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Engineering Development of Coal-Fired High Performance Power Systems Phase II and III

DE-AC22-95PC95144

Quarterly Progress Report

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Prepared for

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EXECUTIVE SUMMARY

This report presents work carried out under contract DE–AC22–95PC95144, "Engineering Development of Coal–Fired High Performance Power Systems Phase II and III." The goals of the program are to develop a coal–fired high performance power generation system (HIPPS) by the year 2000 that is capable of:

- >47% thermal efficiency (HHV)
- NO_x, SO_x and particulates $\geq$ 10% NSPS
- coal $\geq$ 65% of heat input
- all solid wastes benign
- cost of electricity 90% of present plant

Work reported herein is from Task 1.3, HIPPS Commercial Plant Design and Task 2.2, HITAF Air Heaters.

INTRODUCTION

The HIPPS power plants are subject to the same operational consideration and constraints as more traditional power plants. Occasionally, these considerations could lead to significant changes in the configuration initially identified as that which would have the best overall performance. The following sections deal with the operational aspects of the commercial plant design identified in Phase I. Changes to the Phase I configuration are suggested, although their effect on performance and cost have not yet been fully evaluated. A discussion or repowering including a conceptual design and heat and mass balance is included. This conceptual design would be subject to many of the configuration changes suggested for the Commercial Plant Design.

HIPPS OPERATIONS REVIEW

The HIPPS generating plant integrates a combustion gas turbine/HRSG combined cycle arrangement with an advanced coal–fired boiler. The unique feature of the HIPPS plant is the partial heating of gas turbine (GT) compressor outlet air using energy released by firing coal in the high temperature advanced furnace (HITAF). The compressed air is additionally heated prior to entering the GT expander section by burning natural gas. Energy available in the gas turbine exhaust and in the HITAF flue gas are used in a steam cycle to maximize energy production. The HIPPS plant arrangement is thus a combination of existing technologies (gas turbine, heat recovery boilers, conventional steam cycle) and new technologies (the HITAF design especially the heater located in the radiant section).

Background

Figure 1 illustrates the HIPPS concept as identified for the Commercial Plant Design of Phase I. The HITAF provides heat to the compressor outlet air using two air heaters, a convective air heater (CAH), and a radiant air heater (RAH). The HITAF is a slagging furnace which contains the radiant air heater, as well as waterwalls and steam drum for the high pressure (HP) steam system. Hot flue gas leaving the HITAF furnace passes over the CAH prior to entering the heat recovery steam generator (HRSG) #2. Hot exhaust gas from the gas turbine is ducted to HRSG #1 in a typical combined cycle arrangement. The HITAF, gas turbine and HRSGs are configured to achieve the required high efficiency of the HIPPS plant.

The intent of this review is to examine the proposed HIPPS plant from the standpoint of plant operation. The plant must be designed to be safe, reliable and operable in a manner similar to other utility power generating stations. With these goals in mind, the operations review is limited in scope to those plant features and components related to the new technologies and combinations of existing technologies in new arrangements, as follows:

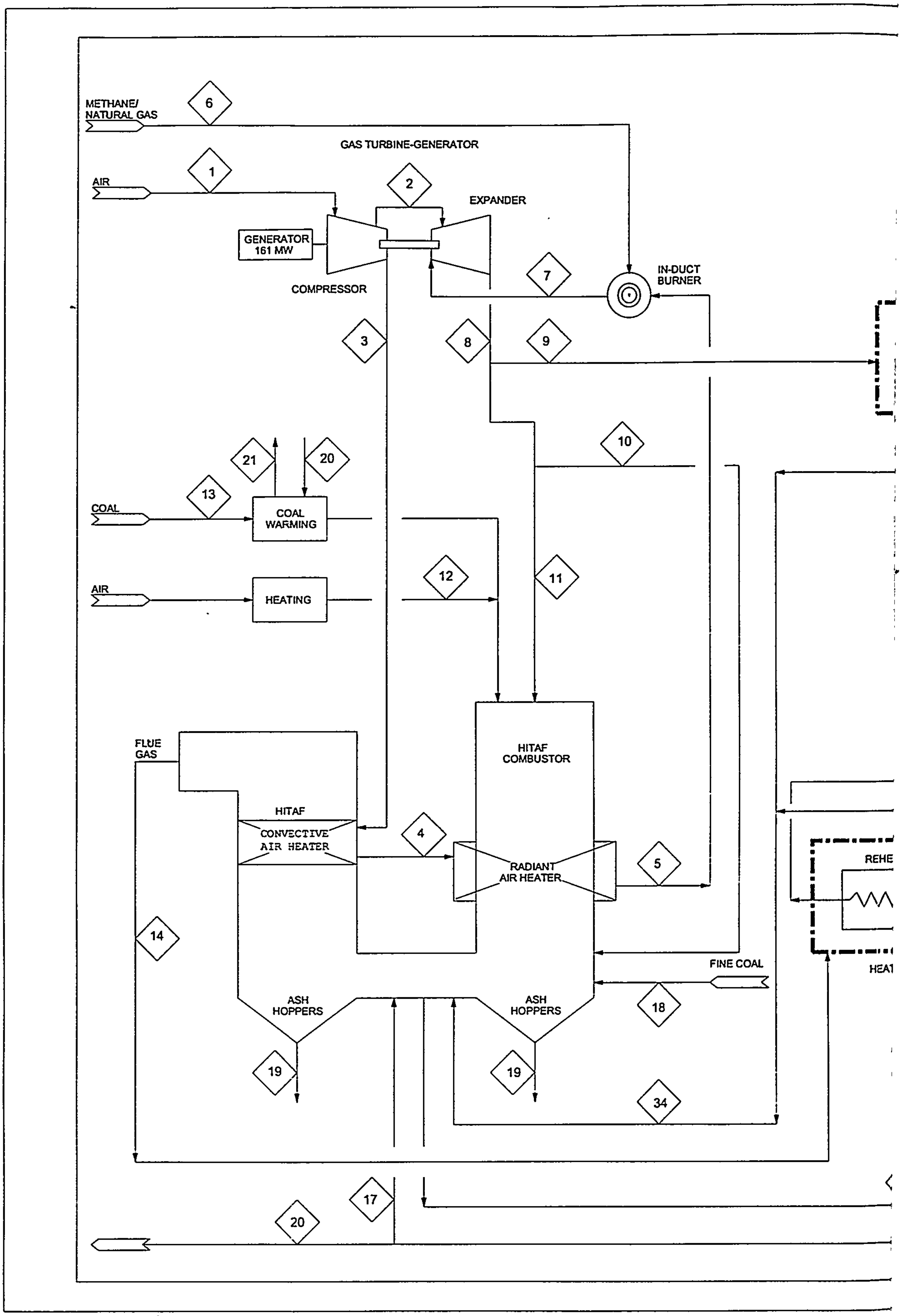
- Combustion gas turbine
- High temperature advanced furnace and the air heaters
- Heat recovery steam generators

The responses of these components to various HIPPS operating conditions, and, in turn, the restrictions on operations caused by equipment limitations are examined in the context of the overall HIPPS conceptual design. In particular, the HITAF CAH and RAH will be sensitive to operating transients, and attention to plant design and safety features is required. Other systems, such as the steam turbine, are considered only in relation to the three components noted above.

Operations Review

The steps taken for the HIPPS operational review are outlined below.

