- Plate stress (thickness dependent)
- Failure mechanism
- Material properties (temperature dependent)

6.6.4.1 Stress Analysis

During operation, gas passed through the plate perforations, causing a pressure loss of 6 in. w.c. The plate stress due to this pressure loss could be calculated from:

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Figure 6.52 Distributor Plate Used in 1/10-Scale Fouling Test Unit

$$\sigma = \frac{CQ}{t^2}$$

where:

σ = bending stress (ksi)

C = constant depending on the edge supports, stress concentration factor, and plate dimensions

Q = loading (lb)

t = plate thickness (in.).

The plate stress as a function of thickness is given in Figure 6.53, assuming the plate edges are fixed (that is, the edges can move longitudinally in response to the plate's thermal expansion but cannot rotate). Note that the stress decreased with increasing plate thickness.

Since the stress on the plate was not abruptly cycled but remained relatively constant for approximately 12-hour periods, failure due to long-term static loading was the primary concern. Hence, the plate was designed to maximize the stress rupture life and to minimize the creep.

The stress rupture life of the plate is given in Figure 6.54.⁸ The distributor plate temperature was measured to be about 1800°F during high fire. To attain a plate life greater than 10,000 hours total (corresponding to 5000 hours at the design temperature of 1850°F), the allowable stress was 1.5 ksi. Similarly, for an acceptable creep of between 1 and 10 percent/1000 hours, the allowable stress was again 1.5 ksi (Figure 6.55). A plate thickness of 1/8 inch was chosen to be an optimum compromise between machining ease and design stress. From Figure 6.53, this resulted in a stress of about 0.7 ksi, well below the allowable stress.

Figure 6.53 Distributor Plate Stress

A-7545

Figure 6.54 Stress Rupture Life of Solution-Treated Inconel 625

A-9099

Figure 6.55 Creep Strength of Solution-Treated Inconel 625

The yield strength of Inconel 625 is shown in Figure 6.56. The design stress again was well below the maximum allowable yield stress for the material.

6.6.4.2 Experimental Results

An initial distributor plate, illustrated in Figure 6.52, operated in the test unit continuously for 900 hours without malfunctioning. (Test operating conditions for Tests I through IV were described in Section 6.6.3 and shown in Figure 6.40.) However, at 900 hours, the plate edge had warped so that the edge seal was lost, leading to escaping flue gases and overheating of the test unit shell. The test unit was shut down and the plate was removed for examination.

Figure 6.57 is a photograph of the plate after 900 hours, at the end of Test IV (6/8 - 6/22). For comparison, Figure 6.58 shows the plate prior to testing. The plate, originally flat, bowed in the direction of the gas flow. The maximum depth of the bow, as measured relative to the plate's edges, was 5 inches. In addition, a crack occurred, starting from a machined slot at the plate edge, as shown in the top of Figure 6.57 and in the close-up (Figure 6.59).

The plate was routinely examined about 150 hours before the failure. Since no distortions were observed at that time, the plate failure was not gradual but, rather, occurred during a period of 150 hours, or less.

The temperature of the plate for a 48-hour period, starting 3 days prior to the failure, is shown in Figure 6.60 with corresponding flue gas temperatures in Figure 6.61. As shown, the plate temperature did not always follow the flue gas temperature. Several plate temperature excursions to 2500° to 3000°F (about 500° to 1000°F higher than

A-9100

Figure 6.56 High-Temperature Properties of Hot-Rolled Solution-Treated Inconel 625

Figure 6.57 Distributor Plate No. 1 After 1190 Hours of Exposure,
at the End of Test IV (6 / 8 - 6 / 22)

Figure 6.58 Typical Distributor Plate Prior to Exposure

Figure 6.59 Close-up of Stress Crack on Plate No. 1

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Figure 6.60 Plate Temperature at the End of Test IV

A-9108

Figure 6.61 Flue Gas Temperature at the End of Test IV

the flue gas temperature) occurred. The day before the plate failed, all plate thermocouple signals ceased. Since Inconel-sheathed Type K thermocouples were used (recommended maximum temperature was 2100°F), it is possible that the thermocouples burned out.

The plate was removed after the failure and sectioned for a photomicrographical analysis, as shown in Figure 6.62. Intergranular attack had occurred on each side of the plate to a depth of about 0.041 inch, or about 1/3 of the plate's thickness. Since the damage occurred equally on each side, the plate's remaining effective thickness was 1/3 of its original thickness.

A second distributor plate, identical to the first, was placed in the test unit and operated in a fashion similar to the first. The history of this experiment, Test V (8/19 - 10/6), was presented in Section 6.6.3.6 of this report. After 900 hours of testing, the distributor plate again failed. Photographs of the plate are shown in Figures 6.63 and 6.64. Excessive bowing and stress cracks were again observed.

Unlike the previous test, no excessive plate temperatures were observed just prior to the failure. An example of the plate and flue gas temperature is shown in Figure 6.65. However, the data acquisition system malfunctioned just prior to the plate failure, so no conclusions could be drawn about the plate's temperature immediately preceding the failure.

6.6.4.3 Discussion of Observations on First Two Distributor Plates

The measured plate temperature could exceed the flue gas temperature only if combustion occurred in the test unit. It is possible that the furnace was operating fuel rich and that some of the excess fuel was entering the test unit and burning on or near the distributor plate. This may have occurred due to a change VMC made in their charging methods prior to the plate failure.

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Figure 6.62 Photomicrograph of Sectioned Distributor
Plate No. 1 After 900 Hours of Operation
(70 x)

Figure 6.63 Edge View of Distributor Plate No. 2 After 900 Hours of Testing, at the End of Test V (8/19 - 10/6)

Figure 6.64 Close-up of Stress Crack on Plate No. 2

Figure 6.65 Typical Distributor Plate Temperature,
at the End of Test V

Originally, VMC loaded their scrap into the open hearth charging well. For the time preceding the failure of plate No. 1, they front-loaded sows directly into the furnace, splashing molten aluminum into the burners. This plugged the burner air passages, reducing the air flow and, hence, increasing the fuel-to-air ratio beyond stoichiometric.

As the burner plugging continued, the burners no longer maintained the required heat flux, even when firing continuously on high fire. This resulted in a reduction in flue gas and melt temperatures, necessitating periodic burner cleaning. After the burners were cleaned and reinstalled, the symptoms disappeared. The symptoms eventually reappeared as the front loading continued. In addition, flames were observed exiting the flue and other furnace openings during this period, indicating a fuel rich condition in the furnace.

The events leading to the failure of plate No. 1 can be reconstructed from the preceding observations. A similar series of events presumably occurred with plate No. 2. It is probable that excess fuel was entering the flue and test unit as a result of the burner air passage plugging. Also, a slight amount of dilution air was periodically passed into the plenum. While calculations showed that the fuel and air flow rates were insufficient to uniformly heat up the test unit to 2500°F, it was possible to locally heat the plate to higher values if the plate acted as a flame holder. In fact, the edges of the plate perforations could act as excellent flame holders for a slow burning mixture such as natural gas and air.⁹

Since corrosion rates were a strong function of temperature, the corrosion rate of the plate could have been accelerated. A guideline used in the gas turbine industry is that corrosion rates increase by a factor of $2\frac{1}{2}$ for every temperature increase of 100°F. As an example,

if the plate temperature increased from a design temperature of 1850° to 2550°F, a 700°F increase, the corrosion rate would have accelerated by a factor of 600 (2.5^7). If this occurred for 10 hours, the observed corrosion at 2550°F would have been equivalent to the corrosion expected for 6000 hours at design temperature.

Corrosion reduced the plate's effective thickness from 0.125 to 0.041 inch. As seen in Figure 6.53, the plate stress would then increase to about 55 ksi. This exceeded the yield strength which led to the excessive plate bowing. In addition, Figure 6.54 shows that stress rupture would also be probable, thus, accounting for the stress crack shown in Figure 6.59.

6.6.4.4 Third Distributor Plate

During Test VI, the 1/10-scale distributor plate operated at its design temperature, as shown in Figure 6.66. The plate temperature followed the flue gas temperature, reaching a maximum of 1845°F when the flue gas temperature was 2050°F. This was the third plate used in the fouling unit; the first two had failed due to high-temperature accelerated corrosion after each had about 900 hours of exposure time.

After 1250 hours of exposure testing the furnace was shut down by VMC. The plate had not failed up to that time nor was there any excessive plate distortion (see Figure 6.67). There are three possible explanations for the difference in plate life between this plate and the first two:

- 1) No over-temperature conditions existed during this test - the plate therefore, sustained less corrosion damage than the previous two plates. In at least one of the previous tests, temperatures were known to have exceeded 2400°F.

Figure 6.66 Distributor Plate Temperature

Figure 6.67 Field Test Unit Experience

- 2) Corrosion damage was significant, but because of typical variations in material properties or furnace conditions failure had not yet occurred at 1250 hours.
- 3) The bed media, not present during testing of the first two plates, mitigated the corrosion effect by cooling the plate, particularly during over-temperature excursions.

6.6.4.5 Conclusions

The following conclusions can be drawn from the preceding section:

- Alloy distributor plates could survive for at least 900 hours out of the approximate goal of 10,000 hours.
- Over-temperature failure of the plate was possible. The hypothesized failure mechanism was high-temperature accelerated corrosion leading to increased stress and plate yielding.
- The life of alloy distributor plates could be extended by:
 - Assuring fail-safe over-temperature protection
 - Using alternate manufacturing techniques, such as laser drilling, to further decrease the design stress by increasing the perforated plate thickness beyond 1/8 inch
 - Using a more costly, higher quality corrosion-resistant material
- The future development of ceramic distributor plates would be ideal, since over-temperature problems would be significantly reduced by the ceramic's excellent high-temperature properties.

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7. LABORATORY TESTING OF THE FBWHR SYSTEM

Two related FBWHR system arrangements were examined during the course of this research. One concept, a dual-fluidized-bed FBWHR system, was composed of a hot stage fluidized bed, a cold stage fluidized bed, and a particle circulatory system. The alternative preferred configuration, a hybrid fluidized bed/raining bed FBWHR system, was composed of a hot stage raining bed, a cold stage fluidized bed, and a particle circulatory system.

Laboratory testing of both integrated FBWHR systems was performed prior to building a full-scale unit so that overall and component performance could be optimized. In particular, the following two objectives were addressed:

1. Integrated Performance Development and Documentation - The performance of the integrated systems, which included two beds, a particle lift line, and associated hardware, was tested and documented over the complete operating range of the system.
2. Component Final Development - Extensive testing and development of individual components, such as the L-valves, pneumatic lift line, and fluidized bed are described in Chapter 6. Further testing and development of these and other components is detailed in this chapter.

7.1 DUAL-FLUIDIZED-BED FBWHR SYSTEM

7.1.1 System Description

An isometric drawing of the 1/3-scale laboratory test unit is shown in Figure 7.1. The unit consisted of a hot side fluidized bed (outside dimensions 8 ft x 6 ft x 2 ft) mounted above a cold side fluidized bed (6 ft x 4 ft x 1½ ft), a particle circulating system, a furnace simulator, a combustion air blower, and an exhaust system. The furnace simulator

Figure 7.1 Laboratory Test Facility

produced clean, high-temperature gas that simulated the temperature, flow rate, and pressure of an aluminum smelting furnace flue. The hot gas passed through the upper fluidized bed, heated the particles, and was then discharged to the atmosphere by the exhaust system. In the lower fluidized bed, a combustion air blower pushed ambient air through the lower distributor plate. The air was heated by the hot particles moving across the lower bed.

The particle circulatory system consisted of two L-valves, one feeding particles into the lower bed and one into the lift line. A pneumatic lift line carried the particles from the lower to the upper bed. The particles entering the lift line from the combustion air heat exchanger were hotter than the lift line air; therefore, the lift line heated the combustion air. This preheated air could then be brought into the plenum of the lower fluidized bed where it mixed with the balance of the combustion air from the blower. Associated controls and instrumentation were provided to allow control and documentation of the unit's performance.

7.1.2 Performance Analysis

To determine the preheat capabilities of the FBWHR system, an analysis of the heat transfer process was performed. Each component was modeled individually; the resultant expressions were assembled to yield a set of simultaneous linear equations. The solution of these equations gave the overall system performance. The following is a brief outline of the analysis.

- Fluidized Bed

A fluidized bed could be modeled as a crossflow heat exchanger. Ideally, if the gas flowed up through the heat exchanger without mixing, and the particles flowed across the heat exchanger with mixing only in the vertical direction, the thermal effectiveness of the fluidized bed would be:

$$\epsilon_p = 1 - \exp(-C_g/C_s)$$

where:

- ϵ_p = thermal effectiveness (based on particle flow)
- C_s = thermal capacity of solids (Btu/hr-°F)
- C_g = thermal capacity of gases (Btu/hr-°F)

However, in a fluidized bed the movement of the particles also caused the particles to mix laterally with adjacent material.

A model for the heat transfer was formulated to predict the temperature profile within the bed based on this concept. When the back-mixing of the particles was modeled as an effective thermal conductivity, an energy balance on a differential element within the bed, shown in Figure 7.2, could be written as:

$$\frac{d^2T}{dx^2} - \frac{C_s}{kwh} \frac{dT}{dx} - \frac{C_g}{kwhL} (T - T_g) = 0$$

where:

- T = bed temperature (°F)
- x = distance along bed (ft)
- C_s = thermal capacity of solids (Btu/hr-°F)
- C_g = thermal capacity of gases (Btu/hr-°F)
- T_g = gas inlet temperature (°F)
- k = conductivity of bed (Btu/hr-ft-°F)
- w = width (ft)
- h = height (ft)
- L = length (ft).

This equation was solved using appropriate boundary conditions to give the temperature profile. A family of curves is shown in Figure 7.3 for $C_g/C_s = 1.1$ and various values of the effective conductivity, k_* , $kwh/C_g L$. When mixing was low, k_* was zero and the temperature profile was that of an ideal crossflow heat exchanger. When k_* was

Figure 7.2 Temperature Profile Model

Figure 7.3 Calculated Temperature Profile In Fluidized Bed

high, the temperature profile became uniform. The degree to which the profile flattened out could be altered by varying the geometric proportions of the bed. A long narrow bed would decrease the effect of having a high conductivity.

When evaluating the complicated solution of this differential equation at the solids outlet of the heat exchanger, two simplifications could be made. The first simplification, for large k_* , was to that for an ideal mixed flow heat exchanger:

$$\epsilon_p = 1 - \frac{1}{\frac{C_g}{C_s} + 1}.$$

The second, for small k_* , was when a reduction to that of an ideal crossflow heat exchanger could be made:

$$\epsilon_p = 1 - \exp(-C_g/C_s).$$

Laboratory testing showed that this model accurately determined the temperature profile within the bed. Figure 7.4 is a comparison of theoretical and measured temperature profiles showing excellent agreement. Testing also showed that the thermal conductivity was dependent on the superficial velocity varying from almost 0 at minimum fluidization to over 5000 Btu/hr-ft-°F at 10 ft/sec. This is illustrated in Figure 7.5.

Comparison of Figures 7.3 and 7.4 revealed the similarity between a shallow fluidized bed and an ideal crossflow heat exchanger. This suggested that applying a constant crossflow efficiency factor (F) to the ideal performance model, as in:

$$\epsilon_p = F [1 - \exp(-C_g/C_s)],$$

would yield results in close agreement with the more complex performance equation. For the FBWHR system design conditions, corresponding to k_* of approximately 0.1, F would range between 0.91 and 0.95.

Figure 7.4 Experimental Temperature Profile Compared to Curve Fit of Theory to Data

Figure 7.5 Conductivity of Fluidized Bed as a Function of Gas Velocity

Laboratory results and performance calculations by these two methods differed by less than ± 2 percent, providing validation of this simplified nonideal performance model.