1. The screening review is for the HIPPS arrangement shown here as Figure 1, the Commercial Plant Design, presented in the Phase II & III technical proposal. The screening



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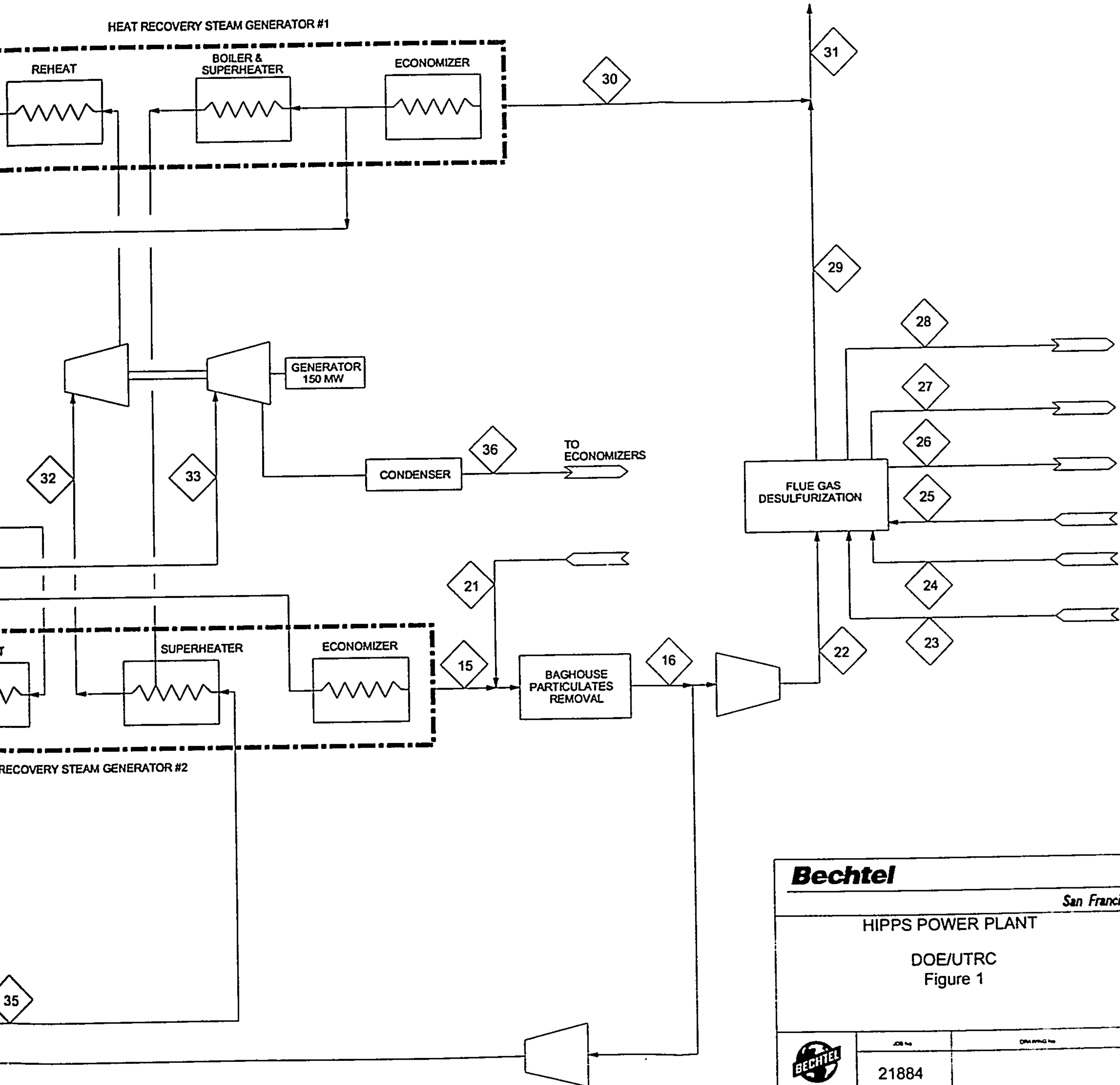

Bechtel
San Francisco
HIPPS POWER PLANT
DOE/UTRC
Figure 1
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21884P15 DGN /Fri Jan 26 1996 07 59 54 /PGLOGINO review's purpose is to highlight plant arrangement issues, which have a major impact on operability. Revisions to the original arrangement are suggested to improve operability. Operability will require further periodic reviews as the plant and HITAF conceptual designs are refined and detailed.

2. The review begins by selecting major issues relevant to the plant operation. These issues are related to maintaining material integrity or operational stability. The review does not consider protection of individual components where the protection is provided by the component manufacturer e.g., the steam turbine–generator is provided with a protection system. Also, conventional integration of several protection systems is not addressed, e.g., HRSG drum water inventory above the startup level is typically a permissive for gas turbine operation.

3. Plant operations from startup through normal operation and shutdown are considered. Transient conditions, from normal ramping through emergency conditions, are also reviewed. Some recommendations are made so that the HIPPS plant design allows operation with procedures similar to other power generation facilities.

A summary of the review's findings are provided, and recommendations presented for additional study.

Plant Arrangement

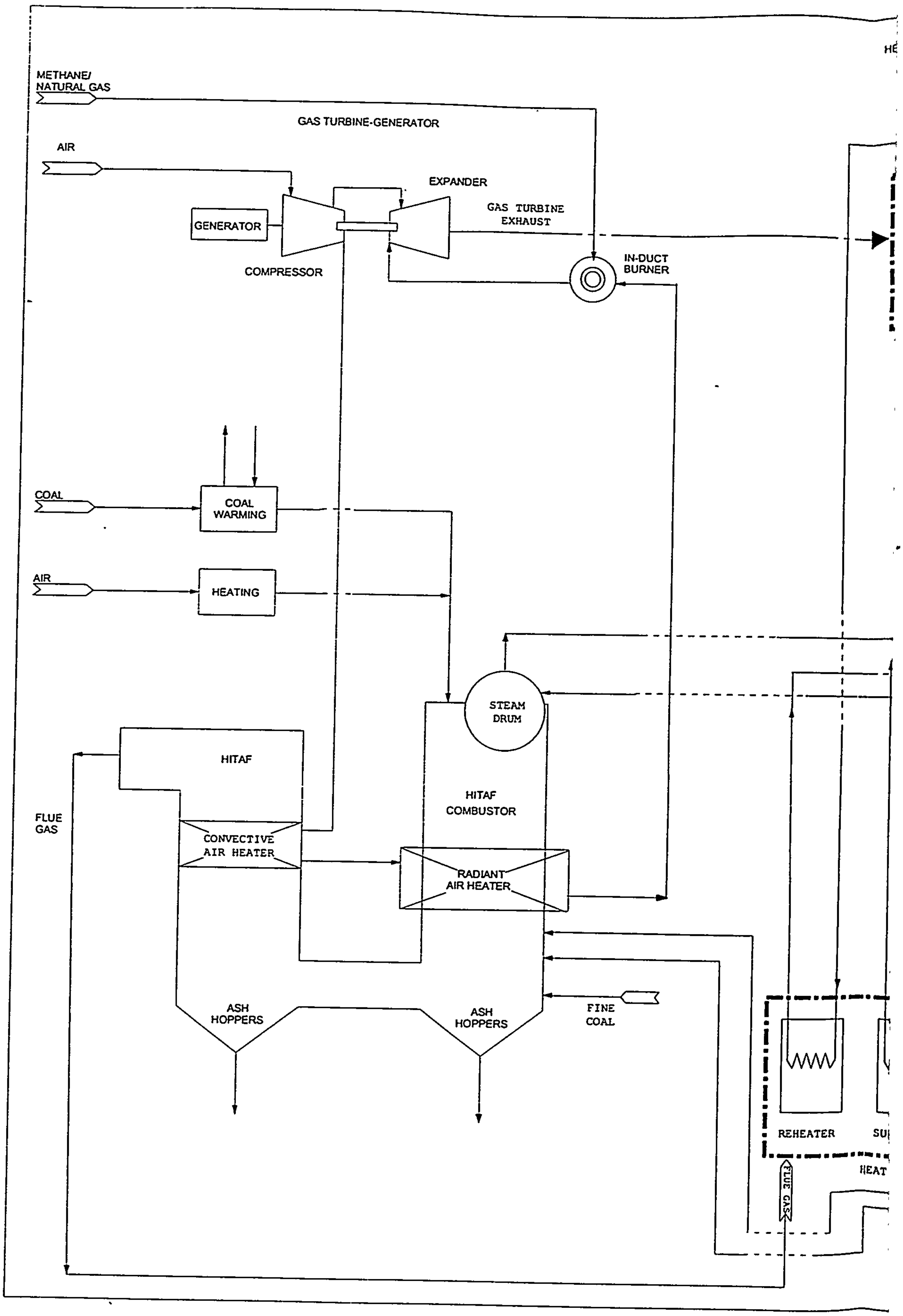
The HIPPS plant arrangement plays an important role in successful operation from normal startup through shutdown, as well as recovery from operating transients. The HIPPS arrangement, shown in Fig. 1, has been reviewed to screen major operability issues.

This initial screening looks at the interconnections of the major components, and examines the interactions which could cause difficulties with plant operation. Revisions of the arrangement are proposed to address operability issues, and also to potentially increase plant efficiency.

Changes of the arrangement are described below. Figure 1 shows the original HIPPS design. Figure 2 shows the revised HIPPS arrangement. As an energy and material balance was not prepared as part of the present task, flow stream numbers are not displayed in Fig. 2.

HITAF Combustion Air

The original HITAF arrangement in Fig. 1 uses the gas turbine exhaust as heat input (Stream 9) to the "clean" HRSG #1 and as a source of HITAF secondary combustion air (Stream 10). This arrangement reduces the number of HITAF forced draft fans, which is a simplification from typical pulverized coal–fired boiler arrangements. However, using the gas turbine exhaust as combustion air is undesirable because of the high draft loss in the HITAF (approximately 33 in. W.G. on the original HIPPS flow diagram). Gas turbines for combined cycle service normally are designed for back pressures on the order of 15 in. W.G. or less. Thus, diversion of gas turbine exhaust to the HITAF



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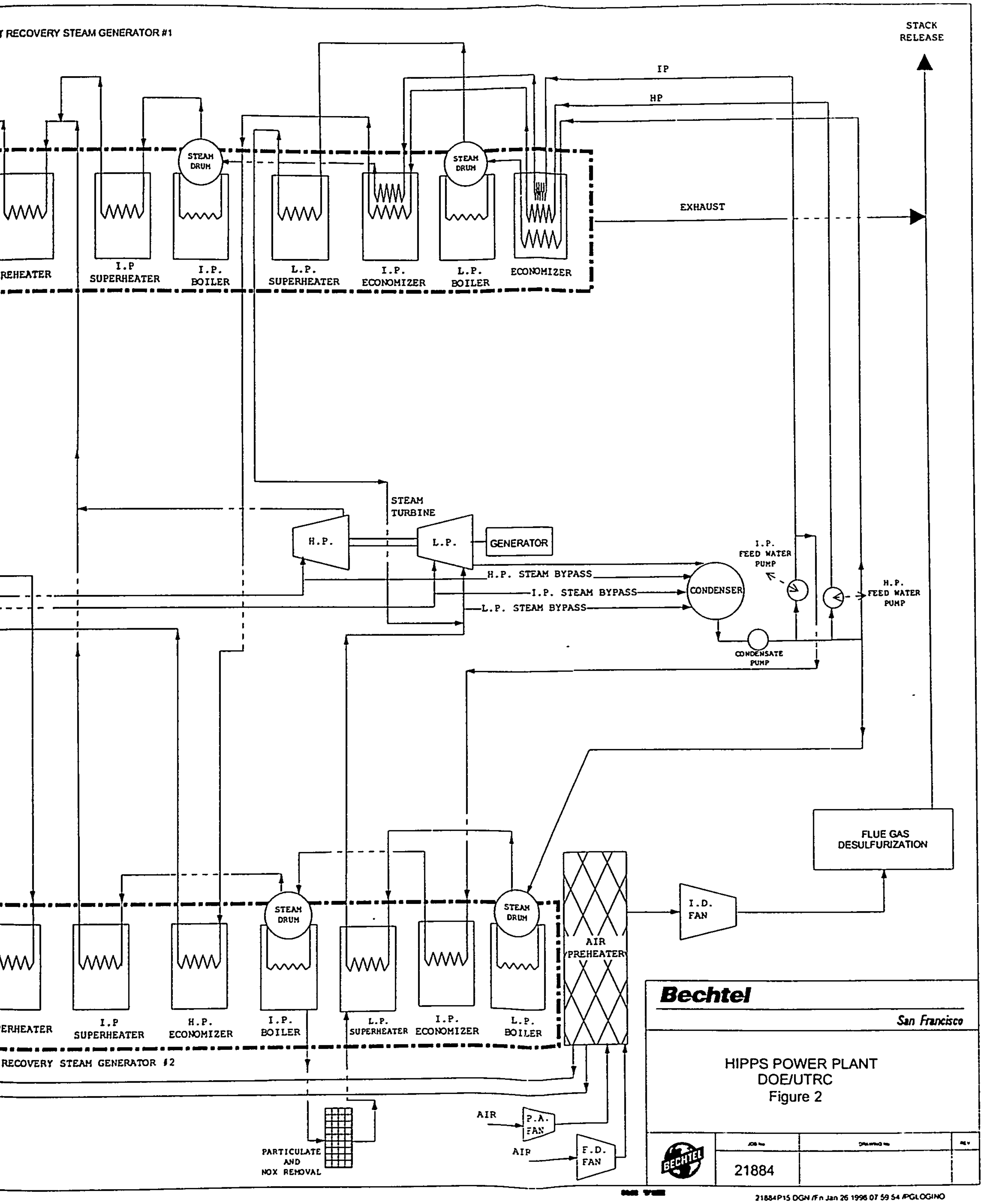

RECOVERY STEAM GENERATOR #1
STACK RELEASE
IP
HP
STEAM DRUM
STEAM DRUM
EXHAUST
REHEATER
I.P SUPERHEATER
I.P. BOILER
L.P. SUPERHEATER
I.P. ECONOMIZER
L.P. BOILER
ECONOMIZER
STEAM TURBINE
H.P.
L.P.
GENERATOR
H.P. STEAM BYPASS
I.P. STEAM BYPASS
L.P. STEAM BYPASS
CONDENSER
I.P. FEED WATER PUMP
H.P. FEED WATER PUMP
CONDENSATE PUMP
FLUE GAS DESULFURIZATION
I.D. FAN
AIR PREHEATER
STEAM DRUM
STEAM DRUM
REHEATER
I.P SUPERHEATER
H.P. ECONOMIZER
I.P. BOILER
L.P. SUPERHEATER
I.P. ECONOMIZER
L.P. BOILER
RECOVERY STEAM GENERATOR #2
PARTICULATE AND NOX REMOVAL
AIR
P.A. FAN
AIR
F.D. FAN
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HIPPS POWER PLANT
DOE/UTRC
Figure 2
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causes a mismatch in draft characteristics, and the gas turbine exhaust is more efficiently used when all of it is sent to HRSG #1.