● Lift Line

In the lift line, both the gas and the particles were travelling in the same direction, thus it could be modeled as a parallel flow heat exchanger. The thermodynamic effectiveness of a parallel flow heat exchanger was:

$$\epsilon = \frac{1 - \exp [- N (1 + C)]}{1 + C}$$

where N was the number of transfer units and C was the thermal capacity ratio $C_{\min}/C_{\max}$ ($C_{\min}$ was the smaller of C_g and C_s , $C_{\max}$ was the larger of C_g and C_s). It could be shown that the number of transfer units, N , could be expressed as:

$$N = \frac{6 h t}{\rho C_p d_p}$$

where:

- h = heat transfer coefficient (Btu/hr-ft²-°F)
- t = residence time (hr)
- ρ = particle density (lb/ft³)
- C_p = particle specific heat capacity (Btu/lb-°F)
- d_p = diameter of the particles (ft).

The value of N should be as high as possible. However, analysis shows that there are diminishing returns as N is increased beyond 1.0.

Laboratory results showed N to be between 1.7 and 2.7.

● Integrated Performance

The performance characteristics of the integrated system were calculated by combining mass and energy balances with the above described performance analysis. Data required to solve the resultant simultaneous equations included:

- Thermal capacity of the simulated flue gas
- Simulated flue gas inlet temperature
- Thermal capacity of the combustion air
- Combustion air inlet temperature
- Thermal capacity of the lift line air
- Lift line air inlet temperature
- Thermal capacity of the particles

Assumptions included in the analysis were:

- Steady-state operation of the unit
- No system heat losses present

Calculations resulted in a prediction of the steady-state integrated performance, including the preheated combustion air temperature.

7.2 DUAL-FLUIDIZED-BED FBWHR SYSTEM TEST RESULTS

7.2.1 Integrated System Testing at Design Flow Rates

Baseline testing of the integrated 1/3-scale dual-fluidized-bed FBWHR system laboratory unit was conducted to characterize its operation, allowing a comparison with predicted system performance.

Typical performance of this FBWHR system under design conditions is shown in Figure 7.6. Results were obtained for a particle-to-flue-gas capacity ratio of 1.13, a flue-gas-to-combustion-air capacity ratio of 1.31, and a simulated flue gas inlet temperature of 2030°F. System performance included a preheated combustion air temperature of 1012°F, which was in excess of the design goal of 1000°F. Measured overall system effectiveness was 0.47, in line with predictions.

Figure 7.7 presents the analytically predicted performance of the dual-fluidized-bed system under identical conditions. Good agreement of theory and data was apparent. For example, the predicted outlet combustion air temperature was 1039°F, as compared to the actual 1012°F.

Figure 7.6 Measured Performance of Dual Fluidized Bed FBWHR Laboratory System Under Design Conditions

Figure 7.7 Predicted Performance of Dual Fluidized Bed FBWHR Laboratory System Under Design Conditions

Comparison of the actual and theoretical baseline system performance is facilitated by referring to Table 7.1, a summary of this and two analogous operating points. Examination of Table 7.1 reveals an average difference between the measured and analytically predicted temperatures of less than 20°F. Less than a 3-percent discrepancy existed between the predicted and actual overall system effectiveness. These results proved the preliminary validity of the analytical model chosen. Furthermore, the design goal of 1000°F preheated combustion air temperature was obtained.

7.2.2 Cold Side Bed

The lower, cold stage fluidized bed is diagrammed in Figure 7.8. A combustion air blower pushed ambient air through the lower distributor plate, and the air was heated by the hot particles that were circulated from the upper bed. The particles moved across and exited the lower bed, dropping by gravity into the lift line.

Close agreement between the simplified, nonideal crossflow heat exchanger equation and this cold side fluidized bed data is apparent in Figure 7.9. The results plotted in the figure were taken over the entire operating range of the FBWHR system, and included preheated air outlet temperatures ranging from 653° to 1202°F, and thermal capacity ratios of 0.499 to 2.62. The fact that the equation

$$\epsilon_p = 0.95 [1 - \exp (- C_g / C_s)]$$

described all the cold side bed data proved that the effects of mixing (fluidization velocity) or radiation (temperature) were less than 6 percent. Thus, for a given component geometry, knowledge of the capacity ratio allowed accurate predictions to be made of the fluidized bed's thermal effectiveness.

TABLE 7.1

COMPARISON OF ACTUAL DUAL-FLUIDIZED-BED FBWHR
SYSTEM INTEGRATED PERFORMANCE WITH THEORETICAL
PREDICTIONS (NOT ALLOWING FOR SYSTEM HEAT LOSSES)

	Run A	Run B	Run C
	Measured / Theoretical	Measured / Theoretical	Measured / Theoretical
Preheated Combustion Air	1045°F/1008°F	1012°F/1039°F	996°F/995°F
Particles Leaving Lower Bed	810°F/806°F	801°F/826°F	802°F/780°F
Particles Entering Upper Bed	686°F/689°F	708°F/735°F	681°F/666°F
Simulated Flue Gas Outlet	1130°F/1164°F	1236°F/1214°F	1158°F/1163°F
Particles Leaving Upper Bed	1423°F/1407°F	1403°F/1458°F	1427°F/1416°F
Lift Line Air Outlet	660°F/642°F	673°F/680°F	637°F/621°F
Overall Effectiveness	0.4877/0.4687	0.4706/0.4847	0.4645/0.4639

Average Conditions for Above Runs:

$$C_F = 973 \text{ Btu/hr-}^{\circ}\text{F}$$

$$C_A = 774 \text{ Btu/hr-}^{\circ}\text{F}$$

$$C_L = 223 \text{ Btu/hr-}^{\circ}\text{F}$$

$$C_P = 1126 \text{ Btu/hr-}^{\circ}\text{F}$$

$$\text{Simulated Flue Gas Inlet} = 2026^{\circ}\text{F}$$

$$\text{Unheated Combustion Air} = 110^{\circ}\text{F}$$

$$\text{Lift Line Air Inlet} = 100^{\circ}\text{F}$$

Figure 7.8 Cold Side Fluidized Bed

Figure 7.9 Comparison of Cold Side Fluidized-Bed Data to Analytical Predictions

7.2.3 Hot Side Bed

The upper, hot side fluidized bed is diagrammed in Figure 7.10. The natural-gas-fired furnace simulator produced clean products of combustion in the same temperature range as the flue gases exiting an aluminum smelting furnace. The hot gas passed through the upper fluidized bed, heated the particles, and was then discharged to the atmosphere. The particles were fed from the lift line, passed through the fluidized bed where they were heated, and were then circulated by gravity to the lower fluidized bed.

Figure 7.11 compares the hot side bed data with the nonideal crossflow heat exchanger equation,

$$\epsilon_p = 0.95 [1 - \exp(-C_g/C_s)].$$

The points in this graph were taken over the entire operating range of the FBWHR system, including flue gas outlet temperatures ranging from 642° to 1472°F, and thermal capacity ratios of 0.323 to 2.258. This figure illustrates that the performance under wide-ranging operating conditions could be predicted with the nonideal crossflow heat exchanger model. The magnitude of mixing, radiation, and other effects was less than 7 percent. In conclusion, for a given fluidized bed geometry, accurate predictions of the thermal effectiveness could be made with only the thermal capacity ratio known.

7.2.4 Lift Line

Figure 7.12 shows the layout of the FBWHR system lift line. This pneumatic transport line carried the particles from the lower to the upper bed. Since the particles entering the lift line from the combustion air heat exchanger were hotter than the air, the air was heated as it travelled up the lift line. The two streams exiting the concurrent heat exchanger were separated in a cyclone and fed to the balance of the system.

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PARTICLE INLET

Figure 7.10 Hot Side Fluidized Bed

Figure 7.11 Comparison of Hot Side Fluidized-Bed Data to Analytical Predictions

Figure 7.12 Lift Line

Previous analysis demonstrated that the value of N , the number of transfer units in a parallel flow heat exchanger, was required in order to predict the thermal performance of the lift line. Given N , the thermodynamic effectiveness was predicted with the equation:

$$\epsilon = \frac{1 - \exp [-N(1 + C)]}{1 + C},$$

for any given capacity ratio. The value of N should have been as high as possible, but could not be determined analytically. The expected value of N was predicted to be between 1.0 and 2.7.

Figure 7.13 plots the results of laboratory tests that verified the predicted behavior of the lift line. Lift air outlet temperatures for these tests ranged between 254° and 877°F, and thermal capacity ratios of 1.05 to 8.99 were tested. The majority of the lift line operating points lay between analytical performance curves for N equal to 1.7 and 2.7. Thus, with less than a 15-percent error in calculating the thermal effectiveness, the actual number of transfer units for the laboratory lift line could be set at 2.2. This laboratory-verified value of N lay near the upper end of the expected range of values.

Establishing the actual number of transfer units in the lift line allowed scale-up to a full-sized FBWHR system. Using the expression:

$$N = \frac{6 h t}{\rho C_p d_p},$$

changes in particle properties, system velocities or geometries could be accounted for. The unit performance could then be calculated by allowing for the lift line capacity ratio. Mixing, radiation, and other effects proved to be of a secondary nature and did not factor in the calculations.

Figure 7.13 Comparison of Lift Line Data to Analytical Predictions

7.2.5 Effect of Particle Circulation Rate

The influence of the particle circulation rate on the overall effectiveness of the FBWHR system was studied over a particle thermal capacity range of 327 to 1297 Btu/hr-°F. The data points from the 12 different solids circulation rates were compared to the theoretical predictions in Figure 7.14. The operating parameters during the tests are outlined in Table 7.2.

Validation of the general trend of increasing the overall effectiveness with an increase in particle-to-flue-gas capacity was apparent. As the media flow was increased at a constant gas flow, more heat could be transferred between the hot and the cold fluidized beds. Above particle-to-gas capacity ratios of unity, however, significant increases in particle flow were required to yield a perceptible improvement in effectiveness. For example, increasing the capacity ratio C_s/C_g from 0.92 to 1.09 resulted in only a 1-percent increase in theoretical overall effectiveness, from 0.475 to 0.480. Above a capacity ratio of 1.09 there is a slight loss in effectiveness due to the lift line air losses.

Because of the insensitivity of the overall FBWHR system performance to capacity ratio at high capacity ratios, the remainder of the tests were conducted at high capacity ratios. The use of a high relative solids flow rate minimized the impact of solids flow perturbations and measurement errors on the overall effectiveness measurements. In this manner, more accurate documentation of the integrated performance could be made.

To summarize the particle circulation rate tests:

- Particle-to-gas thermal capacity ratios were tested over a 4-to-1 range;
- The actual laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness;

Figure 7.14 Effect of Particle Circulation Rate on Overall FBWHR System Effectiveness

TABLE 7.2
AVERAGE CONDITIONS DURING
PARTICLE CIRCULATION RATE TESTING

$$C_g = 921 \text{ Btu/hr-}^\circ\text{F}$$

$$C_a = 734 \text{ Btu/hr-}^\circ\text{F}$$

$$C_l = 270 \text{ Btu/hr-}^\circ\text{F}$$

$$C_p = 327 \text{ to } 1297 \text{ Btu/hr-}^\circ\text{F}$$

$$\text{Simulated Flue Gas Inlet} = 2015^\circ\text{F}$$

$$\text{Unheated Combustion Air} = 105^\circ\text{F}$$

$$\text{Lift Line Air Inlet} = 100^\circ\text{F}$$

- Better than 99 percent of the asymptotic value of the thermal performance was attained at a capacity ratio of 1.09; and
- All possible performance tests of the FBWHR system could be made above a particle-to-gas thermal capacity ratio of 1.09, minimizing the system's sensitivity to media flow variations.

7.2.6 Effect of Turndown

Determining the turndown capabilities of the FBWHR system was required prior to designing an industrial-scale system. The higher the flow turndown, the larger the fraction of flue gas energy that could be recovered. A 2:1 flow turndown would allow over 90 percent of the recoverable energy to be captured.

The flow turndown capabilities of the FBWHR system were limited by two parameters, the amount of air used for the lift line and the stability of the fluidized beds. From previous work, it was shown that a 2:1 turndown could be achieved with stable fluidization. Figure 7.15 summarizes these limits.

Nine tests were performed over a 2:1 turndown range to verify the influence of flow turndown on the overall effectiveness of the FBWHR system. A comparison of the laboratory results with the theoretical performance is given in Figure 7.16. The operating parameters during the tests are outlined in Table 7.3.

Visible in both the analytical predictions and in the test results was the lack of sensitivity of the system thermodynamic effectiveness to capacity ratio above ratios of 1. The previously described asymptote for the quantity of heat that could be transferred between the upper and lower beds was approached in this series of tests. Increasing the relative media flow, by reducing the gas flow at constant media flow rate, had little influence on system effectiveness.

Figure 7.15 Limitations of Turndown Due to Air Used in Pneumatic Lift

Figure 7.16 Effect of Flow Turndown on Overall FBWHR System Effectiveness

TABLE 7.3
AVERAGE CONDITIONS DURING
FLOW TURNDOWN TESTING

C_g = 502 to 1039 Btu/hr-°F

C_a = 479 to 904 Btu/hr-°F

C_l = 209 Btu/hr-°F

C_p = 1128 Btu/hr-°F

Simulated Flue Gas Inlet = 2030°F

Unheated Combustion Air = 105°F

Lift Line Air Inlet = 100°F

Comparisons of the actual and predicted overall effectiveness at capacity ratios of 1.11 and 2.26, the extremes of operation, showed good agreement of theory and data, as well as the flat performance of the unit. The measured effectiveness at the low- and high-capacity ratios were 0.463 and 0.451, respectively; the theoretical effectiveness was calculated to be 0.453 and 0.430. The average discrepancy between measured and analytical performance was 3.4 percent, showing excellent agreement between the two. Finally, both the predicted and actual variation in performance due to the 2:1 turndown was less than 3 percent. Major increases in the relative particle flow rate due to gas turndown had little effect on the thermodynamic effectiveness of the system being studied.

To summarize the turndown tests:

- A gas turndown ratio of 2-to-1 was tested;
- The actual laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness; and
- Less than a 3-percent variation in performance was measured over the turndown range.

7.2.7 Effect of System Infiltration/Exfiltration

System infiltration or exfiltration affected the overall performance of an industrial FBWHR system. As an example of exfiltration, the loss of flue gases between the furnace and the FBWHR system would reduce the supply of high-temperature gas. This loss would reduce the availability of heat to the FBWHR system, reducing the system performance. Infiltration, such as the leakage of air into the furnace chamber, would lower the chamber and exhaust temperatures. Most control systems would compensate for these drops by increasing the firing rate. The net effect of this type of infiltration would be an increase in the supply of 2000°F flue gases to the FBWHR system, raising the overall effectiveness of heat recovery. The location of the leak, as well as whether

the leak was into or out of the system, determined the effect of system infiltration/exfiltration. The magnitude and type of system leaks were dependent on the furnace equipment arrangements and operating conditions. The generalized performance model permitted calculation of the effects of infiltration and exfiltration upon performance of the FBWHR system.

Validation of the utility of the FBWHR system analytical model for predicting the effects of system infiltration/exfiltration on performance is apparent in Figure 7.17. The operating parameters during the tests are summarized in Table 7.4. Very good agreement existed between the theory and the data. For the 14 data points taken in the 1.9-to-1 range of combustion air to flue gas capacity ratios, the average discrepancy between predicted and measured performance was 1.2 percent.

As the air-to-gas capacity ratio increased from 0.538 to 1.00, the measured thermodynamic effectiveness dropped from 0.567 to 0.409. The comparable change in theoretical performance was from 0.566 to 0.419. This translated into approximately a 16-percent reduction or gain in performance resulting from a 27-percent increase or decrease in the baseline air-to-gas thermal capacity ratio. This less than 1-to-1 correspondence between changes in performance with changes in capacity ratio implied that the FBWHR system thermodynamic effectiveness was only moderately sensitive to infiltration/exfiltration.

To summarize the system infiltration/exfiltration tests:

- A 1.9-to-1 range of combustion-air-to-flue-gas capacity ratios was tested;
- The measured laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness; and
- Approximately a 16-percent reduction or gain in measured performance resulted from a 27-percent increase or decrease in the baseline air-to-gas thermal capacity ratio.