Separating of the gas turbine exhaust from use with the HITAF will simplify plant operating procedures. For example, air flow regulation of the gas turbine exhaust to the HITAF (in the original arrangement) would require modulation of a control damper in the GT exhaust gas ducting to the HITAF. Also, if the gas turbine exhaust flows to the HITAF during startup, inadvertent changes in the gas turbine exhaust flow entering HRSG #1 can be avoided by disconnecting the flow as shown in Figure 2, and control actions in the HITAF will not lead to upsets in HRSG #1. In the revised arrangement, the gas turbine with HRSG #1 can be started independently of the HITAF with HRSG #2. This flexibility should simplify and stabilize plant startups and shutdowns.

The addition of primary air (P.A.) and forced draft (F.D.) fans for the HITAF is shown in Figure 2.

Steam Cycle Interactions

The original arrangement's steam system is another area of possible operations instability. Stability of steam temperature and pressure are of utmost importance to operations. The steam turbine is especially sensitive to changes in steam temperature, and strict limits are often imposed by manufacturers on the high pressure steam temperature rate–of–change. Also, pressure variations during transients could result in drum level instability (low/high water conditions).

For increased operations reliability and maximum flexibility, separating the steam systems of HRSGs #1 and #2 is desirable. Where interaction between the steam cycles is deemed appropriate (such as for higher efficiency, as described below), these interactions should be limited to the maximum extent possible to those areas least sensitive to temperature and pressure effects.

The original HIPPS arrangement shows interactions between HRSG #1 and HITAF/HRSG #2 at the high pressure (HP) superheaters and reheaters. In these locations, the gas turbine exhaust temperature is not high enough to generate the required 1000°F steam temperatures in HP and reheat steam supplied to the steam turbine. Moreover, interaction between the HP superheaters is not desirable because the steam turbine is sensitive to changes in HP steam temperature, and when the units are connected, there is a general need to match steam conditions between HRSG #1 and HITAF/HRSG #2 under all operating conditions. Also, with the original HIPPS arrangement, rapid transients in either HRSG #1 or HITAF/HRSG #2 would have a detrimental effect on pressure conditions in the HP steam system, which in turn could cause dynamic instability in HRSG #1 and/or HITAF steam drum levels. Interaction at the reheaters is probably necessary to maximize the efficient use of flue gas from the HITAF and gas turbine exhaust.

The HRSG economizers are areas with lower sensitivity to temperature changes than the superheaters. Thus, changes in the economizer operating conditions are more easily accommodated, and interactions between HRSG #1, the HITAF, and HRSG #2 economizers are operationally acceptable.

The revised arrangement eliminates interactions between the HRSG #1 and HRSG #2 superheaters, and adds interactions between the economizers and intermediate pressure (IP) and low pressure (LP) steam systems. The single superheater arrangement eliminates steam temperature matching issues in the HP steam system. Changing the steam cycle interactions to lower pressure steam systems should increase the dynamic stability of the plant.

Dynamic modeling will be required, later in the project, to verify the stability of the final steam cycle arrangement under all operating conditions.

Gas Turbine and HRSG #1 Efficiency

The type of fuel and the way it must be processed impacts the efficiency of HRSG #1 and #2. Flue gas sent to HRSG #1 is from the natural gas–fired turbine, whereas HRSG #2 uses coal derived flue gas. Flue gas from the combustion of natural gas is free of SO_2, and maintaining the flue gas exit temperature above dew point is not critically important. Flue gas from the combustion of coal will include a significant amount of SO_2, which will form sulfuric acid if flue gas temperature falls below the acid dew point. Thus, sufficient heat must be left in flue gas at the exit of HRSG #2 to assure that condensation does not occur on heat transfer surfaces, or in downstream components (such as ID fan and ductwork).

The revised HIPPS arrangement shown in Fig. 2 rearranges heat transfer modules to take optimal advantage of the non–corrosive flue gas flowing through HRSG #1. In particular, several economizer sections are added to HRSG #1 to maximize available heat use prior to exhausting to the stack. As discussed above, interactions in the area of the economizer are considered acceptable.

Major Operating Issues

In the review, the main design feature of concern in the HIPPS plant operations is the HITAF, and its function to preheat gas turbine combustion air. The HITAF is a slagging furnace and is designed to provide 65% of the total plant heat input. The remaining gas turbine heat input is provided by combustion of natural gas in a duct burner located near the gas turbine. The target efficiency of the HIPPS plant is 47% at ISO conditions, and to achieve this efficiency the temperature of gas turbine combustion air leaving the HITAF needs to be about 1700°F. To reach the desired temperature, the HITAF includes a convective air heater (CAH) followed in series by a radiant air heater (RAH). These increase the air temperature from 731°F (gas turbine compressor outlet temperature) to the required 1700°F (inlet conditions to the gas turbine combustion air duct burner).

A major concern with HIPPS plant operability is related to the ability of CAH and RAH materials to withstand temperature transients. The RAH in particular will operate at very high temperatures due to the high film coefficient on the air side of the RAH (compared to waterwalls in a conventional boiler).

Another area of concern is the steam cycle. The HIPPS design concept combines a high thermal mass system, the HITAF and HRSG #2, with a low thermal mass system, the gas turbine and HRSG #1. The stability of these systems during transients is very important to the success of the design concept.

Air Heater Materials

The RAH and CAH are critical components of the HIPPS plant and are expected to be the new design items with the most impact on plant operation considerations.

The RAH is planned to be constructed of alloy metals with a ceramic coating, or refractory tile covering, to protect the tubing from overheating. Rapid temperature fluctuations are likely to cause problems with ceramic coatings or tiles, especially at the interface with the alloy tubes.

High temperature in either the coating or tubes during temperature excursions or surges, whether rapidly or slowly occurring, may cause overstress due to thermal growth or lowering of allowable stress values.

Due to the low heat transfer characteristics of air (as compared to boiling water), RAH materials will operate at higher temperatures than waterwall boiler tubes in a typical steam plant boiler, which may leave a low margin of difference between operating temperatures and maximum allowable temperature limits. In addition, because there is no inventory of heat exchange fluid in the HITAF system (i.e., gas turbine combustion air), as is provided by water in the boiler drum and waterwalls of a conventional boiler, the RAH will be very sensitive to changes in gas turbine combustion air flow.

Similar issues could also exist for CAH tubing (although ceramic coatings are not required for the CAH). Plant design and operating procedures will address the limits on RAH and CAH temperatures and rate of temperature changes.

Steam Cycle Issues

The HIPPS plant is configured to use heat from gas turbine exhaust and from the HITAF flue gas to obtain throttle conditions of 2400 psi/1000°F/1000°F in a reheat steam cycle. To achieve these steam cycle operating conditions for the revised arrangement (Fig. 2), HRSG #1 (gas turbine exhaust) interacts with HRSG #2 (HITAF flue gas) at the reheaters. These interactions are required because of the low gas turbine exhaust temperature, which is insufficient to achieve the required 1000°F steam conditions. Other interactions also occur at the LP steam system, IP steam system, and economizers to improve overall plant efficiency.