Figure 7.17 Effect of Infiltration/Exfiltration on Overall FBWHR System Effectiveness

TABLE 7.4
AVERAGE CONDITIONS DURING
INFILTRATION/EXFILTRATION TESTING

$$C_g = 971 \text{ Rtu/hr-}^\circ\text{F}$$

$$C_a = 500 \text{ to } 991 \text{ Btu/hr-}^\circ\text{F}$$

$$C_l = 192 \text{ Btu/hr-}^\circ\text{F}$$

$$C_p = 1127 \text{ Btu/hr-}^\circ\text{F}$$

$$\text{Simulated Flue Gas Inlet} = 2030^\circ\text{F}$$

$$\text{Unheated Combustion Air} = 107^\circ\text{F}$$

$$\text{Lift Line Air Inlet} = 104^\circ\text{F}$$

7.2.8 Effect of Reduced Flue Gas Temperature

There were three effects upon the FBWHR system performance when the flue gas temperature was reduced. The first, which slightly reduced the overall system performance, was simply the lessened availability of energy due to the lowering of all temperatures in the FBWHR system. This effect was allowed for in the analytical model. The second was the potential changes in the system heat transfer due to radiation effects. The third consequence, which would raise the overall effectiveness, was the reduction in heat losses because of the lower average system temperature.

Laboratory results obtained over a simulated flue gas temperature range of 2030° to 1382°F were compared to analytical predictions in Figure 7.18. Overall agreement between the nine data points and the theoretical predictions was quite good, with the average disagreement between the two being 1.7 percent.

Examination of the graph showed that effects attributable to radiation heat transfer or changes in system heat losses were unimportant. The measured 6-percent lowering of effectiveness compared extremely well with the 8-percent decrease predicted by the analytical model. This 2-percent discrepancy between the laboratory and theoretical results was well within the possible system errors. Thus, this verified that radiation contributed little to the high heat transfer rates in a fluidized bed, and that the laboratory FBWHR system was very well insulated.

To summarize the reduced flue gas temperature tests:

- Simulated flue gas temperature of 2030° to 1382°F were tested;
- The measured laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness;

Figure 7.18 Effect of Reduced Flue Gas Temperature on Overall FBWHR System Effectiveness

- Approximately a 7-percent lowering in measured or predicted system performance resulted from a 648°F reduction in flue gas temperature; and
- Variations in effectiveness due to radiation heat transfer effects or system heat loss changes were not observed.

7.2.9 Effect of Turndown and Reduced Flue Gas Temperatures

Actual off-design performance of the integrated system entailed both gas flow turndown and reduced flue gas temperatures. In this off-design condition, the furnace flow rate was reduced by a factor of 2, and the flue gas temperature was reduced to 1400°F. Figure 7.19 illustrates the laboratory results of this operating point, where the preheated combustion air temperature was 653°F, and the gas flow was reduced to 44 percent of the high-fire condition. The analytically predicted results at this operating point are presented in Figure 7.20, where the theoretical combustion air temperature was 612°F.

Comparison of the two sets of results showed fair agreement between the two. The measured component effectiveness for each fluidized bed was approximately 6 percent higher than the comparable theoretical value. This indicated that at these conditions, both fluidized beds were more nearly approaching the performance of ideal cross-flow heat exchangers. This higher performance could be ascribed to the lower fluidizing velocity, due to the turndown in gas mass flow and the lower gas temperatures. As shown in Figure 7.5, lower fluidizing velocities resulted in a decreased bed conductivity. This smaller conductivity resulted in a higher crossflow efficiency factor (F). Figure 7.21 illustrates the beneficial effect on predicted performance of increasing F from 0.95 to 1.0. The difference between predicted and actual combustion air temperatures dropped from 41° to 4°F; the variance between the theoretical and measured effectiveness was decreased from 7.4 to 0.6 percent. Thus F was a function of superficial velocity, varying from 0.95 at high velocities to 1.0 at low velocities.

Figure 7.19 Measured Performance of FBWHR Laboratory System Under Off-Design Conditions

Figure 7.20 Predicted Performance of FBWHR Laboratory System Under Off-Design Conditions

Figure 7.21 Performance of FBWHR Laboratory System Under Off-Design Conditions (Alternately Predicted)

To summarize the turndown and reduced flue gas temperature tests:

- The simulated flue gas temperature was reduced to 1423°F, with the flue gas flow reduced to 44 percent of the high-fire condition, yielding a measured preheated combustion air temperature of 653°F;
- The measured off-design laboratory performance of the FBWHR system was slightly better than the analytically predicted effectiveness; and
- The off-design laboratory data indicated fluidized performance more nearly approaching that of an ideal crossflow heat exchanger, due to the decreased fluidizing velocity and concurrently decreased bed conductivity.

7.2.10 Conclusions

The following conclusions could be drawn from the dual-fluidized-bed laboratory FBWHR system test results:

- The component behavior of the hot and cold side fluidized beds could be accurately described by the simplified nonideal cross-flow heat exchanger model:

$$\epsilon_p = 0.95 [1 - \exp(-C_g/C_s)];$$

- The component performance of the lift line heat exchanger could be accurately predicted with the parallel flow heat exchanger model:

$$\epsilon = \frac{1 - \exp[-N(1+C)]}{1+C},$$

using N equal to 2.2;

- Mixing, radiation, and other effects were of a secondary nature and did not affect the component performance calculations; and

- The actual laboratory performance of the FBWHR system measured over a 4-to-1 range of particle to gas thermal capacities, 2-to-1 gas turndown ratio, 1.9-to-1 range of infiltration/exfiltration tests, 2030° to 1382°F simulated flue gas temperature range, and at an off-design test with both turndown and reduced gas temperature, agreed well with the analytically predicted model.

7.3 HYBRID FLUIDIZED BED/RAINING BED FBWHR SYSTEM

The hybrid fluidized bed/raining bed FBWHR system was composed of a hot stage raining bed, a cold stage fluidized bed, and a particle circulatory system. This preferred design was selected in order to:

- Allow a wide selection of corrosion-related materials of construction;
- Minimize flue gas pressure loss;
- Minimize fouling; and
- Maximize combustion air preheat capabilities.

7.3.1 System Description

An isometric drawing of the 1/3-scale hybrid fluidized bed/raining bed laboratory test unit is shown in Figure 7.22. The unit consisted of a hot side raining bed (outside dimensions 5 ft diameter x 8 ft high) mounted above a cold side fluidized bed (6 ft x 4 ft x 1½ ft), a particle circulating system, a furnace simulator, a combustion air blower, and an exhaust system. The furnace simulator produced clean, high-temperature gas that simulated the temperature, flow rate, and pressure of an aluminum smelting furnace flue. The hot gas passed up through the upper raining bed, heated the falling particles, and was then discharged to the atmosphere by the exhaust system. In the fluidized bed, a combustion air blower pushed ambient air through the distributor plate. The air was heated by the hot particles moving across the lower bed.

Figure 7.22 Laboratory Test Facility
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The particle circulatory system consisted of three L-valves, one feeding particles into the lower bed, one into the lift line, and one into the upper bed. A pneumatic lift line carried the particles from the lower to the upper bed. The particles entering the lift line from the combustion air heat exchanger were hotter than the lift line air, therefore, the lift line heated the combustion air. This preheated air could then be brought into the plenum of the lower fluidized bed where it mixed with the balance of the combustion air from the blower. Associated controls and instrumentation were provided to allow control and documentation of the unit's performance.

7.3.2 Performance Analysis

To determine the preheat capabilities of the FBWHR system, an analysis of the heat transfer process was performed. Each component was modeled individually; the resultant expressions were assembled to yield a set of simultaneous linear equations. The solution of these equations gave the overall system performance. The following is a brief outline of the analysis.

- Fluidized Bed

A fluidized bed could be modeled as a crossflow heat exchanger. This heat transfer model was detailed in Section 7.1.2.

- Raining Bed

In the upper, hot stage raining bed (shown in Figure 7.23) the hot gas passed up through the raining bed, heated the falling particles and was then discharged to the atmosphere. The particles were fed from the lift line, passed down through the raining bed where they were heated, and were then circulated by gravity to the cold stage fluidized bed.

The thermodynamic effectiveness of a counterflow heat exchanger was:

Figure 7.23 Hot Stage Raining Bed

$$\epsilon = \frac{1 - \exp [- N (1 - C)]}{1 - C \exp [- N (1 - C)]}$$

where N was the number of transfer units, and C was the thermal capacity ratio $C_{\min}/C_{\max}$ ($C_{\min}$ was the smaller of C_g and C_s , $C_{\max}$ was the larger of C_g and C_s).

One limiting case of this equation was of particular interest for FBWHR system development: when C was equal to unity. Under this condition, the equation reduced to:

$$\epsilon = \frac{N}{1 + N} \quad (C = 1).$$

It could be shown that the number of transfer units, N , could be expressed as:

$$N = \frac{6 h t}{\rho C_p d_p}$$

where:

- h = heat transfer coefficient (Btu/hr-ft²-°F)
- t = residence time (hr)
- ρ = particle density (lb/ft³)
- C_p = particle specific heat capacity (Btu/lb-°F)
- d_p = diameter of the particles (ft).

The value of N should be as high as possible. However, analysis shows that there are diminishing returns as N is increased beyond 1.0.

It was expected that N would be between 1.5 and 3.0.

● Lift Line

In the lift line, both the gas and the particles travelled in the same direction, thus it could be modeled as a parallel-flow heat exchanger. This heat transfer model was detailed in Section 7.1.2.

- **Integrated Performance**

The performance characteristics of the integrated system were calculated by combining mass and energy balances with the described performance analysis. Data required to solve the resultant simultaneous equations included:

- Thermal capacity of the simulated flue gas;
- Simulated flue gas inlet temperature;
- Thermal capacity of the combustion air;
- Combustion air inlet temperature;
- Thermal capacity of the lift line air;
- Lift line air inlet temperature; and
- Thermal capacity of the particles.

Assumptions included in the analysis were:

- Steady-state operation of the unit; and
- No system heat losses present.

Calculations resulted in a prediction of the steady-state integrated performance, including the preheated combustion air temperature.

7.4 HYBRID FLUIDIZED BED/RAINING BED FBWHR SYSTEM TEST RESULTS

The characterization of the thermal performance of all individual components in the hybrid fluidized bed/raining bed FBWHR laboratory system, as well as the integrated system, is described below. Good agreement between the actual and predicted performance of the three heat exchanger components was observed, providing further confirmation of the validity of the analytical model. The measured effectiveness of the hot stage raining-bed component was, as expected, better than that of a fluidized bed. The thermal performance of the cold stage fluidized bed, lift line heat exchanger, and integrated system was consistent with the tests discussed. A description of the hybrid system test results is given in this section.

7.4.1 Integrated System Testing at Design Flow Rates

Baseline testing of the integrated 1/3-scale hybrid FBWHR laboratory system was completed. These design flow tests were conducted to characterize the operation of the facility, allowing a comparison with predicted system performance.

Typical full-flow performance of the FBWHR system equipped with the initial raining-bed configuration and without recycle of the lift air is shown in Figure 7.24. Results were obtained for a particle-to-flue-gas capacity ratio of 1.01, a flue-gas-to-combustion-air capacity ratio of 1.45, and a simulated flue gas inlet temperature of 2049°F. System performance included a preheated combustion air temperature of 1103°F, which was well in excess of the design goal of 1000°F. Measured overall system effectiveness was 0.51, in line with predictions.

In Figure 7.25, which presents the analytically predicted performance of the system under identical conditions, good agreement of theory and data is apparent. For example, the predicted outlet combustion air temperature was 1132°F, as compared to the actual 1103°F.

Recycling the preheated lift air produced approximately a 100°F increase in combustion air preheat temperature. This performance improvement due to lift air recycle is obvious in Figure 7.26, where the FBWHR laboratory system was operated under similar design flow conditions. The improvement in overall system performance because of recycling the lift air was 6 percentage points, to an overall system effectiveness of 0.57.

Figure 7.27 shows the corresponding analytical predictions. There was a close match between the theoretical and laboratory results, with identical 1219°F air preheat temperatures, and only a 7°F difference between flue gas outlet temperatures.

Figure 7.24 Measured Design Performance Using the First Raining-Bed Configuration in the FBWHR Laboratory System Without Recycle

Figure 7.25 Predicted Design Flow Performance Using the First Raining-Bed Configuration in the FBWHR Laboratory System Without Recycle

Figure 7.26 Measured Design Flow Performance Using the First Raining-Bed Configuration in the FBWHR Laboratory System With Recycle

Figure 7.27 Predicted Design Flow Performance Using the First Raining-Bed Configuration in the FBWHR Laboratory System With Recycle

Comparison of the effects on laboratory system performance because of the recycling of lift air is facilitated by referring to Table 7.5, a summary of these two operating points. Examination of Table 7.5 reveals there was average difference of less than 14°F between the measured and analytically predicted temperatures. Less than a 2-percentage-point discrepancy existed between the predicted and actual overall system effectiveness. These results proved the validity of the analytical model chosen. Furthermore, the design goal of surpassing a 1000°F preheated combustion air temperature was obtained, with the use of lift air recycle producing the highest preheat, 1219°F.

7.4.2 Cold Stage Fluidized Bed Component

In the lower cold stage fluidized bed, diagrammed in Figure 7.28, a combustion air blower pushed ambient air through the lower distributor plate. For improved FBWHR system performance, preheated lift air could be recycled to the inlet of the fluidized bed. With lift air recycle, a concurrent decrease in makeup combustion airflow was required to maintain the desired flow rate. The air was heated by the hot particles that were circulated from the upper bed. The particles moved across and exited the lower bed, dropping by gravity into the lift line.

Cold stage fluidized bed data are shown in Figure 7.29, extracted from the entire operating range of the FBWHR laboratory system. Results from this first set of raining bed tests spanned preheated air outlet temperatures of 413 to 1326°F, and thermal capacity ratios of 0.329 to 3.12. Data were segregated into two classifications, depending on whether the lift line air was recycled to the fluidized bed.

Two theoretical performance lines are also plotted in this figure, using the equation:

$$\epsilon_p = F [1 - \exp (- C_g / C_s)].$$

TABLE 7.5
COMPARISON OF FBWHR LABORATORY
SYSTEM DESIGN FLOW PERFORMANCE
WITH THEORETICAL PREDICTIONS

	Run A	Run B
	Measured / Theoretical	Measured / Theoretical
Preheated Combustion Air	1103/1132°F	1219/1219°F
Particles Leaving Lower Bed	941/931°F	1083/1069°F
Particles Entering Upper Bed	793/795°F	899/891°F
Simulated Flue Gas Outlet	1227/1215°F	1314/1321°F
Particles Leaving Upper Bed	1599/1621°F	1683/1672°F
Lift Line Air Outlet	724/689°F	783/774°F
Overall Effectiveness	0.5078/0.5233°F	0.5679/0.5679
Raining-Bed Baffle Configuration	Initial	Initial
Lift Air Recycle	No	Yes

Average Conditions for Above Runs:

$$C_f = 1196 \text{ Btu/hr-}^\circ\text{F}$$

$$C_a = 784 \text{ Btu/hr-}^\circ\text{F} \text{ (Run A) or } 507 \text{ Btu/hr-}^\circ\text{F} \text{ (Run B)}$$

$$C_l = 310 \text{ Btu/hr-}^\circ\text{F}$$

$$C_p = 1152 \text{ Btu/hr-}^\circ\text{F}$$

Simulated Flue Gas Inlet = 2044°F

Unheated Combustion Air = 130°F

Lift Line Air Inlet = 150°F

Figure 7.28 Cold Stage Fluidized Bed

Figure 7.29 Comparison of Cold Stage Fluidized-Bed Data to Analytical Predictions

These two lines corresponded to a cold air inlet (no lift air recycle, $F = 0.93$), and a hot air inlet (lift air recycle, $F = 0.91$). The grouping of data around these two lines indicates that there was a 2-percent effect due to variable air inlet temperature (gas velocity) on fluidized bed thermal effectiveness, with F ranging from 0.93 to 0.91. The higher gas velocities increased bed mixing and bed conductivity, decreasing both F and performance. The remainder of the scatter in the data, averaging ± 2 percent, could be attributed to radiation effects (component temperature) and experimental error. Thus, for a given component geometry, knowledge of the capacity ratio (and to a lesser degree, the presence of lift air recycle) allowed accurate predictions to be made of the thermal effectiveness of the fluidized bed.