One consideration for all steam generation systems is the prevention of overheating in superheater and reheater tubes. The temperature of gas turbine exhaust will not be high enough during transients to damage tubes in properly designed exhaust ducting, superheaters and reheaters. However, flue gas at the outlet of the HITAF could damage reheater and superheater tubes. Protection of HITAF reheater and superheater tubes can be covered by typical fired boiler startup and operating procedures not unique to the HIPPS plant.

The gas turbine and HRSG #1 and HITAF and HRSG #2 have different thermal operating characteristics. The gas turbine and HRSG #1 arrangement is a low thermal mass system and is capable of rapid startups and load changes. The HITAF and HRSG #2 arrangement is a high thermal mass system and will not be as responsive to changes in heat input or changes originating in the steam piping system. To accommodate these differences, plant design and operating procedures will be required to assure capability to independently remove heat from both HRSGs #1 and #2 during periods of transition or mismatched operation.

Plant Startup

Plant startup is a process of alternately ramping up the gas turbine and HITAF to an initial holding condition. Subsequent loading of the units would be in parallel, with appropriate holding points during the ramp ups for temperature equalization, soak periods, and so forth as required by the equipment manufacturers. An important startup consideration is to limit thermal stress in the RAH and CAH during their heat up. Due to the difference in startup rates between the gas turbine and HITAF, steam flows between these components will be mismatched, and a system for removing heat from the HITAF and HRSGs (via the steam system) will be required.

With the HRSGs and HITAF filled with water and ready, the HITAF would be placed in service using natural gas fired startup burners. The HITAF would be fired at a rate to slowly raise the temperature of the RAH and CAH. Following the initial gas firing of the HITAF, the gas turbine would be started. The HITAF firing rate would be held constant while the gas turbine ramps to the first holding level for warm up of HRSG #1.

As HRSG #1's pressure increases, a majority of steam is generated in the low pressure portion of the system. The LP steam turbine, however, cannot accept steam until the HP steam turbine stop valves are opened, which does not occur until HP steam from the HITAF has adequate superheat (generally 90–110°F). Any steam generated in HRSG #1 prior to steam turbine operation is bypassed to the main condenser in a steam bypass system. Figure 2 shows these steam bypasses.

At the end of the first HRSG #1 holding level, the gas turbine and HITAF will ramp up at a rate consistent with maintaining RAH, CAH, and HP drum stress levels within allowable limits (IP and LP drums are generally thin walled and not subject to overstress during startup). During low load gas turbine operation, combustion air flow is reduced by modulation of the gas turbine inlet guide vanes, hence air flow through the CAH and RAH would be below design values. At this time, however, HITAF heat input is low and sufficient gas turbine combustion air will be flowing through the CAH and RAH to prevent overheating. The steam turbine is not yet in operation, so HITAF and HRSG #2 steam would be bypassed to the main condenser.

The gas turbine and HITAF will increase load in parallel until HITAF HP steam is admitted to the steam turbine. The steam turbine requires a ramped warm up period, and during initial steam turbine operation, excess steam generated in the HITAF and HRSGs will be bypassed to the main condenser. Loading would continue up to full load, with the ramp rate limited by either steam turbine heat up rates or HITAF heat up rates.

The startup sequence will be semi–automated so that plant operators do not have to manually balance the operation of the numerous plant components. Instrumentation in critical areas will monitor ramp rates. Predetermined hold points, based on design input and data collected during initial commissioning, will be maintained automatically. Typical plant instrumentation used in modern power plants is sufficient for automatic startup sequencing, with additional temperature elements in critical areas of the RAH and CAH.

An item which may cause problems during startup (and normal operation) is unbalanced combustion air flow in various portions of the CAH and RAH. The present (for Fig. 1) heat balance shows very low pressure differentials between the inlet and outlet of these heaters, which could result in uneven flow distribution. The potential problem would be worse during startup where air flow through the CAH and RAH is reduced by gas turbine operation at low load. As a minimum, instrumentation is required to verify adequate air flow and heat removal in all areas of the CAH and RAH, otherwise local overheating may lead to material failures. The location of the thermocouples will be determined by analysis and model testing of the CAH and RAH. Modeling and tests may also show that during startup, the gas turbine may be raised more rapidly so that full load rate combustion air is flowing to the HITAF air heaters. If this is the case, the steam bypass system would dump generated steam prior to steam turbine operation.

The steam bypass system consists of piping, isolation valves, control valves, desuperheaters, and discharge connections in the main condenser. Steam bypass system sizing will depend on the final plant process design, the startup sequence, the need to minimize the impacts of transients during operation at load, and the desired operating flexibility for integration of the HIPPS plant with the local utility grid. The steam bypass system will be sized to accommodate the maximum bypass flow requirements, as determined by reviewing each operating mode requiring bypass operation. Due to the very high energy in bypass steam, the impact of the steam bypass system operation on the main condenser design criteria and sizing must be considered.

Normal Plant Operations

During normal plant operation at design conditions, the gas turbine, HITAF, HRSGs, and steam turbine are operating at full load. HITAF heat input will normally be controlled by the HP superheater outlet pressure. The gas turbine will be base loaded and controlled by gas turbine inlet temperature limits. HRSGs #1 and #2 will be in sliding pressure operation. Pressures in the HRSGs will not be controlled, and will follow conditions established by the gas turbine and HITAF control systems.

Reduced Load Operation

The conceptual commercial plant design includes a requirement for the plant to be able to cycle down to 50 percent capacity. Initial turndown of the plant power output would be most easily accomplished by reducing load on the gas turbine. Gas turbine combustion air would remain at full load flow until modulation of the compressor inlet guide vanes (IGV) occurs. At this point, HITAF load would be reduced to match lower air flow through the CAH and RAH caused by gas turbine IGV modulation.

Further load reduction would be accomplished by downward load ramping of the HITAF and gas turbine. The maximum combustion air temperature exiting the RAH will be maintained to preserve overall plant efficiency. The operation will be automatically controlled at a rate to prevent thermal shocking of critical components. An operator would select the target load and ramp rate (within prescribed ramp rate limits). Instrumentation may be used during the load change to verify stress levels are within allowable limits (similar to instrumentation to regulate plant startup).

Rapid Load Changes

The HIPPS plant will allow for rapid load changes. However, the magnitude of incremental changes will likely be limited to avoid thermal stresses in equipment and materials. For the CAH and RAH, rate and degree of load changes will be partly dependent on changes to the gas turbine compressor air flow. Load changes, which do not modulate the gas turbine IGVs, will not substantially change the flow or temperature of gas turbine combustion air and ramp rates within this range can be achieved rapidly.

Rapid load reductions will be accomplished by ramping the HITAF and gas turbine simultaneously. To enhance stability of steam conditions in the HP steam system during rapid load reduction, modulation of the steam turbine control valves will be integrated into the control system to minimize HITAF HP steam system pressure swings. Excess steam generation during rapid load reductions will be bypassed to the main condenser. Steam temperature drops due to control valve throttling may limit the range of such load reductions.

Rapid load increases are subject to the same restraints covering startup, and the responsiveness of the plant will depend on the plant output at the start of the load change. In general, the higher the plant output, the faster the expected ramp rate.

The ability of the HITAF and steam systems to accomplish rapid load changes and determining ramp rate limits will require additional study. Dynamic modeling will help clarify the interactions between plant components and determine ramp rate limits and interactive control schemes that assure stability.