7.4.3 Hot Stage Raining Bed Component

The upper, hot stage raining bed is shown in Figure 7.30. The natural-gas-fired furnace simulator produced clean products of combustion ranging in temperature from 1400 to 2000°F, the same range as most of the flue gases exiting an aluminum smelting furnace. The hot gas passed up through the raining bed, heated the falling particles, and was then discharged to the atmosphere. The particles were fed from the lift line, passed down through the raining bed where they were heated, and were then circulated by gravity to the cold stage fluidized bed.

The raining bed configuration was designed for:

- Uniform initial distribution of both flue gas and particles;
- Minimum bypassing of gas or particles;
- Maximum retention time of particles;
- High heat transfer rates;
- Low flue gas pressure loss; and
- Ease of assembly and disassembly.

Figure 7.30 Hot Stage Raining Bed

The resultant raining bed configuration successfully addressed the above concerns. Flow visualization tests confirmed the anticipated particle flow characteristics.

Figure 7.31, a graph of the hot side raining bed performance, presents thermal effectiveness data for a 0.576-to-2.62 range of solids-to-gas thermal capacity ratios, including a 586-to-1334°F range of flue gas outlet temperatures. (The number of transfer units was obtained for each capacity ratio test by substituting ϵ_g , C_g and C_s for each test into the theoretical counterflow heat exchanger performance equation; solving the resultant expression yielded N for those data.) Specific results at the design solids-to-gas thermal capacity ratio of 1.0 included a component thermal effectiveness of 0.654, and a flue gas pressure loss of 0.30 in. w.c.

The theoretical performance prediction is also shown in this figure, using the counterflow heat exchanger performance equation,

$$\epsilon_g = \frac{C_{\min}}{C_g} \frac{1 - \exp[-N(1-C)]}{1 - C \exp[-N(1-C)]}$$

with an average number of transfer units for all tests of $N = 1.967$. There was good agreement between data and theory, with an average discrepancy between actual and theoretical effectiveness of 4.4 percent, and a maximum error of 13.8 percent. In other words, over the wide ranging test conditions, thermal performance of a raining-bed heat exchanger could be described by the counterflow heat exchanger model, using $N = 1.967$ (for the first raining bed configuration).

7.4.4 Lift Line Component

Figure 7.32 shows the layout of the FBWHR system lift line. This pneumatic transport line carried the particles from the lower to the upper bed. Since the particles entering the lift line from the fluidized-bed heat exchanger were hotter than the air, the air was

Figure 7.31 Comparison of Hot Stage Raining-Bed Data to Analytical Predictions

Figure 7.32 Lift Line

heated as it travelled up the lift line. The two streams exiting the concurrent heat exchanger were separated in a cyclone and fed to the balance of the system. The particles reentered the top of the raining bed. The hot air could be either vented to the atmosphere, or recycled to the cold stage fluidized bed.

Previous analysis required the value of N , the number of transfer units in a parallel flow heat exchanger, to predict the thermal performance of the lift line. Given N , the thermodynamic effectiveness was predicted with the equation:

$$\epsilon = \frac{1 - \exp [- N (1 + C)]}{1 + C},$$

for any given capacity ratio. To produce the highest heat exchanger effectiveness, the value of N should be as high as possible.

Figure 7.33 plots the results of the laboratory tests that verified the predicted behavior of the lift line. For these tests, coupled with the first raining-bed configuration, lift air outlet temperatures ranged between 172 and 895°F, and thermal capacity ratios of 0.879 to 5.17 were tested. (The number of transfer units was obtained for each capacity ratio test by substituting ϵ_ℓ , C_ℓ and C_s for each test into the theoretical counterflow heat exchanger performance equation; solving the resultant expression yielded N for those data.) The average value of N for the 24 lift line operating points was $N = 1.6$, with an average error of 3.0 percent. Additionally, with less than an 11-percent error in calculating the thermal effectiveness, the actual number of transfer units for the laboratory lift line could be set at 1.6.

Establishing the actual number of transfer units in the lift line allowed scale-up to a full-sized FBWHR system. By using the expression:

$$N = \frac{6 h t}{\rho C_p d_p}$$

Figure 7.33 Comparison of Lift Line Data to Analytical Predictions

changes in particle properties, system velocities or geometries could be taken into account. The unit performance could then be calculated by allowing for the lift line capacity ratio. Mixing, radiation, and other effects proved to be of a secondary nature and did not factor in the calculations.

7.4.5 Effect of Particle Circulation Rate on the Integrated FBWHR System

The influence of the particle circulation rate on the overall effectiveness of the FBWHR system was studied over a range of solids-to-gas thermal capacity ratios of 0.576 to 2.41. The data points from the seven different solids circulation rates were compared with the theoretical predictions in Figure 7.34, which also includes the data point from the design air-to-gas capacity ratio and the corresponding analytical performance curve. The operating parameters that existed during the tests are outlined in Table 7.6.

The tests validated the general trend of increasing the overall effectiveness by means of an increase in particle-to-flue-gas capacity. As the media flow was increased at a constant gas flow, more heat could be transferred between the hot and the cold beds. Above particle-to-gas capacity ratios of unity, however, significant increases in particle flow were required to yield a perceptible improvement in effectiveness. For example, at the design air-to-gas capacity ratio, increasing the capacity ratio C_s/C_g from 0.95 to 1.10 resulted in only a 1-percent increase in theoretical overall effectiveness, from 0.485 to 0.492. Negligible effectiveness increases occurred above a capacity ratio of 1.10.

To summarize the particle circulation rate tests:

- Particle-to-gas thermal capacity ratios were tested over a 4.2-to-1 range;

Figure 7.34 Effect of Particle Circulation Rate on Overall FBWHR System Effectiveness (Without Recycle)

TABLE 7.6
AVERAGE CONDITIONS DURING
PARTICLE CIRCULATION RATE TESTING

$$C_g = 506 \text{ Btu/hr-}^\circ\text{F}$$

$$C_a = 779 \text{ Btu/hr-}^\circ\text{F}$$

$$C_\ell = 293 \text{ Btu/hr-}^\circ\text{F}$$

$$C_p = 289 \text{ to } 1195 \text{ Btu/hr-}^\circ\text{F}$$

Simulated Flue Gas Inlet = 1410°F

Unheated Combustion Air = 107°F

Lift Line Air Inlet = 133°F

First Raining-Bed Configuration

No Lift Air Recycle

- The actual laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness; and
- Better than 99 percent of the asymptotic value of the thermal performance was attained at a capacity ratio of 1.10.

7.4.6 Effect of Turndown

Determining the turndown capabilities of the FBWHR system was required before an industrial-scale system could be designed. The higher the flow turndown, the larger the fraction of flue gas energy that could be recovered. For typical operations, a 2:1 flow turndown would allow over 90 percent of the recoverable energy to be captured.

The flow turndown capabilities of the FBWHR system were limited by two parameters, the amount of air used for the lift line and the stability of the fluidized bed. From previous work, it was shown that a 2:1 turndown could be achieved with stable fluidization.

Five tests were performed over a 1.7:1 turndown range to verify the influence of flow turndown on the overall effectiveness of the FBWHR system. Figure 7.35 provides a comparison of the laboratory results with the theoretical performance, and Table 7.7 outlines the operating parameters during the tests.

Apparent in both the analytical predictions and in the test results was the lack of sensitivity of the system thermodynamic effectiveness to capacity ratio above ratios of 1. The previously described asymptote for the quantity of heat that could be transferred between the upper and lower beds was approached in this series of tests. Increasing the relative media flow rate, by reducing the gas flow at constant media flow rate, had little influence on system effectiveness.

Comparison of the actual and predicted overall effectiveness at capacity ratios of 0.920 and 1.53, the extremes of operation, showed good agreement of theory and data, as well as the flat performance

Figure 7.35 Effect of Flow Turndown on Overall System Effectiveness (With Recycle)

TABLE 7.7
TEST RANGE FOR OFF-DESIGN OPERATION

$$C_g = 1259 \text{ to } 761 \text{ Btu/hr-}^{\circ}\text{F}$$

$$C_a = 507 \text{ to } 0 \text{ Btu/hr-}^{\circ}\text{F}$$

$$C_{\lambda} = 365 \text{ Btu/hr-}^{\circ}\text{F}$$

$$C_p = 1159 \text{ Btu/hr-}^{\circ}\text{F}$$

Simulated Flue Gas Inlet = 2009°F

Unheated Combustion Air = 117°F

Lift Line Air Inlet = 139°F

First Raining-Bed Configuration

Lift Air Recycle

of the unit. The measured effectiveness at the low- and high-capacity ratios was 0.568 and 0.589, respectively. The theoretical effectiveness was calculated to be 0.568 and 0.548. The average discrepancy between measured and analytical performance was 2.6 percent, showing excellent agreement between the two. Finally, both the predicted and actual variation in performance due to the 1.7:1 turndown was less than 4 percent. Major increases in the relative particle flow rate due to gas turndown had little effect on the thermodynamic effectiveness of the system being studied.

To summarize the turndown tests:

- A gas turndown ratio of 1.7-to-1 was tested;
- The actual laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness; and
- Less than a 4-percent variation in performance was measured over the turndown range.

7.4.7 Effect of System Infiltration/Exfiltration

System infiltration or exfiltration affected the overall performance of the FBWHR system. As an example of exfiltration, the loss of flue gases between the furnace and the FBWHR system would reduce the supply of high-temperature gas. This loss would reduce the availability of heat to the FBWHR system, reducing the system performance. Infiltration, such as the leakage of air into the furnace chamber, would lower the chamber and exhaust temperatures. Most control systems would compensate for these losses by increasing the firing rate. The net effect of this type of infiltration would be an increase in the supply of 2000°F flue gases to the FBWHR system, raising the overall effectiveness of heat recovery. The location of the leak, as well as whether the leak was into or out of the system, determined the effect of system infiltration/exfiltration. The magnitude and type of system

leaks were dependent on the furnace equipment arrangements and operating conditions. The generalized performance model permitted calculation of the effects of infiltration and exfiltration upon performance of the FBWHR system.

Figure 7.36 shows the utility of the FBWHR system analytical model for predicting the effects of system infiltration/exfiltration on performance. The operating parameters that existed during the tests are summarized in Table 7.8. There was good agreement between the theory and the data. For the five data points taken in the 1.8-to-1 range of combustion air to flue gas capacity ratios, the average discrepancy between predicted and measured performance was 2.9 percent.

As the air-to-gas capacity ratio increased from 0.568 to 1.00, the measured thermodynamic effectiveness dropped from 0.645 to 0.463. The comparable change in theoretical performance was from 0.623 to 0.467. This translated into approximately a 17-percent reduction or gain in performance resulting from a 27-percent increase or decrease in the baseline air-to-gas thermal capacity ratio. This less than 1-to-1 correspondence between changes in performance with changes in capacity ratio implied that the FBWHR system thermodynamic effectiveness was only moderately sensitive to infiltration/exfiltration.

To summarize the system infiltration/exfiltration tests:

- A 1.8-to-1 range of combustion-air-to-flue-gas capacity ratios was tested;
- The measured laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness; and
- Approximately a 17-percent reduction or gain in measured performance resulted from a 27-percent increase or decrease in the baseline air-to-gas thermal capacity ratio.

Figure 7.36 Effect of Infiltration/Exfiltration on Overall FBWHR System Effectiveness (With Recycle)

TABLE 7.8
AVERAGE CONDITIONS DURING
INFILTRATION/EXFILTRATION TESTING

$$C_g = 1065 \text{ Btu/hr-}^\circ\text{F}$$

$$C_a = 296 \text{ to } 784 \text{ Btu/hr-}^\circ\text{F}$$

$$C_\ell = 263 \text{ Btu/hr-}^\circ\text{F}$$

$$C_p = 1148 \text{ Btu/hr-}^\circ\text{F}$$

Simulated Flue Gas Inlet = 2006°F

Unheated Combustion Air = 120°F

Lift Line Air Inlet = 142°F

First Raining-Bed Configuration

Lift Air Recycle

7.4.8 Effect of Reduced Flue Gas Temperature

There were three effects upon the FBWHR system performance when the flue gas temperature was reduced. The first, which slightly reduced the overall system performance, was simply the lessened availability of energy due to the lowering of all temperatures in the FBWHR system. This effect was allowed for in the analytical model. The second was the potential change in the system heat transfer due to radiation effects. The third consequence, which would raise the overall effectiveness, was the reduction in heat losses because of the lower average system temperature.

Figure 7.37 shows a comparison of the laboratory results obtained over a simulated flue gas temperature range of 2039 to 1407°F with analytical predictions, and Table 7.9 summarizes the operating parameters that existed during the tests. Overall agreement between the six data points and the theoretical predictions was quite good, with the average disagreement between the two being 2.2 percent.

Examination of the graph shows that effects attributable to radiation heat transfer or changes in system heat losses were unimportant. The measured 5-percent lowering of effectiveness compared extremely well with the 3-percent decrease predicted by the analytical model. This 2-percentage-point discrepancy between the laboratory and theoretical results was well within the possible system errors. Thus, this verified that radiation contributed little to the high heat transfer rates in a fluidized bed or a raining bed, and that the laboratory FBWHR system was very well insulated.

To summarize the reduced flue gas temperature tests:

- Simulated flue gas temperatures of 2039 to 1407°F were tested;
- The measured laboratory performance of the FBWHR system closely matched the analytically predicted effectiveness;

Figure 7.37 Effect of Reduced Flue Gas Temperature on Overall FBWHR System Effectiveness (With Recycle)

TABLE 7.9
AVERAGE CONDITIONS DURING
REDUCED FLUE GAS TEMPERATURE TESTING

$$C_g = 1100 \text{ Btu/hr-}^\circ\text{F}$$

$$C_a = 470 \text{ Btu/hr-}^\circ\text{F}$$

$$C_l = 301 \text{ Btu/hr-}^\circ\text{F}$$

$$C_p = 1124 \text{ Btu/hr-}^\circ\text{F}$$

Simulated Flue Gas Inlet = 1407° to 2039°F

Unheated Combustion Air = 129°F

Lift Air Inlet = 150°F

First Raining-Bed Configuration

Lift Air Recycle

- Approximately a 4-percent lowering in measured or predicted system performance resulted from a 632°F reduction in flue gas temperature; and
- Variations in effectiveness due to radiation heat transfer effects or system heat loss changes were not observed.

7.4.9 Effect of Turndown and Reduced Flue Gas Temperature

Actual off-design performance of the integrated system entailed both gas flow turndown and reduced flue gas temperatures. In this off-design condition, the furnace flow rate was reduced by a factor of 2, and the flue gas temperature was reduced to 1400°F. Figure 7.38 illustrates the laboratory results of this operating point, with lift air recycle, where the preheated combustion air temperature was 827°F and the flue gas flow was reduced to 37 percent of the high-fire condition. Figure 7.39 presents the analytically predicted results at this operating point, where the theoretical combustion air temperature was 766°F.

To summarize the turndown and reduced flow gas temperature tests:

- The simulated flue gas temperature was reduced to 1433°F, with the flue gas flow reduced to 37 percent of the high-fire condition, yielding a measured preheated combustion air temperature of 827°F; and
- The measured off-design laboratory performance of the FBWHR system was slightly better than the analytically predicted effectiveness.

Figure 7.38 Off-Design Performance Using the First Raining-Bed Configuration in the FBWHR Laboratory System (With Recycle)

Figure 7.39 Predicted Off-Design Performance Using the First Raining-Bed Configuration in the FBWHR Laboratory System (With Recycle)

7.4.10 Conclusions

Based on the test results from the hybrid FBWHR laboratory system using an initial raining bed configuration, the following conclusions could be made:

- The component behavior of the cold stage fluidized bed could be accurately described by the simplified nonideal crossflow heat exchanger model:

$$\epsilon_p = F [1 - \exp (- C_g / C_s)],$$

where $F = 0.93$ for operation without recycle of preheated lift air, and $F = 0.91$ when lift air is recycled;

- The component effectiveness of the hot stage raining bed could be accurately predicted by the counterflow heat exchanger model:

$$\epsilon = \frac{1 - \exp [- N (1 - C)]}{1 - C \exp [- N (1 - C)]},$$

using $N = 1.967$ for operation with the first raining-bed configuration;

- The component performance of the lift line heat exchanger could be accurately predicted with the parallel flow heat exchanger model:

$$\epsilon = \frac{1 - \exp [- N (1 + C)]}{1 + C},$$

using N equal to 1.6;

- Mixing, radiation, and other effects were of a secondary nature; and

- The actual laboratory performance of the FBWHR system measured over a 4.2-to-1 range of particle-to-gas thermal capacities 1.7-to-1 gas turndown ratio, 1.8-to-1 range of infiltration/exfiltration tests, 2039 to 1407°F simulated flue gas temperature range, and at off-design tests with both turndown and reduced gas temperature, agreed well with the analytically predicted model.

7.5 HYBRID FLUIDIZED BED/RAINING BED FBWHR SYSTEM TEST RESULTS USING AN ALTERNATE RAINING-BED CONFIGURATION

Earlier work, discussed in Section 7.4, resulted in a preliminary hybrid fluidized bed/raining bed FBWHR laboratory system whose performance exceeded design predictions. In order to further characterize the FBWHR system concept, as well as gather scale-up data, a second raining-bed configuration was built and tested. The design changes incorporated in the second configuration included opening up the area for gas flow while simultaneously increasing the particle residence time. These two changes were planned to allow an increase in flue gas throughout (and heat exchanger duty), while maintaining system pre-heat capabilities. This section contains the test results that verified the predicted 30-percent increase in the number of transfer units, N , in the raining bed as well as performance comparisons of the FBWHR system with both raining-bed configurations.

7.5.1 Integrated System Testing at Design Flow Rates

Baseline test results of the integrated 1/3-scale FBWHR laboratory system equipped with the first configuration were discussed in Section 7.4.1. These design flow tests were conducted to characterize the operation of the facility, allowing a comparison with predicted system performance. Typical full-flow performance of the FBWHR system equipped with the first raining-bed configuration and with recycle of the lift air included a preheated combustion air temperature of 1219°F,

which was well in excess of the design goal of 1000°F. Measured overall system effectiveness was 0.57, in line with predictions. There was a close match between the theoretical and laboratory results, with identical 1219°F air preheat temperatures, and only a 7°F difference between flue gas outlet temperatures.

Figure 7.40 shows the comparable performance of the FBWHR system equipped with the alternative raining-bed configuration. The combustion air preheat temperature increased 30°F, to 1249°F; the flue gas outlet temperature dropped 100°F, to 1214°F. Measured overall system effectiveness increased to 0.60. Good agreement with analytical predictions is shown in Figure 7.41, with the theoretical system effectiveness equal to 0.59.

Table 7.10 compares the theoretical predictions with the baseline laboratory system performance. Examination of this table reveals an average difference between the measured and analytically predicted temperatures of less than 15°F. Less than a 2-percentage-point discrepancy existed between the predicted and actual overall system effectiveness. These results proved the validity of the analytical model chosen. Furthermore, the design goal of surpassing a 1000°F preheated combustion air temperature was obtained, with the second raining bed configuration producing the highest preheat, 1249°F.

7.5.2 Effect of Turndown and Reduced Flue Gas Temperature

Results given in Section 7.4.9 indicated that off-design system operation using the preliminary raining bed configuration, 37 percent of the design flue gas flow, and a flue gas temperature of 1433°F, produced 827°F preheated combustion air temperature. This showed that substantial heat could be transferred during off-design operation.

Figure 7.42 shows comparable system operation using the second configuration. A 27°F increase in measured combustion air preheat, to 854°F, was produced using 1449°F flue gas. The utility of the

Figure 7.40 Design Flow Performance Using the Second Raining-Bed Configuration in the FRWHR Laboratory System (With Recycle)

Figure 7.41 Predicted Design Flow Performance Using the Second Raining-Bed Configuration in the FBWHR Laboratory System (With Recycle)

TABLE 7.10
COMPARISON OF FBWHR LABORATORY
SYSTEM DESIGN FLOW PERFORMANCE
WITH THEORETICAL PREDICTIONS

	Run B	Run C
	Measured / Theoretical	Measured / Theoretical
Preheated Combustion Air	1219/1219°F	1249/1221°F
Particles Leaving Lower Bed	1083/1069°F	1089/1066°F
Particles Entering Upper Bed	899/891°F	949/930°F
Simulated Flue Gas Outlet	1314/1321°F	1214/1182°F
Particles Leaving Upper Bed	1683/1672°F	1685/1700°F
Lift Line Air Outlet	783/774°F	796/791°F
Overall Effectiveness	0.5679/0.5679	0.6003/0.5859
Raining-Bed Configuration	First	Second
Lift Air Recycle	Yes	Yes

Average Conditions for Above Runs:

$$C_f = 1169 \text{ Btu/hr-}^\circ\text{F}$$

$$C_a = 536 \text{ Btu/hr-}^\circ\text{F}$$

$$C_\ell = 284 \text{ Btu/hr-}^\circ\text{F}$$

$$C_p = 1160 \text{ Btu/hr-}^\circ\text{F}$$

Simulated Flue Gas Inlet = 2025°F

Unheated Combustion Air = 114°F

Lift Line Air Inlet = 135°F

Figure 7.42 Off-Design Performance Using the Second Raining-Bed Configuration in the FBWHR Laboratory System (With Recycle)

analytical model is shown in Figure 7.43, where a 24°F increase in theoretical combustion air temperature resulted. In other words, use of the second raining bed configuration improved the measured overall system effectiveness from 0.53 to 0.55, and increased the predicted effectiveness from 0.49 to 0.50. Clearly, the design changes incorporated in the second raining bed configuration improved overall performance over the entire FBWHR system operating range.

7.5.3 Cold Stage Fluidized Bed Component

Results given in Section 7.4.2 indicated that fluidized bed component performance was dependent on the degree of lift air recycle. Consequently, all tests using the second raining bed configuration were conducted with recycle of lift air. Figure 7.44 shows a comparison of data for fluidized bed performance versus capacity ratio extracted from the two series of raining bed tests. Both sets of data follow the same functionality of increasing performance with increases in solids-to-air capacity ratio.

The theoretical performance line is also plotted in this figure, using the equation:

$$\epsilon_p = F [1 - \exp (- C_a / C_s)].$$

Analysis of the fluidized bed data from the second raining bed tests yielded the same functionality as in the first tests, with $F = 0.91$ for both series of experiments. This indicated an identical, repeatable effect on fluidized bed performance due to recycling lift air. The remainder of scatter in the data, averaging ± 2 percent, could be attributed to radiation effects and experimental error. Thus, for a given component geometry, knowledge of the capacity ratio (and to a lesser degree, the presence of lift air recycle) allowed accurate predictions to be made of the thermal effectiveness of the fluidized bed.

Figure 7.43 Predicted Off-Design Performance Using the Second Raining-Bed Configuration in the FBWHR Laboratory System (With Recycle)

Figure 7.44 Comparison of Cold Stage Fluidized-Bed Data to Analytical Predictions

7.5.4 Hot Stage Raining Bed Component

The results given in Section 7.4.3 indicated that raining bed performance using the first raining bed configuration was described by a counterflow heat exchanger model, with the number of transfer units, N , equal to 1.967. Specific results at the design point included a component thermal effectiveness of 0.654, and a flue gas pressure loss of 0.30 in w.c.

Design changes incorporated in the second raining bed configuration allowed an increase in flue gas throughput and heat exchanger duty, while maintaining system preheat capabilities.

Figure 7.45 presents a comparison of the hot stage raining bed component performance with the two configurations. There was marked improvement in component performance, which could be attributed to the second raining bed configuration. Over the 0.402-to-3.65 range of thermal capacity ratios, the thermal effectiveness at every operating point using the second configuration was better than with the first configuration.

Theoretical performance lines for the two sets of tests are also plotted in this figure, using the equation:

$$\epsilon_g = \frac{C_{\min}}{C_g} \frac{1 - \exp[-N(1-C)]}{1 - C \exp[-N(1-C)]}$$

Analysis of the raining bed data from the second tests indicated that N increased to 2.960, representing a 50-percent increase from the result of the first test, N equal to 1.967. This verified the 30-percent improvement of N predicted for the second raining bed configuration, and demonstrated an adequate knowledge of the heat transfer mechanism in the raining bed.

The raining bed design changes were predicted to increase component thermal performance at a constant flow rate, as well as to allow

Figure 7.45 Comparison of Hot Stage Raining-Bed Data to Analytical Predictions

constant thermal performance at increased flue gas throughput. In order to accommodate an increase in gas throughput and continue to allow a natural draft exhaust system, the area for gas flow was increased. The resultant decrease in measured design point flue gas pressure loss was from 0.30 in. w.c. to 0.12 in. w.c. This indicated that a 58-percent increase in flue gas throughput with the second configuration would produce the same pressure loss as design flow with the first configuration. The potential 58-percent increase in raining bed throughput closely matched the 50-percent increase in heat exchanger N, demonstrating a theoretical increase in heat exchanger duty with constant preheat capabilities.

7.5.5 Lift Line Component

Results given in Section 7.4.4 indicated that lift line heat exchanger performance was described by a parallel flow heat exchanger model, with the number of transfer units, N, equal to 1.6. Figure 7.46 shows a comparison of data for lift line thermal performance versus capacity ratio extracted from the two series of raining-bed tests. Both sets of data followed the identical functionality of increasing performance with increases in solids-to-air capacity ratio, over the range of 0.879 to 6.56 in capacity ratio.

The theoretical performance line is also plotted in this figure, using the equation:

$$\epsilon_{\ell} = \frac{C_{\min}}{C_{\ell}} \frac{1 - \exp [-N (1 + C)]}{1 + C}.$$

Analysis of the lift line data from the second raining bed configuration yielded the same functionality as in the first tests, with N = 1.6 for both series of experiments. This indicated identical, repeatable lift line performance. The average error between data and theory for these 35 operating points was 3.4 percent; the maximum error was 22 percent.

Figure 7.46 Comparison of Lift Line Data to Analytical Predictions

Establishing the actual number of transfer units in the lift line allowed scale-up to a full-sized FBWHR system. Using the expression:

$$N = \frac{6 h t}{\rho C_p d_p},$$

changes in particle properties, system velocities or geometries could be taken into account. The unit performance could then be calculated by allowing for the lift line capacity ratio. Mixing, radiation, and other effects proved to be of a secondary nature and did not factor in the calculations.

7.5.6 Conclusions

The following conclusions could be drawn from the current laboratory test results:

- The component behavior of the cold stage fluidized bed could be accurately described by the simplified nonideal crossflow heat exchanger model:

$$\epsilon_p = F [1 - \exp (- C_g / C_s)],$$

where $F = 0.93$ for operation without recycle of preheated lift air, and $F = 0.91$ when lift air was recycled;

- The component effectiveness of the hot stage raining bed could be accurately predicted by the counterflow heat exchanger model:

$$\epsilon = \frac{1 - \exp [- N (1 - C)]}{1 - C \exp [- N (1 - C)]},$$

using $N = 1.967$ for operation with the first raining bed configuration, and $N = 2.960$ for the second configuration;

- The component performance of the lift line heat exchanger could be accurately predicted with the parallel flow heat exchanger model:

$$\epsilon = \frac{1 - \exp [- N (1 + C)]}{1 + C},$$

using N equal to 1.6;

- Mixing, radiation, and other effects were of a secondary nature; and
- The actual laboratory performance of the FBWHR system measured over a 0.402-to-3.65 range of particle-to-gas thermal capacity ratios, 0.603-to-3.21 range of particle to combustion air thermal capacity ratios, 1.60-to-6.45 range of particle to lift air thermal capacity ratios, 2030° to 1378°F simulated flue gas temperature range, and at off-design tests with both turndown and reduced gas temperature, agreed well with the analytically predicted model.

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8. FULL-SCALE FBWHR SYSTEM DESIGN

The full-scale FBWHR system was composed of a hot stage raining bed, a cold stage fluidized bed, and a particle circulatory system. This hybrid design was selected in order to:

- Allow a wide selection of corrosion-resistant materials of construction;
- Minimize flue gas pressure loss;
- Minimize fouling; and
- Maximize combustion air preheat capabilities.

Individual components, such as the L-valves, pneumatic lift line, hot stage raining bed, and cold stage fluidized bed, underwent extensive testing and development in the 1/3-scale laboratory FBWHR system. The design task of scaling up to a full-scale FBWHR system was greatly simplified by relying heavily on these results. These results are further described in References 10-17. This factor of three scaleup to a full-scale FBWHR system, based on capacity, is described in this chapter.

8.1 SYSTEM DESCRIPTION

The full-scale FBWHR system is shown in Figures 8.1 and 8.2. The unit consisted of a hot stage raining bed mounted above a cold stage fluidized bed, a particle circulating system, a combustion air blower, and the existing open hearth aluminum smelting furnace system. The FBWHRs/furnace integration included retrofitting two preheated combustion air burners to the furnace and making provisions to use the existing stack only as a bypass. The furnace exhaust gas passed up through the upper raining bed, heated the falling particles, and was then discharged to the atmosphere by a new stack. In the fluidized bed, a new combustion air blower pushed ambient air through the distributor plate. This air was heated by the hot particles moving across the lower bed.

Figure 8.1 Fluidized-Bed Waste-Heat Recovery System

Figure 8.2 Perspective View of FBWHR System

The particle circulatory system consisted of three L-valves, one feeding particles into the lower fluidized bed, one into the lift line, and one into the upper raining bed. A pneumatic lift line carried the particles from the lower to the upper bed. The particles entering the lift line from the combustion air heat exchanger were hotter than the lift line air; therefore, the lift line heated the lift air. This preheated air was then brought into the plenum of the lower fluidized bed where it mixed with the balance of the combustion air from the blower. Associated controls and instrumentation were provided to allow control and documentation of the unit's performance.

8.2 PERFORMANCE ANALYSIS

To design the preheat capabilities of the FBWHR system, a heat transfer analysis was performed. Each component was modeled individually and the resulting expressions assembled to yield a set of simultaneous equations. The solution of these equations gave the overall system performance. Following is a brief outline of the analysis.

8.2.1 Cold Stage Fluidized Bed Component

In the lower cold stage fluidized bed, diagrammed in Figure 8.3, a combustion air blower pushed ambient air through the lower distributor plate. For improved FBWHR system performance, preheated lift air was recycled to the inlet of the fluidized bed. With lift air recycle, a concurrent decrease in makeup combustion airflow was required to maintain the desired flow rate. The air was heated by the hot particles that were circulated from the upper bed. The particles moved across and exited the lower bed, dropping by gravity into the lift line.

A fluidized bed could be modeled as a crossflow heat exchanger. Ideally, if the gas flowed up through the heat exchanger without mixing, and the particles flowed across the heat exchanger with mixing only in the vertical direction, the thermal effectiveness of the fluidized bed would be:

Figure 8.3 Cold Stage Fluidized Bed

$$\epsilon_p = 1 - \exp(-C_g/C_s)$$

where:

ϵ_p = thermal effectiveness (based on particle flow)

C_s = thermal capacity of solids (Btu/hr-°F)

C_g = thermal capacity of gases (Btu/hr-°F)

However, in a fluidized bed the movement of the particles also caused the particles to mix laterally with adjacent material.