Normal Shutdown

The same issues discussed for normal plant startup also apply to plant shutdown. The shutdown procedure is essentially the reverse of the normal startup procedure.

Emergency Conditions

Emergency conditions are defined as abnormal operating conditions, which could cause personnel injury and/or equipment damage. While plant equipment is designed with sufficient margin to withstand many excursions beyond normal limits, instrumentation is required to monitor critical plant parameters, and to initiate control actions to protect the equipment from damage or failure during extreme excursions. For vessels and piping, safety valves are provided as a final pressure relief protective feature. This review considers HIPPS plant crucial components unique to the plant. Other equipment and processes, which are typically of power plants and usually controlled by design codes and standards, are not included in the discussion of emergency conditions.

The HIPPS plant control alarm system will be designed to notify plant operators when operating conditions exceed normal limits. Alarm set points are chosen just beyond the normal operating band to prevent nuisance alarms, but also below the maximum limits to allow the operators the time to assess the condition and to take corrective action. Trip set points are the maximum limits to initiate automatic, emergency actions to prevent further failures. The most drastic emergency action is tripping (turning off) a piece of equipment as fast as possible, without regard to normal shutdown procedures. For the HIPPS plant, tripping the following items would be major events.

- Gas turbine
- HITAF
- Steam turbine

In addition to abnormal operating conditions in the units, their tripping can be required because of conditions in other equipment such as the:

- Draft fans (ID, FD)
- Gas turbine auxiliary equipment: Lube oil, control oil, fuel systems, etc.
- Steam turbine auxiliary equipment (similar to gas turbine equipment)
- Balance of plant equipment, such as feed water and cooling water pumps, vacuum equipment, etc.
- Instrument failures related to the above items

An emergency trip of one the three major pieces of equipment listed above could require an emergency plant shutdown. The following paragraphs consider the consequences of major equipment trips and design features to prevent or limit damage.

Gas Turbine Trip

Gas turbine protection includes a control system provided by the turbine supplier, and for most events, damage to the turbine is unlikely. The main concern following a gas turbine trip is the loss of combustion air flow from the turbine to the CAH and RAH. Following a gas turbine trip, the HITAF will also need to be tripped, or the firing–rate rapidly reduced to decrease heat input to the CAH and RAH. In any scenario, residual heat in the HITAF can cause overheating of the air heater materials, unless an emergency source of air is provided to cool the CAH and RAH.

The extent of cooling and source of cooling air will be the subject of further studies to determine maximum temperature limits and allowable cool down rates in the CAH and RAH under various conditions.

Several options for providing emergency cooling to the CAH and RAH are available. For example, warm air from the HITAF primary or forced draft fan discharge duct could be one source of cooling air. The fans are not available if there is a blackout, and a separate, limited capacity fan, driven by a reliable power source is anticipated. A source of heat to warm the air supply may be required to prevent shocking the CAH or RAH with air that is too cool compared to the air heater material temperatures.

For a scenario where the HITAF is operated at reduced load following a gas turbine trip, future work needs to determine the effect of losing heat input to the HRSG #1 reheater and economizers. There are concerns that loss of heat input to the HRSG #1 reheater may cause water induction, which precludes continued steam turbine operation. Similarly, loss of heat input to the HP steam system economizers may cause unacceptable low feed water temperatures for the HP drum.

HITAF Trip

A HITAF trip, for example, caused by failure of the ID fan, will cause a pressure decay of the HP steam system. Operation of the steam turbine is possible for a short time with the significant residual heat present in the HITAF. Unless the problem is rapidly solved and the HITAF placed back on–line, the steam turbine will be tripped and further cool down accomplished using the steam bypass lines to the main condenser.

Following a HITAF trip, the gas turbine can remain on line with HRSG #1 steam dumping to the main condenser until the HITAF returns to operation.

Loss of External Electrical System –Load Rejection

The HIPPS plant design will include the capability to shut down, without equipment damage, following a loss of the external electrical system, termed a site blackout.

All normal AC power will be lost during the blackout, and a backup power supply is needed for the following critical items, as a minimum:

- the instrument and control system.
- the lube oil systems for bearings of the gas turbine generator, steam turbine generator, and large mechanical equipment (such as fans, boiler feed pumps where forced lubrication is provided in the design).
- the feed water supply to the HITAF for residual heat removal.
- the emergency cooling air supply to the CAH and RAH.

Emergency power can be provided by a number options, or combination of options, including:

Batteries: Batteries can provide several hours of power to a minimal number of plant components. At other power plants, the control system and DC driven emergency lube oil pumps use batteries for emergency power. The battery power provides lubrication to turbine generator and other equipment bearings until the equipment stops rotating. Battery power is also used to operate the control system. The batteries are recharged with AC inverters, and eventually require re–establishment of normal AC power from the external electrical system for recharging.

Standby diesel generator: A standby diesel generator will supplement the battery system and provide the additional power necessary for other loads, such as emergency feed water to the HITAF and emergency cooling air to the CAH and RAH. Diesel generators are very reliable, and well suited to the type of occasional service required by emergency conditions.

Steam turbine drives (emergency feed water pump and cooling air supply fan): Steam turbine drives may be provided for emergency feed water and cooling air service. Steam for the drives could be generated from the residual heat in the HITAF. The auxiliary equipment for the steam turbines use DC motors with backup power provided by batteries (for operation under site blackout conditions, similar to steam and gas turbine lube oil systems). The decision to use steam turbine drives rather than AC drives in combination with a standby diesel generator will be studied as a site specific design issue.

Areas for Further Study

Plant Arrangement

Plant arrangement is an important element of the process to optimize plant generating efficiency. While further work is needed to be more definitive, the HIPPS plant arrangement may need to be modified to improve the reliability and flexibility of operations. Examples of operational versus efficiency trade–offs are noted below.

- A design objective for the HIPPS plant is part load operation down to 50% output. Studies under part load conditions are required to determine the relationship during part load operation between the gas turbine and HRSG #1 and HITAF and HRSG #2. Changes to the plant arrangement may be made to support part load operations flexibility.
- Industrial frame gas turbines with exhaust temperatures in excess of 1,000°F are currently available. Combined cycle plants use these higher gas turbine exhaust temperatures to achieve 1,000°F throttle steam and 1,000°F reheat steam conditions. For the HIPPS plant, this capability would allow separation of the HRSG #1 and HRSG #2 reheaters, which could provide improved simplicity of the operation.

The use of an advanced gas turbine in the HIPPS plant arrangement should also be considered to increase the overall efficiency. For example, gas turbines with inter–stage cooling are being developed to provide improved gas turbine output, higher gas turbine efficiency, and higher overall cycle efficiency.

HITAF Combustion Air

The separation of the gas turbine exhaust from the HITAF is presently considered appropriate, due to the draft differences between these two heat sources. Additionally, separation of the gas turbine exhaust provides more flexible plant operation. Future development of the HIPPS plant may also examine lowering the operating pressure in the HITAF furnace to better match the draft characteristics of HRSG #1 to those of the HITAF.

Plant Efficiency

Further refinements of the plant arrangement and design of heat transfer modules in the HRSGs may increase full load efficiency. Such refinements will need to consider steam cycle interactions, as previously discussed.