A model for the heat transfer was formulated to predict the temperature profile within the bed based on this concept. When the back-mixing of the particles was modeled as an effective thermal conductivity, an energy balance on a differential element within the bed, shown in Figure 8.4, could be written as:

$$\frac{d^2T}{dx^2} - \frac{C_s}{kwh} \frac{dT}{dx} - \frac{C_g}{kwhL} (T - T_g) = 0$$

where:

T = bed temperature (°F)

x = distance along bed (ft)

C_s = thermal capacity of solids (Btu/hr-°F)

C_g = thermal capacity of gases (Btu/hr-°F)

T_g = gas inlet temperature (°F)

k = conductivity of bed (Btu/hr-ft-°F)

w = width (ft)

h = height (ft)

L = length (ft).

This equation was solved using appropriate boundary conditions to give the temperature profile. A family of curves is shown in Figure 8.5 for $C_g/C_s = 1.1$ and various values of the effective conductivity, k_* , kwh/C_sL . When mixing was low, k_* was zero and the temperature profile was that of an ideal crossflow heat exchanger.

Figure 8.2 Perspective View of FBWHR System

Figure 8.4 Temperature Profile Model

Figure 8.5 Calculated Temperature Profile in Fluidized Bed

When k_* was high, the temperature profile became uniform. The degree to which the profile flattened out could be altered by varying the geometric proportions of the bed. A long narrow bed would decrease the effect of having a high conductivity.

When evaluating the complicated solution of this differential equation at the solids outlet of the heat exchanger, two simplifications could be made. The first simplification, for large k_* , was to that for an ideal mixed flow heat exchanger:

$$\epsilon_p = 1 - \frac{1}{\frac{C_g}{C_s} + 1}.$$

The second, for small k_* , was when a reduction to that of an ideal crossflow heat exchanger could be made:

$$\epsilon_p = 1 - \exp(-C_g/C_s).$$

Laboratory testing showed that this model accurately determined the temperature profile within the bed. Figure 8.6 is a comparison of theoretical and measured temperature profiles showing excellent agreement. Testing also showed that the thermal conductivity was dependent on the superficial velocity, varying from almost 0 at minimum fluidization to over 5000 Btu/hr/ft/°F at 10 ft/sec. This is illustrated in Figure 8.7

Comparison of Figures 8.5 and 8.6 showed the similarity between a shallow fluidized bed and an ideal crossflow heat exchanger. This suggested that applying a constant crossflow efficiency factor (F) to the ideal performance model, as in:

$$\epsilon_p = F [1 - \exp(-C_g/C_s)],$$

yielded results in close agreement with the more complex performance equation. For the FBWHR system design conditions, corresponding to k_* of approximately 0.1, F would range between 0.91 and 0.95.

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Figure 8.6 Experimental Temperature Profile Compared to Curve Fit of Theory to Data

Figure 8.7 Conductivity of Fluidized Bed as a Function of Gas Velocity

Laboratory results and performance calculations by these two methods differed by less than ± 2 percent, providing validation of this simplified nonideal performance model.

Figure 8.8 shows cold stage fluidized bed component thermal effectiveness data extracted from the entire operating range of the FBWHR laboratory system. The data followed the same functionality of increasing performance with increases in solids-to-air capacity ratio.

The theoretical performance line is also plotted in this figure, using the equation:

$$\epsilon_p = F [1 - \exp(-C_a/C_s)].$$

Analysis of the fluidized bed data yielded a functionality of $F = 0.91$ for all experiments. This indicated an identical, repeatable effect on fluidized bed performance during the recycle of lift air. The remainder of scatter in the data, averaging ± 2 percent, could be attributed to radiation effects and experimental error. Thus, for a given component geometry, knowledge of the capacity ratio allowed accurate predictions to be made of the thermal effectiveness of the fluidized bed.

8.2.2 Hot Stage Raining Bed Component

In the upper, hot stage raining bed, shown in Figure 8.9, the hot gas passed up through the raining bed, heated the falling particles, and was then discharged to the atmosphere. The particles were fed from the lift line, passed down through the raining bed where they were heated, and were then circulated by gravity to the cold stage fluidized bed.

The thermodynamic effectiveness of a counterflow heat exchanger was:

$$\epsilon = \frac{1 - \exp[-N(1 - C)]}{1 - C \exp[-N(1 - C)]}$$

Figure 8.8 Comparison of Cold Stage Fluidized-Bed Data to Analytical Predictions

Figure 8.9 Hot Stage Raining Bed

where N was the number of transfer units, and C was the thermal capacity ratio $C_{\min}/C_{\max}$ ($C_{\min}$ was the smaller of C_g and C_s , $C_{\max}$ was the larger of C_g and C_s).

One limiting case of this equation was of particular interest for FBWHR system development: when C was equal to unity. Under this condition, the equation reduced to:

$$\epsilon = \frac{N}{1 + N} \quad (C = 1).$$

It could be shown that the number of transfer units, N , could be expressed as:

$$N = \frac{6 h t}{\rho C_p d_p}$$

where:

h = heat transfer coefficient (Btu/hr-ft²-°F)

t = residence time (hr)

ρ = particle density (lb/ft³)

C_p = particle specific heat capacity (Btu/lb-°F)

d_p = diameter of the particles (ft).

The value of N was to be as high as possible, but could not be determined analytically.

A series of tests were conducted in order to verify the analytical model and collect scale-up data for the raining bed. Figure 8.10 presents a comparison of the hot stage raining bed component performance with the two configurations that were used. Design changes incorporated in the second raining bed configuration allowed an increase in flue gas throughput and heat exchanger duty, while maintaining system preheat capabilities.

There was a marked improvement in component performance attributable to the second raining bed configuration. Over the 0.402-to-3.65

Figure 8.10 Comparison of Hot Stage Raining-Bed Data to Analytical Predictions

range of thermal capacity ratios, the thermal effectiveness at every operating point using the second configuration was better than with the first configuration.

Theoretical performance lines for the two sets of tests are also plotted in this figure, using the equation:

$$\epsilon_g = \frac{C_{\min}}{C_g} \frac{1 - \exp[-N(1-C)]}{1 - C \exp[-N(1-C)]}.$$

Analysis of the raining bed data from the second tests indicated that N increased to 2.960, representing a 50-percent increase from the result of the first baffle set, N equal to 1.967. This verified the 30-percent improvement of N predicted for the second raining bed configuration, and demonstrated an adequate knowledge of the heat transfer mechanism in the raining bed.

The raining bed design changes were predicted to increase component thermal performance at a constant flow rate, as well as to allow constant thermal performance at increased flue gas throughput. In order to accommodate an increase in gas throughput and continue to allow a natural draft exhaust system, the area for gas flow was increased. The resultant decrease in measured design point flue gas pressure loss was from 0.30 in. w.c. to 0.12 in. w.c. This indicated that a 58-percent increase in flue gas throughput with the second configuration would produce the same pressure loss as design flow with the first configuration. The potential 58-percent increase in raining bed throughput closely matched the 50-percent increase in heat exchanger N, demonstrating a theoretical increase in heat exchanger duty with constant preheat capabilities.

Establishing the actual number of transfer units in the raining bed allowed scale-up to a full-sized FBWHR system. Using the expression

$$N = \frac{6 h t}{\rho C_p d_p}$$

changes in particle properties, system velocities or geometries could be accounted for. The unit performance could then be calculated by allowing for the raining bed capacity ratio. Mixing, radiation, and other effects proved to be of a secondary nature and did not factor in the calculations.

8.2.3 Lift Line Component

Figure 8.11 shows the layout of the FBWHR system lift line, in which a pneumatic transport line carried the particles from the lower to the upper bed. Since the particles entering the lift line from the fluidized-bed heat exchanger were hotter than the air, the air was heated as it travelled up the lift line. The two streams exiting the concurrent heat exchanger were separated and fed to the balance of the system. The particles reentered the top of the raining bed. The hot air could either be vented to the atmosphere or recycled to the cold stage fluidized bed.

In the lift line, both the air and the particles travelled in the same direction; thus, it could be modeled as a parallel-flow heat exchanger. The thermodynamic effectiveness of a parallel-flow heat exchanger was:

$$\epsilon = \frac{1 - \exp [- N (1 + C)]}{1 + C}$$

where N was the number of transfer units and C was the thermal capacity ratio $C_{\min}/C_{\max}$ ($C_{\min}$ was the smaller of C_l and C_s , $C_{\max}$ was the larger of C_l and C_s). It could be shown that the number of transfer units, N , could be expressed as:

$$N = \frac{6 h t}{\rho C_p d_p}$$

Figure 8.11 Lift Line

where:

- h = heat transfer coefficient (Btu/hr-ft²-°F)
- t = residence time (hr)
- ρ = particle density (lb/ft³)
- C_p = particle specific heat capacity (Btu/lb-°F)
- d_p = diameter of the particles (ft).

The value of N was to be as high as possible, but could not be determined analytically.

Figure 8.12 plots the results of laboratory tests that verified the predicted behavior of the lift line. The data showed a trend of increasing thermal performance with increases in solids-to-air capacity ratio over the range of 0.879 to 6.56 in capacity ratio.

The theoretical performance line is also plotted in this figure, using the equation:

$$\epsilon_L = \frac{C_{\min}}{C_{\max}} \frac{1 - \exp[-N(1 + C)]}{1 + C}$$

Analysis of the lift line data revealed $N \approx 1.6$ for this series of experiments, indicating identical, repeatable lift line performance. The average error between data and theory for these 35 operating points was 3.4 percent; the maximum error was 22 percent.

Establishing the actual number of transfer units in the lift line allowed scale-up to a full-sized FBWHR system. Using the expression:

$$N = \frac{6 h t}{\rho C_p d_p},$$

changes in particle properties, system velocities, or geometries could be accounted for. The unit performance could then be calculated by allowing for the lift line capacity ratio. Mixing, radiation, and other effects proved to be of a secondary nature and did not factor in the calculations.

Figure 8.12 Comparison of Lift Line Data to Analytical Predictions

8.2.4 Integrated Performance

The performance characteristics of the integrated system were calculated by combining mass and energy balances with the above described performance analysis. Data required to solve the resultant simultaneous equations included:

- Thermal capacity of the flue gas;
- Flue gas inlet temperature;
- Thermal capacity of the combustion air;
- Combustion air inlet temperature;
- Thermal capacity of the lift line air;
- Lift line air inlet temperature; and
- Thermal capacity of the particles.

Assumptions included in the analysis were:

- Steady-state operation of the unit; and
- No system heat losses present.

Calculations resulted in a prediction of the steady-state integrated performance, including the preheated combustion air temperature.

8.3 FBWHR SYSTEM OPERATION

8.3.1 Full-Fire Furnace Operation

The full-scale FBWHR system was designed for retrofit onto an existing open hearth aluminum smelting furnace to produce preheated combustion air, allowing the furnace firing rate to be reduced. The operations of the furnace at the installation site were described elsewhere in this report, as was the planned method of using laboratory results and correlations to calculate the operating characteristics and design basis of a demonstration unit. This section describes the results of that work.

The pre-retrofit furnace operated at a firing rate of 16 MMBtu/hr, with 13,520 lb/hr of cold combustion air. The resultant waste flue gas flow rate was 14,190 lb/hr, at a temperature of 2000°F. This information was used as input to the integrated performance analysis described above. Results of the analysis included a prediction of the steady-state full-fire integrated system performance.

The post-retrofit furnace/FBWHR system flow sheet at this full-fire condition is shown in Figure 8.13. Results included a 36-percent reduction in furnace fuel input, from 16 MMBtu/hr to 10.2 MMBtu/hr; 1209°F preheated combustion air temperature; 2080 Btu/hr-°F preheated combustion air thermal capacity; a reduction in furnace waste gas temperature to 1160°F, waste gas thermal capacity of 2699 Btu/hr-°F; and a circulating particle thermal capacity of 2969 Btu/hr-°F. Half of the 2080 Btu/hr-°F of combustion air was passed up the lift line to carry the particles up to the separator, where the preheated lift air and alumina particles were separated. The 746°F lift air was then returned to the fluidized bed plenum, where it mixed with an equal amount of 120°F combustion air, resulting in a 433°F plenum temperature. Particle temperatures in the system ranged between a maximum of 1619°F at the discharge from the raining bed, to a minimum of 856°F at the entrance of the raining bed.

8.3.2 Minimum Operating Point

The post-retrofit furnace/FBWHR system flow sheet at its chosen minimum operating condition is shown in Figure 8.14. This operating point included a 50-percent reduction in furnace firing rate from the full fire recuperated point, to 5.1 MMBtu/hr, and a furnace flue gas thermal capacity of 1350 Btu/hr-°F, at a temperature of 1400°F. At this minimum operating point, only 1040 Btu/hr-°F of 883°F preheated combustion air was produced. All of this combustion air was used to carry 2969 Btu/hr-°F of particles from the lower fluidized bed to the

Figure 8.13 Post-Retrofit Furnace/FBWHRs Flow Sheet
(10.2 MMBtu/hr at Full-Fire)

Figure 8.14 Post-Retrofit Furnace/FBWHRS Flow Sheet
(5.1 MMBtu/hr Minimum Operating Point)

separator above. The preheated lift air and particles were separated in the separator. The 612°F lift air was then returned to the fluidized bed plenum, where it provided all the fluidization air for the system. Particle temperatures in the system ranged from a maximum of 965°F at the raining bed outlet to a minimum of 698°F at the raining bed inlet.

8.3.3 Operation Below Minimum Operating Point

In order to reach the low-fire conditions of the furnace, 3 MMBtu/hr with no recuperation, the furnace/FBWHR system firing rate was reduced from its maximum operating point of 10.2 MMBtu/hr. The FBWHR system was designed to reach this turndown point by turning the particle media flow rate off, thereby stopping system heat transfer. Concurrently reducing the combustion air flow rate to 612 Btu/hr-°F allowed the furnace to be run at its 3 MMBtu/hr low-fire condition.

At this operating condition, relatively cool 1400°F flue gases passed up the raining bed. Cold air slowly flowed up the lift line and back down to the fluidized bed. This low flow of air was insufficient to fluidize the bed, so it merely "trickled" up through the now packed bed of particles. Since the bed was not fluidized and particle flow was stopped, the combustion air was not heated significantly beyond the 120°F combustion air fan outlet temperature. The unrecuperated air passed to the burners.

The operating limitations of a pneumatic lift line and a fluidized bed made it necessary to stop the media flow and temporarily cease the air preheating process. Figure 8.15 illustrates these boundaries, where the system limitations are shown at various fractions of total combustion air used for pneumatic lift. At low ratios of lift air, the fluidized bed limited FBWHR system operation due to minimum fluidization limits. Conversely, at high ratios of lift air, the FBWHR system was limited by minimum lift velocity limits. Consequently, the optimal

Figure 8.15 Limitations of Turndown Due to Air Used in Pneumatic Lift

system turndown was reached at 50 percent lift air, where the fluidized bed and lift line limits were concurrent.

Finally, the effects on energy savings due to the selected 2:1 FBWHR system turndown needed to be examined. It was known from the above characterization of the aluminum smelting furnace that over 90 percent of the potential energy savings were available during 2:1 turndown on a furnace. Any increase in FBWHR system turndown would produce only a minimal increase in energy savings. For example, improving system turndown from 2-to-1 to 5-to-1 would produce no more than a 10-percent increase in annual energy savings. Thus, FBWHR system turndown was set at 2:1, allowing recuperation of over 90 percent of the potential waste energy from the furnace.

8.4 FBWHR SYSTEM DESIGN BASIS

Combining the above laboratory process and operational details with installation site furnace sizing information resulted in the full-scale FBWHR system design basis, as shown in Table 8.1. This table indicates the inputs to the detailed design. Particle properties were chosen to match those of the laboratory unit and included an average particle size of 24 mesh, a particle sphericity of 0.9, a particle density of 3.6 specific gravity, a bulk density of 130 lb/ft³, and an average particle heat capacity of 0.294 Btu/lb-°F.

The raining bed flue gas inlet temperature was chosen to be 2000°F, with a system preheated air temperature of 1200°F. The burner firing rate was set at 10.2 MMBtu/hr, with a preheated combustion air flow rate of 8028 lb/hr, and a flue gas flow rate of 8454 lb/hr.

The fluidized bed height was 2 inches when fluidized, matching the laboratory unit. Simple scale-up of the fluidized bed plan area resulted in a dimension of 9.80 ft²; similar scale-up of the raining bed stack area resulted in a dimension of 23.0 ft², and the lift line flow area increased

TABLE 8.1
FBWHR SYSTEM DESIGN BASIS

Particle Diameter	24 Mesh
Particle Sphericity	0.9
Particle Density	3.6 S.G.
Bulk Density	130 lb/ft ³
Particle Heat Capacity	0.294 Btu/hr-°F
Flue Gas Inlet Temperature	2000°F
Preheated Air Temperature	1200°F
Burner Firing Rate	10.2 MMBtu/hr
Air Flow Rate	8028 lb/hr
Flue Gas Flow Rate	8454 lb/hr
Fluidized Bed Height	2 inches
Fluidized Bed Plan Area	9.80 ft ²
Raining Bed Stack Area	23.0 ft ²
Lift Line Flow Area	0.253 ft ²
Ratio of Fluidized Bed Distributor Plate Pressure Difference at Minimum Design Point to Fluidized Bed Pressure Difference	0.50

to 0.253 ft^2 . Finally, the ratio of the fluidized bed distributor plate pressure difference, at the minimum design point, to the fluidized bed pressure difference was maintained at 0.50, matching that of the laboratory unit.

8.5 DESIGN OUTPUTS

The information just presented coupled with standard calculational techniques already outlined resulted in the FBWHR system design outputs shown in Table 8.2.

The fluidized bed pressure difference at full fire was 4 in. w.c., while the fluidized bed distributor plate pressure difference was 6.69 in. w.c. Inlet conditions of the fluidized bed plenum included an air temperature of 433°F , an air density of 0.0445 lb/ft^3 , and a mass flow rate of 133.8 lb/min. The distributor plate orifice diameter was 0.078 in.; the corresponding orifice coefficient times expansion factor was 0.8. The total number of orifices was 8094, resulting in an open area for flow of 0.279 ft^2 , with a corresponding open area of 2.85 percent. The velocity at the orifices was 180 ft/sec. Fluidized bed outlet air temperature was 1209°F , at a density of 0.0238 lb/ft^3 . Average bed superficial velocity was 9.55 ft/sec, resulting in a ratio of full fire to minimum operating point superficial velocities of 2.49. The calculated terminal velocity of the particles was 23 ft/sec.

The raining bed pressure difference was 0.06 in. w.c. Inlet flue gas conditions included 2000°F waste gas temperature, flue gas density of 0.0162 lb/ft^3 , and inlet flue gas mass flow rate of 140.9 lb/min. The raining bed inlet diameter was 47 in., resulting in an inlet flue gas superficial velocity of 12.0 ft/sec. The outlet flue gas temperature was 1160°F , with a density of 0.0246 lb/ft^3 , yielding an outlet flue gas superficial velocity of 4.15 ft/sec.

System particle flow rate was 168.3 lb/min, resulting in a lift line pressure loss of 9 in. w.c. Pneumatic lift line air inlet conditions

TABLE 8.2
FBWHR SYSTEM DESIGN OUTPUTS

Fluidized Bed Pressure Difference	4 in. w.c.
Fluidized Bed Distributor Plate Pressure Difference	6.69 in. w.c.
Inlet Air Temperature	433°F
Inlet Air Density	0.0445 lb/ft ³
Inlet Air Mass Flow Rate	133.8 lb/min
Orifice Diameter	0.078 in.
Orifice Coefficient x Expansion Factor	0.8
Number of Orifices	8094
Open Area	0.279 ft ²
Percent Open Area	2.85%
Velocity at Orifices	180 ft/sec
Outlet Air Temperature	1209°F
Outlet Air Density	0.0238 lb/ft ³
Average Bed Superficial Velocity	9.55 ft/sec
Ratio of Full Fire to Minimum Operating Point Superficial Velocities	2.49
Terminal Velocity	23 ft/sec
Raining Bed Pressure Difference	0.06 in. w.c.
Inlet Flue Gas Temperature	2000°F
Inlet Flue Gas Density	0.0162 lb/ft ³
Inlet Flue Gas Mass Flow Rate	140.9 lb/min
Raining Bed Inlet Diameter	47 in.
Inlet Flue Gas Superficial Velocity	12.0 ft/sec

TABLE 8.2 (Cont'd)
FBWHR SYSTEM DESIGN OUTPUTS

Outlet Flue Gas Density	0.0246 lb/ft ³
Outlet Flue Gas Temperature	1160°F
Outlet Flue Gas Superficial Velocity	4.15 ft/sec
Particle Flow Rate	168.3 lb/min
Lift Line Pressure Loss	9 in. w.c.
Inlet Lift Air Temperature	120°F
Inlet Air Density	0.0686 lb/ft ³
Inlet Air Mass Flow Rate	66.9 lb/min
Inlet Lift Pipe Diameter	6.81 in.
Inlet Lift Velocity	64.2 ft/sec
Outlet Lift Air Temperature	746°F
Outlet Lift Air Density	0.0330 lb/ft ³
Average Lift Air Velocity	98.9 ft/sec
Combustion Air Blower Discharge	44 oz/in ²

were: a temperature of 120°F, an air density of 0.0686 lb/ft³, a mass flow rate of 66.9 lb/min, a lift pipe diameter of 6.81 inches, and an inlet lift velocity of 64.2 ft/sec. Lift line air outlet conditions included a temperature of 746°F and a density of 0.0330 lb/ft³. The average lift air velocity was 98.9 ft/sec. Combustion air blower discharge pressure requirements were 44 oz/in².

8.6 FURNACE RETROFIT

The installation site for the FBWHR system demonstration, Vulcan Materials Company (VMC), Oak Creek, Wisconsin, was responsible for the installation and retrofit work. This involvement of VMC with the project was intended to assist with FBWHR system operation by familiarizing the host company with the system early on. This early involvement led to design inputs from plant and corporate personnel, resulting in an improved demonstration unit. The improvements to the demonstration unit reduced the number and severity of startup and operational changes required for successful FBWHR system operation.

8.6.1 Placement

The location of the FBWHR system was chosen to be near the north end of the Oak Creek furnace room, as shown in Figure 8.16. This site was selected because it provided easy access from three sides of the waste-heat recovery system. Furnace No. 1 was on the north end of a row of furnaces; thus, placement to the north of Furnace No. 1 afforded fairly unlimited access on the sides of the FBWHR system facing away from the furnace. This good access to the installation area eased the equipment handling requirements during field installation.

Secondary considerations that affected the placement of the demonstration unit included the anticipated furnace operating time to the next major furnace rebuild. Furnace No. 1 was rebuilt recently and was to have additional maintenance performed on it while the FBWHR

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Figure 8.16 FBWHR System Placement

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system was installed. There was to be only one short maintenance shutdown during the FBWHR system performance testing; thus, interruptions would be minimal during the test program.

Other factors that affected FBWHR system siting included furnace room limitations. These spanned the range of typical site-specific requirements and included:

- Keeping FBWHR system components away from a nearby driveway and the molten aluminum pour pit,
- Keeping all FBWHR system components, except the exhaust stack, below the roof line, and
- Not moving any existing major structural steel.

8.6.2 System Integration

As described above, the FBWHR system was being installed to the north of and immediately adjacent to Furnace No. 1. See Figures 8.17 and 8.18. An alloy stack-cap damper was to be used to seal the top exit of the existing flue when the recuperator was operating. When this occurred, a transition duct attached to the end of the existing flue was to channel the 2000°F waste gas out the end of the flue and into the bottom of the raining bed. An alloy guillotine panel could be lowered into the transition duct to block waste gas flow to the FBWHR system when the recuperator was off-line.

Combustion air was to be piped from a new, floor-mounted blower to the inlet flanges at the bottom of the fluidized bed and the bottom of the lift line. Preheated air was to be brought from the top of the lift line separator/hopper to a flange at the bottom of the fluidized bed, where it would mix with the balance of combustion air from the blower. This tempered air would be passed through the fluidized bed, heated to 1209°F, and then exit out the side of the lower bed. This preheated air would be passed to a header atop Furnace No. 1, and then through

Figure 8.17 Plan View of FBWHR System Retrofit

Figure 8.18 Elevation View of FBWHR System Retrofit

two branches to the individual burners. All piping containing preheated air was surrounded by four inches of external insulation.

When the FBWHR system was off-line, combustion air was to be provided by an existing blower. The air would enter the piping downstream of the fluidized bed, allowing the bed to be isolated from the furnace. This isolation could be implemented only when two hand valves were operated, avoiding any chance of accidental changeover. Use of the existing combustion air blower, along with switching valves in the flue gas and combustion air piping, would allow for uninterrupted furnace operation. Aluminum production could occur either with the FBWHR system in operation, or with the bypass air system, eliminating any furnace downtime.

Expansion joints were required in the piping to accommodate the differential thermal expansion. Axial, angular, and lateral forces could bend or break piping, or produce air leakage at flanges. Standard expansion joints were placed at required locations in the piping system, including the individual ducts leading to the burners. The piping was supported and guided by standard alloy anchors and guides from the support superstructure. Figure 8.19 is a piping schematic of the preheated combustion air piping between the FBWHR system and the two burners on the furnace, and shows its expansion joint/guide/anchor system.

The burners selected for the FBWHR system retrofit were high-temperature versions of the burners already in use at the Oak Creek plant, and were capable of firing both gas and fuel oil. The need for dual-fuel burners was dictated largely by VMC's requirement of burning either gas or oil, depending upon price and availability, in order to keep the plant running.

Furnace pressure was to be controlled by air damping in the same manner as it was before using the recuperator. Air was to be supplied

Figure 8.19 Preheated Combustion Air Piping

to the damper by the new FBWHR system combustion air blower. One air damper was placed atop the raining bed stack, blowing air across the recuperator opening. Air flow would be controlled by the existing VMC control system, using their high-fire/low-fire, two-position control strategy.

8.7 FBWHR SYSTEM PROCESS CONTROL AND INSTRUMENTATION

The instrumentation for the full-scale FBWHR system included 47 sensors for measuring temperatures, pressures, pressure differences, and flow rates. These instruments were to be used for performance data gathering, system monitoring, and process control. Figure 8.20 is a block diagram of the instrumentation and control components. Most of the instrumentation was intended to determine system performance during the test period. Thirteen of the sensors provided inputs to the control system.

Abbreviations used in this section are generally in accordance with the standard instrument symbols and identification system outlined by ANSI Y32.20. A summary of pertinent designations is listed in Table 8.3.

8.7.1 Overall Operating Concept

The FBWHR/furnace system was designed to operate at either of two set firing rates. During charging and melting, the burners were to run at high fire; during periods of furnace idling, the fuel flow to the burners was to be greatly reduced. The firing rate was determined by any one of three inputs. If the molten aluminum temperature or exiting flue gas temperature was too high, the furnace was to automatically change to low-fire operation. Low fire could also be switched on manually.

During high fire, the fuel flow was to be set at a fixed rate previously determined to maintain flue gas temperature at optimum values.

Figure 8.20 Instrumentation and Control

TABLE 8.3
DESIGNATIONS SUMMARY

FCV	Flow Control Valve
FFV	Flow Ratio (Fraction) Valve
FIC	Flow Indicating Controller
FT	Flow Transmitter
HS	Hand Switch
HSV	Hand Switching Valve
I	Interlock
LV	L Valve
M	Motor Actuator
PCV	Pressure Control Valve
PDI	Pressure Difference Indicator
PDIS	Pressure Difference Indicating Switch
PI	Pressure Indicator
S	Solenoid Actuator
TAH	Temperature Alarm High
TAHH	Temperature Alarm Very High
TCV	Temperature Control Valve
TE	Temperature Element
TIC	Temperature Indicating Controller
TSV	Temperature Switching Valve
TT	Temperature Transmitter
TY	Temperature Low-Pass Relay

The furnace stack would be shut and the exiting flue gases would pass through the transition duct and up through the raining bed. Furnace pressure was to be maintained by a constriction at the top of the raining bed stack and an associated air damper. Combustion air from the FBWHR system blower would be throttled to maintain the air-to-fuel ratio entering the burners by a combustion controller capable of continuous temperature compensation.

Particle flow rate was to be controlled by using three L-valves. In an L-valve, an L-shaped pipe, particles enter at the top of the vertical leg, filling it to form a seal between the inlet and outlet (see Figure 8.21). Particle flow through the valve is initiated and controlled by "trigger" air blowing down the length of the horizontal leg to the outlet. The flow rate of the trigger air determines the flow rate of the particles. The L-valves not only maintain the proper particle flow through the system, but also keep the flue gas, hot combustion air, and cold combustion air separate.

The final ingredient necessary for the FBWHR system to operate at high fire was a means of transporting the particles to the top of the raining bed. As the particles exited the fluidized bed L-valve, they would be blown up a lift line to a separator/hopper at the top of the raining bed by a portion of the combustion air. The amount of combustion air going up the lift line was to be controlled to maintain a set velocity for particle transport. The remainder of the combustion air would go directly to the fluidized bed plenum.

When the furnace would go to low fire, the flow rate of combustion air dropped below the amount necessary to transport particles up the lift line. This flow rate was to be measured; a low flow switch would stop the trigger air in all the L-valves, so that particles did not collect at the base of the lift line to cause a blockage. The flue gas would be allowed to pass through the raining bed in the absence of falling particles

Figure 8.21 L-Valve On-Off Feed Control System

because its temperature would be greatly reduced at low fire. When low fire was initiated, air jet dampers were to be operated fully to maintain furnace pressure.

Isolation valves were provided so that the FBWHR system could be completely bypassed, enabling the use of the original furnace blower and flue stack. The valving was arranged so that either the flue gas or combustion air flow could be bypassed independently if required. Only the flue gas was to be bypassed in the event of a high temperature excursion in the raining bed. During shakedown, cold combustion air supplied by the original blower could be used for the burners, while the air from the FBWHR system blower would then be vented after preheating. During FBWHR system maintenance the entire recuperation system could be isolated.

8.7.2 Instrumentation

All instrument readings necessary for determining system performance would output to a trend/data logger chart recorder. The recorder was capable of transmitting all sensor output onto a tape or disk for future data reduction and analysis. The recorder was to scan all input channels and display those needed during any phase of operation.

In addition, there was a digital temperature display which could be switched to any thermocouple channel for continuous monitoring. These were mounted in a NEMA-12 control panel, along with the control instrument readouts discussed in the next section.

All temperatures were to be measured using type K chromel/alumel thermocouples. The 24 thermocouples are listed in Table 8.4, along with a description of the temperature measured and sensor function. The thermocouple positions are shown on a simplified schematic in Figure 8.22. TE213 and TE305 were surface mounted to the raining bed and the fluidized bed to record outer shell surface temperatures.

TABLE 8.4
TEMPERATURE MEASUREMENTS

Thermocouple	Description	Function
TE115	Air Leaving Blower No. 2	P
TE116	Particles in Separator/Hopper	P
TE117	Lift Line Return Air	P
TE209a,b,c	Particles in Fluidized Bed	P
TE210	Air in Fluidized Bed Outlet	P
TE212a,b,c	Air in Fluidized Bed Inlet Plenum	P
TE213	Outer Shell of Fluidized Bed Chamber	M
TE214	Particles at Fluidized Bed Exit	P
TE303	Flue Gas in Transition Duct	P
TT304a,b	Raining Bed Plenum	M,C
TE305	Raining Bed Outer Shell	M
TE306a,b	Flue Gas Exiting Raining Bed	P
TE307	Particles Entering Raining Bed	P
TE308	Particles Exiting Raining Bed	P
TT401	Metal in Furnace	C
TT402	Flue Gas in Furnace	C
TE418	Air Leaving Blower No. 1	P
TT419	Air Entering Burners	P,C

P - Performance Data
C - Control
M - Monitoring

Figure 8.22 Post-Retrofit Furnace/FBWHRS Temperature Measurement

TE303, TE304 a,b, and TE306 a,b had mullite (63 percent alumina) sheaths for protection from the high-temperature, corrosive environment of the flue gas. The remaining sensors had metallic sheathing. TE209 a,b,c were inserted at a height of two inches above the distributor plate to measure bed temperature near the longitudinal centerline.