Air Heater Materials

Studies and testing of air heater materials will be done to further define the operating constraints for the CAH and RAH. The capability of these materials to survive thermal shock, as well as high temperature excursions, will be determined. This information is vitally important to plant design in the following areas:

- Allowable heat–up/cool–down rates during normal startup and shutdown. The rates will be used in the HITAF operating procedures, and may impact the gas turbine ramp rate relative to the HITAF ramp rate. The information will also be used to design the emergency cooling air system.
- Consequences of loss of all gas turbine combustion air following a gas turbine trip. The conditions for operation of the emergency air supply to the CAH and RAH will use this information to determine:

1. How soon emergency air is required following loss of gas turbine combustion air;
2. Required flow of the emergency air; and
3. Minimum temperature of the emergency air supply.

- Consequences of unbalanced air flow in the CAH and/or RAH. The need for balanced flow in the CAH and RAH during transients and normal operation will depend on the maximum temperature capability of the CAH and RAH materials. Modeling of CAH and RAH operations will predict material temperature profiles at various physical locations in the heaters. Models and tests will determine if higher CAH and RAH pressure differentials are required to assure a more uniform flow distribution in the heaters and to prevent areas of low flow and corresponding higher metal temperatures. If higher differentials are required, this may reduce plant generation capacity. The data can also be used to define critical areas of the CAH and RAH for installation of temperature elements.

Steam Bypass System

A steam bypass system is required, as a minimum, to start the plant. The bypass system would serve several purposes, as follows:

- Provide a means to remove heat from the HITAF and HRSGs during plant startup and shutdown.
- Provide a means to remove heat from either the HITAF or HRSGs during operating transients.
- Provide a means to remove heat during rapid load changes accomplished by throttling the steam turbine control valves.

Emergency Conditions.

- Gas Turbine Trips. Work is needed to determine the emergency cooling air system requirements and its components. Modeling and further design will define the impact on the HITAF reheater and economizers caused by rapid cooling of HRSG #1 following a gas turbine trip.
- HITAF Trip. Conditions in the CAH, RAH, HRSG #1, and sizing of the steam bypass system for continued operation of the gas turbine following a HITAF trip all will be reviewed. The ability to restart the HITAF with the gas turbine operating at full load will be examined.
- Electrical Load Rejection. Emergency power facilities will be determined for the detailed plant design.

Dynamic Modeling.

The complexity of interactions between the gas turbine, the HITAF, and the HRSGs warrants dynamic modeling of the integrated operation of the plant. Data produced from the modeling will confirm the plant arrangement, or show where changes are needed to provide an acceptable level of operating stability. Modeling would include operations during startup, normal load changes, and rapid load changes. The dynamic modeling data will also be used for detailed design decisions and for preparation of plant operating procedures.

REPOWERING

The following section describes the preliminary investigations of repowering coal–fired stations. In place of the heavy frame machine used in the commercial plant design of Phase 1, a new generation aero–derivative gas turbine in combined cycle and Humid Air Turbine (HAT) configurations is used. An 100 MW + industrialized version of the Pratt & Whitney 4000 aero engine is the basis of intercooled gas turbine cycles, which use both coal and natural gas to heat the gas turbine combustion air. Unlike the commercial plant, however, the manner in which the heat from the intercooler and the turbine exhaust is used in repowering depends on the desired steam conditions and power level. Information on this approach was presented in Ref 1 and is briefly reviewed here to establish the parameters important for repowering applications. The preliminary results of an evaluation of a site specific application are presented. The considerations discussed in Sections 1 and 2 are equally applicable to repowering.

Introduction

The UTRC Combustion 2000 team approach to the HIPPS was described in previous Sections. The high performance of this system, 47.3% efficiency (HHV) with conventional sub–critical steam systems, led the investigators to consider applying the HIPPS/HITAF approach to the repowering of existing steam stations. Repowering is not a new concept. A number of older stations have been upgraded, usually by the addition of a natural gas–fired, modern, frame or aero–derivative gas turbine with the waste heat used either as preheated combustion air in a new gas–fired boiler or to heat feedwater/superheat steam for the gas–fired boiler. The steam turbine system was reused, although often with some modification.

Steam station sizes from 90 to 200 MW, with steam conditions ranging from 1250 psi/950°F non–reheat to 2400 psi/1050 F/1050°F reheat, were investigated. The preliminary results indicate efficiencies in the 42–47% range (HHV) are possible. While the cost of electricity is application specific, it should be significantly less than the original station at emission levels which are one–quarter or less than current New Source Performance Standards.

Repowering Configurations

There are several possible ways to repower an existing plant. A study carried out by Bechtel for EPRI (Ref. 2) described five approaches currently used for repowering:

- Substitute HRSG
- Supplemental HRSG
- Hot Wind Box
- Feedwater Heating
- Station Repowering

The first two use a HRSG to replace or add capacity to the existing boiler. The hot wind box uses the turbine exhaust as preheated combustion air. The feedwater heating version uses the exhaust in feed heaters and steaming economizers, while the fifth approach completely replaces the plant, using only the location and some infrastructure. All of these approaches were considered for the HIPPS/HITAF investigation. The most attractive used a variation of hot wind box and feed heating combined. A generic schematic of the repowering scheme investigated is shown in Fig. 3. The gas turbine in Fig. 3 is an advanced industrialized version of the P&W PW4000. This engine, previously been described in Ref. 1, was used as the gas turbine portion of a repowering system parametric investigation.