Pressures and pressure differences were to be measured at various points to monitor component performance. Figure 8.23 shows the location of these instruments on a schematic. Pressure gauges PI404 a,b and PI431 a,b were to be used to balance fuel and air flow to the burners. The remainder of the measurements would have been made on locally mounted differential pressure gauges. A complete list of the pressure measuring instruments is given in Table 8.5. This table includes the pressure differential switches used to maintain the seal height in the L-valves. Because the pressure differential switch connections were in a particle-filled region, small preset quantities of purge air were to be constantly brought through the high and low pressure legs of the switches. This low purge air flow was intended to prevent the tap openings from becoming blocked by particles. The function of these switches will be explained in a subsequent subsection of the report.

Air and fuel flow to the burners was to be measured using orifice plates. The air flow in the lift line would have been measured using a pitot tube. Table 8.6 lists these flowmeters.

8.7.3 Combustion Control System

The FBWHRs combustion control system is shown in Figure 8.24. TCV403 had preset limits, so that high/low control could drive the temperature control valve to the high or low fuel flow rate. A low-pass relay, TY430, was normally in the high-fire position, but would have switched to low fire when any one of three low-fire signals were inputted. If the molten metal temperature in the furnace exceeded a set temperature, TIC401 would input a low-fire signal to the low-pass relay.

Figure 8.23 Post-Retrofit Furnace/FBWHRs Pressure Measurement

TABLE 8.5
PRESSURE MEASUREMENTS

Component	Description	Function	Range
PDIS120a,b	Fluidized Bed Outlet L-Valve Seal Height	C	0-40 in. w.c.
PI130	Blower No. 2 Outlet Pressure	P	0-5 psig
PI132	Lift Line Inlet Pressure	P	0-5 psig
PDI133	Lift Line Pressure Difference	P	0-40 in. w.c.
PDIS134	Separator/Hopper Particle Height	C	0-40 in. w.c.
PI135	Lift Line Return Pressure	P	0-4 psig
PI201	Fluidized Bed Air Outlet Pressure	P	0-2 psig
PDI202	Distributor Plate Pressure Difference	P	0-10 in. w.c.
PDI203	Fluidized Bed Pressure Difference	P	0-10 in. w.c.
PDI204	Air Inlet to Air Outlet Pressure Difference in Fluidized Bed	P	0-20 in. w.c.
PDIS220a,b	Raining Bed Outlet L-Valve Seal Height	C	0-40 in. w.c.
PDI302	Raining Bed Pressure Difference	P	0-0.25 in. w.c.
PDIS320	Lift Line Outlet L-Valve Seal Height	C	0-40 in. w.c.
PI404a,b	Fuel Inlet Pressure to Burners	B	0-30 in. w.c.
PI411	Furnace Pressure	P	± 0.25 in. w.c.
PI431a,b	Air Inlet Pressure to Burners	B	0-30 in. w.c.

P - Performance Data
B - Burner Balancing
C - Control

TABLE 8.6
FLOW MEASUREMENTS

Component	Description	Function
FT131	Lift Line Air Velocity	P,C
FT403	Fuel Flow Rate	P,C
FT430	Preheated Air Flow Rate	P,C

C - Control

P - Performance Data

Figure 8.24 Post-Retrofit Furnace/FBWHRS Combustion Control

Similarly, if the flue gas temperature exceeded a certain value, TIC402 would signal for low fire. Low fire could also have been initiated manually by using HS430.

During standard operation, combustion air was to be supplied by Blower No. 2 and passed through the fluidized bed for preheating prior to entering the burners. The fuel-to-air ratio would have been maintained by throttling the air leaving Blower No. 2 using FFV130. This valve would be actuated by a signal from a combustion air-temperature-compensated ratio controller. This combustion controller required three inputs. FT403 inputted a signal that was proportional to the planned fuel flow rate. FT430 sent the same information regarding the combustion air flow to the burners, which needed temperature compensation because of the variability of the preheat temperature. This input was supplied by TT419. Using these inputs, the blower throttling valve was to be controlled to keep the combustion air in ratio with the fuel.

As the flue gases passed through and out the raining bed stack, furnace pressure was to be controlled at the stack outlet above the roof line. Since high fire was a set flow rate, this could have been accomplished by constricting the stack outlet and presetting the closed position of PCV311 to obtain the required furnace pressure. When TY430 actuated low fire it also sent a signal to PCV311 causing it to open to a second preset position. This signal closed and opened a pressure control valve in the line from the output of Blower No. 2 to an air jet damper located at the raining bed stack outlet. The control valve's preset limits would have allowed the proper air flow to the damper, maintaining proper furnace pressure.

8.7.4 Particle Circulation Control System

The flow rate of the particles used as transfer media was to be controlled by three L-valves located between the separator hopper and

the raining bed (LV320), the raining bed and the fluidized bed (LV220), and the fluidized bed and the lift line (LV120). Particles were to fill the vertical leg of an L-valve to form a seal between the components on its inlet and outlet ends. The height of the vertical leg of each L-valve was set to ensure that a large enough column of particles could collect to overcome the pressure difference between the various components. Particle flow through the valves was to be controlled by the amount of trigger air flowing through the horizontal leg of each valve. Figure 8.25 shows the location of the L-valves on a simplified schematic.

LV120 and LV220 had two trigger air connections, while only one connection was required for LV320. The amount of air entering each connection was preset using a control valve to modulate the air supply. The valve was to have been actuated by a solenoid that stopped the air flow when closed or allowed the preset amount to flow when open. LV320 was set to circulate the desired amount of particles through the system. The particle height in that valve was monitored by a pressure differential switch (PDIS320) that had one tap at the top of the separator/hopper and the other on the vertical leg of the valve at a height safely above that necessary to maintain the seal between the separator/hopper and the raining bed. If the particle column height approached a minimum level, PDIS320 would have sounded an alarm warning that particles needed to be added to the system. If the particle column height dropped to an even lower level in the vertical leg, the pressure difference measured by the switch would have caused the trigger air to stop until the column height rose to an acceptable level.

LV120 and LV220 had two taps in their vertical legs, one at the minimum required safe column height, and one higher. When the column height was below the minimum, both trigger air lines were to be closed until a safe level was attained. When the level was between the two taps, PDIS120A or PDIS220A (respectively) opened one of the

Figure 8.25 Post-Retrofit Furnace/FBWHRS Particle Flow Control

trigger air ports and initiated particle flow. If the column height exceeded the upper tap level, PDIS120B or PDIS220B was to open a second trigger air port. This would keep all the surplus particle inventory in the separator/hopper, and eliminate excessive particle collection at the intermediate points in the system.

To guard against overfilling the separator/hopper, a differential pressure switch, PDIS134, was used, with one tap at the top of the separator/hopper and one at the maximum desired particle inventory height. If the particle level exceeded the lower tap level, PDIS134 would have sounded an alarm and signalled the interlocks on LV120 and LV220 to stop all trigger air flow until the particle level in the separator/hopper was lowered. The most likely cause of an overfull separator/hopper was predicted to be an overfilling of particles in the system during periodic media makeup occurrences.

The lift line air flow transported the particles exiting LV120 to the separator/hopper at the top of the raining bed. Air velocity in this line was to be maintained at a set level to ensure proper operation of the particle circulation system. A minimum velocity was required to lift the particles to the hopper; exceeding this value could cause excessive abrasion in the lift line and hopper as well as higher piping pressure losses. FT131 measured the lift line air velocity and provided the input to control FCV131, which would have bypassed any air in excess of that needed for minimum lift line velocity directly to the fluid bed inlet plenum.

When the furnace switched to low fire, the flow rate of combustion air dropped below the minimum required to maintain lift line velocity. In this event, FIC131 was to signal all the L-valve interlocks to shut down the trigger air supplies and stop particle flow.

8.7.5 Recuperator Bypass Control System

The FBWHR system could be partially or completely bypassed for special operating conditions or if system maintenance was required. Figure 8.26 shows the valves, piping, and support equipment required during various bypass operations.

The furnace could have been run independently of Blower No. 2 during maintenance or FBWHR system cold testing. The combustion air would then be supplied by the original furnace blower. HSV230 and HSV430 were mechanically cross-linked so that when HSV430 was opened to allow Blower No. 1 to feed the burners, HSV230 closed to isolate the fluidized bed. Fuel-to-air ratio would then be controlled by a direct mechanical cross-linkage between TCV403 (controlling fuel flow) and FFV430 (throttling the air flow from Blower No. 1). In the case of FBWHR system shakedown and testing, the system could be run with the flue gases passing through the raining bed and the preheated combustion air from Blower No. 2 vented by opening HSV231.

During maintenance, TSV432 could have been opened to vent furnace flue gases through the original furnace flue stack. A guillotine valve, HSV309, was shut to isolate the raining bed from the flue gases. HSV433 was opened to allow air dampers in the transition duct to operate. This allowed the guillotine valve to be installed during furnace operation. The transition duct air dampers could be left open for cooling the guillotine after it was in place. Furnace pressure was to be controlled in the same manner as it was during FBWHR system operation. A preset constriction and minimal air jet damping in the flue stack were to be used during high fire operation; the air jet damper was to be opened fully at low fire. PCV411 was actuated by a signal from TY430.

During standard FBWHR system operation, the raining bed could be automatically bypassed in the event of an over-temperature excursion

Figure 8.26 Post-Retrofit Furnace/FBWHRS Recuperator Bypass

in the raining bed plenum. If the plenum temperature exceeded a preset level, TT304 would trigger a high-temperature alarm to alert the operator. If the temperature continued past a higher alarm point, TSV432 would automatically have been opened to vent flue gases out the original furnace flue stack.

8.8 ECONOMIC ANALYSIS

The benefits of installing the recuperator are twofold: an increase in production capacity and a reduction in energy use. The energy savings were calculated in Section 8.3 of this report. For the same production rate, energy consumption at the maximum firing rate will be reduced from 16 MMBtu/hr to 10.2 MMBtu/hr. Using the duty cycle presented in Figure 3.24 and 325 days of production annually result in a fuel savings of 38,000 MMBtu/yr. At \$3.50 per MMBtu the annual cost savings is \$133,000, and at \$6.00 per MMBtu the savings is \$228,000 annually.

The real benefit of recuperation is, however, the increase in production capability. With 1200°F air preheat, production capability would increase approximately 35 percent. Thus an 85,000-pound-per-day melting furnace should be able to melt 115,000 pounds of aluminum per day. At \$1.00 per pound, this extra aluminum has a market value of \$30,000 and a net value to the manufacturer of \$3000. Over a 325 day year, this is an increase in margin of nearly \$1,000,000 annually.

Cost of the prototype FBWHR system was \$312,000 and installation an additional \$100,000, for an installed cost of \$412,000. For a subsequent production unit, cost of the recuperator could be reduced by 25 percent, for a total installed cost of \$334,000. Thus payback on an energy conservation basis is 2½ years and an increased production basis is 4 months. However, the secondary aluminum industry currently has surplus capacity and is therefore very limited in its interest for increased capacity. Therefore, the only industry interest is based upon operating cost savings. To meet the industry criteria of a 1½ year payback of investment, energy costs would have to rise to \$6.00 per MMBtu.

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9. FIELD INSTALLATION OF THE FULL-SCALE FBWHR SYSTEM

9.1 SHOP FABRICATION OF THE FBWHR SYSTEM

The full-scale FBWHR system was fabricated at an ASME-coded subcontractor's facility in Massachusetts. Both the flue gas ductwork to the FBWHR system and the recuperator's outer shell were constructed from mild steel (shown in Figures 9.1 and 9.2). The FBWHR system internals and preheated combustion air piping were fabricated from various alloy steels depending upon the required materials, properties, costs, and availabilities. Chapters 4 and 6 of this report contain details of the materials selection process.

A portion of the refractory anchors and system insulation was shop-installed, shown in Figure 9.3. Where appropriate, other portions of the FBWHR system were installed prior to shipping, in order to minimize the unit's installed cost.

9.2 FIELD WORK

Vulcan Materials Company (VMC) was responsible for preparing the field demonstration site to accept the FBWHR system, as well as all field installation work.

9.2.1 Site Preparation

Preliminary FBWHR system shop fabrication drawings were supplied to VMC in November, 1985; final drawings were sent in February, 1986. These drawings, which detailed the interfaces between Tecogen-supplied equipment and VMC-supplied equipment, were used by VMC to plan the field site modifications prior to receiving the FBWHR system.

VMC, as was their custom with plant capital improvements, planned to use their in-house maintenance staff for this project. Initially, existing service facilities that obstructed the site were moved or removed to clear the installation area. On March 10, 1986, VMC broke

Figure 9.1 Flue Gas Ductwork

1210-N

Figure 9.2 Recuperator Outer Shell

Figure 9.3 Installing Refractory Anchors

ground for the foundation. Subsequently, the concrete foundation was poured and the structural steel support frame (Figure 9.4) was erected. Preparations were completed in advance of the fall delivery of the partially pre-assembled FBWHR system components.

9.2.2 Field Installation

On October 6, 1986, the first shipment of FBWHR system components arrived at the VMC-Oak Creek plant. VMC's field erection of the various subassemblies closely followed the anticipated schedule. Figure 9.5 shows work during the installation of the fluidized bed component, and Figure 9.6 shows the site during the erection of the particle lift line subassembly.

A view of the installation of the preheated combustion air piping is shown in Figure 9.7. Close-up photographs of the preheated combustion air burners are included as Figures 9.8 and 9.9. By late November, 1986, all major components were in place (Figures 9.10 and 9.11).

Unfortunately, on November 20, a six-week postponement was forced on the project by VMC. The reason given by VMC for this delay was that the maintenance workers assigned to the project had to be moved to an urgent plant repair job, to repair their only ingot casting machine and one of their rotary kilns.

The early months of 1987 saw slow progress on the FBWHR system installation. Repeated delays occurred as men were removed from FBWHRs installation work to perform necessary plant maintenance work. Additionally, the installation of a new small rotary kiln dryer system was given higher priority by VMC. Consequently, only a limited amount of additional work had been done by the spring of 1987.

9.3 SALE OF THE OAK CREEK PLANT

All work on the FBWHR system installation was halted May 8, 1987, pending final sale of the Oak Creek plant. Wabash Alloys, purchaser

Figure 9.4 Structural Steel Support Frame

1211-N

Figure 9.5 Installation of the Fluidized Bed Component

Figure 9.6 Erection of the Particle Lift Line Subassembly

Figure 9.7 Preheated Combustion Air Piping

Figure 9.8 Preheated Combustion Air Burner - Exterior View

1207-N

Figure 9.9 Preheated Combustion Air Burner - Interior View

Figure 9.10 East Face of FBWHR

Figure 9.11 West Face of FBWHR
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of the plant, took possession of the plant effective June 6, 1987. Prior to purchasing the facility, Wabash had agreed to complete the FBWHR system demonstration project.

During the ensuing two months, Wabash was busy incorporating the Oak Creek plant into their newly expanded aluminum smelting operations (30+ furnaces at 5 plants), and did not yet have personnel available for the FBWHRS installation. These personnel were not provided within the time constraints of the project and hence the installation was never completed.

10. CONCLUSIONS

The FBWHR system is a viable concept for recovering waste heat from dirty industrial flue gases. Subscale laboratory testing provided the scale-up data required for building a full-scale demonstration unit. Experimental results of this program have shown:

- The measured design point overall thermodynamic effectiveness is approximately 57 percent, resulting in an air preheat temperature of 1200°F from 2000°F flue gases.
- The measured laboratory performance of the FBWHR system taken over wide-ranging particle circulation, gas flow, and gas temperature conditions agreed well with the analytical predictions.

The technology to build and operate a commercial system now exists. However, the economic viability of recuperator systems is predicated upon expensive energy. With energy prices dropping, payback periods stretch, and recuperation systems are less attractive. This is true for any recuperation system, as well as for the FBWHR system. The Department of Energy, therefore, has reoriented its heat exchanger research and development programs away from recuperation and toward higher technology applications. In the future, when energy prices increase to higher levels, FBWHR system technology will be available.

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