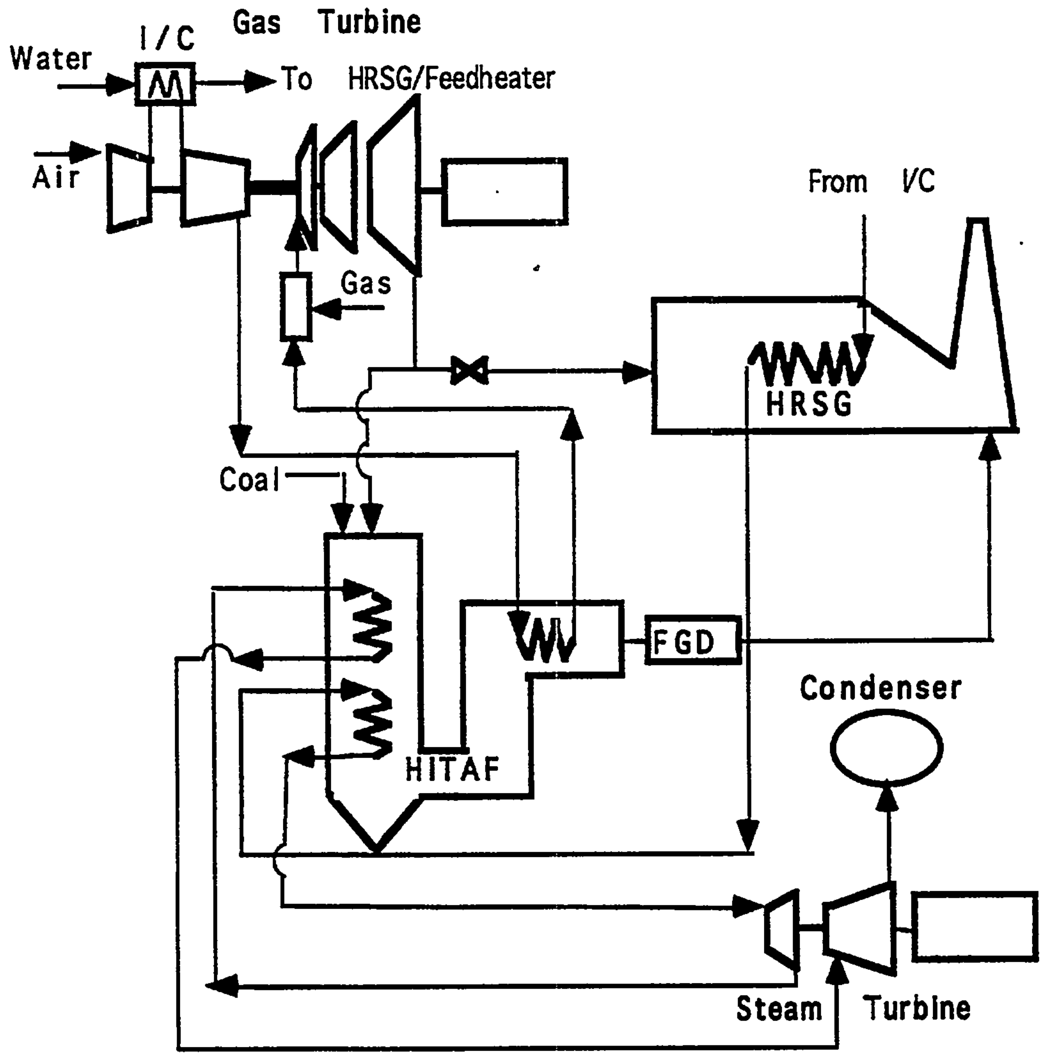

Figure 3. Generic Repowering Diagram.

Unlike the HIPPS/HITAF commercial plant, the repowering application investigated does not use a radiator section in the HITAF to preheat the compressor discharge air. Rather, all the preheat is done in the convection section. The main reason for this change was one of development time; i.e., the radiator section of the HITAF represents the highest technical risk in the system. By eliminating this section, the HITAF concept could have an earlier introduction in repowering applications by using current technology in the heat exchangers. One of the goals of the investigation was to evaluate the potential performance of this simplified approach over a range of heat exchanger and steam turbine conditions. Flexibility was obtained by varying the ratio of turbine exhaust used as preheated combustion air to that used to heat boiler feedwater. The valve, shown in Fig. 1, is a "virtual " valve shown in the drawing only to indicate the ability to change flow ratios. The ability to split the GT exhaust flow is key to high performance in the repowered system.

Parametric Studies

Repowering of existing steam cycle equipment with aero–derivative engines offers a potential first step toward achieving the HIPPS program goal of replacing natural gas with coal in high–performance power generating systems, while using currently available technology. Before designing a site specific plant, a preliminary assessment was made to identify the range of sizes and steam conditions that could be handled by one FT4000 IC. This was done by defining the largest size, or "base," repowering system, the one that used all the GT exhaust as combustion air. The exhaust was then bypassed around the boiler to HRSG's that actually were used to heat feedwater. In this manner, smaller boilers were identified; the range of sizes that one engine could handle varied from less than 90 MW to over 220 MW.

The FT4000 IC is a high pressure ratio, intercooled gas turbine, having a high simple cycle efficiency (>45% HHV) and is well suited to provide preheated air to a coal fired boiler as was shown schematically in Fig. 3. The key part of the system shown in Fig. 3 is the gas turbine combustion air (HPC discharge air) convective preheater, which makes it possible to utilize a portion of the coal heat in the gas turbine cycle. This preheater, in effect, takes the place of a regenerative heat exchanger that would be used with a conventional regenerative gas turbine cycle. In the case of the FT4000 IC, the high pressure ratio design does not produce an exhaust temperature high enough to make the use of a regenerator attractive. The exhaust temperature, however, makes it ideal as the combustion air for a coal fired boiler.

The boiler in Fig. 3 is assumed to be of conventional design in regard to steam generation, superheat and reheat provisions. The absence of a regenerative air preheater allows the exhaust to be used for feedwater heating and eliminates the need for extraction heaters. The gas turbine combustion air heater (referred to as the convective heat exchanger in the HITAF system) has a design HITAF exhaust stream inlet temperature of 1800°F. This temperature was selected for the HITAF system since it is ideal for non–catalytic NO_x reduction and would permit reasonable operating times between removal of deposits to avoid densification.

To make this concept applicable to existing steam equipment, a flow diverter is included, allowing a portion of the gas turbine exhaust to bypass the furnace. That bypass gas is sent to a HRSG, where its heat content is used to heat feedwater for the steam cycle. Figure 4 shows the range of steam cycles that can be accommodated by varying the amount of exhaust gas sent to the HRSG. The values considered are intended to cover the range of probable steam conditions that would be encountered in a repowering operation. They range from a moderately high– performance 2400 psi/1050°F/1050°F reheat cycle to a 1250 psi non–reheat cycle. The corresponding efficiencies for each cycle are given in

Fig 5, as a function of the fraction of exhaust gas sent to the HRSG. The output for each system includes the gas turbine contribution, which is constant at approximately 110 MW for the cases presented. At the minimum convective air preheat outlet temperature of 1300°F, the heat required to achieve the desired GT firing temperature is reduced by approximately 25%. This forms the basis for the desirability of the cycle, which is the reduction of natural gas consumption and its replacement by coal utilization.

As the fraction of exhaust gas sent to the HRSG is increased, the system would be expected to approach a normal combined cycle in appearance. This occurs because the fraction of turbine exhaust for boiler combustion air decreases, resulting in a decrease of steam cycle output approaching levels that would be expected in combined cycle. However, the low gas turbine exhaust temperatures will not support the steam temperature levels used here. Thus, there is a practical limit to the bypass fraction, both from a thermodynamic as well as steam turbine size standpoint.

The primary results indicate that as the fraction of gas turbine exhaust bypassing the boiler is increased, steam power decreases, efficiency increases and so does the fraction of heat from methane.

In the commercial plant HITAF design, air from the convective heat exchanger is heated further in the radiant section of the furnace. This additional air preheat allows the performance goal to be reached while burning only 35% methane. For the repowering configuration, increasing the duty of the convective heat exchanger was investigated. For the 2400 psi/1050°F/1050°F steam cycle the effect of increased air heater exit temperature on steam cycle output is shown in Fig. 6. At full flow to the boiler (no flow to the HRSG), there is virtually no change in steam cycle output as air preheat temperature is increased. This is the result of an increase in oxygen content of the gas turbine exhaust, which allows more coal to be burned making up for the increased duty of the convective exchanger. As gas turbine exhaust flow to the HRSG is increased, the heat required by the convective exchanger becomes a more significant part of the total and steam power drops off somewhat with increased air preheat temperature. Both output and overall efficiency for the system with the 2400 psi/1050°F/1050°F steam cycle are virtually unchanged with increased combustion air preheat. Analysis indicates that with an air preheat temperature of 1500°F, an efficiency of 47% can be achieved at a methane use rate of 35%. (This coincides with the Commercial Plant Design results.)

Comments on Repowering

The use of the aero–derivative engine with the HITAF offers higher efficiencies than realizable with heavy frame machines. The technology advancements in materials and cooling techniques developed for aircraft engines shows up more quickly, giving the aero–derivative engine even more of a potential advantage in the future.

When used in repowering, systems based on using aeroderivative engines could add as much as 10 or 12 points (around 30–35%) to the efficiencies of older steam stations. The repowering configurations do not have to use the radiative heat exchangers, somewhat simplifying the HITAF, and should result in systems having lower overall cost of electricity than current repowering alternatives. These advantages are realizable with coal providing 60–70% of the cycle input heat requirement.

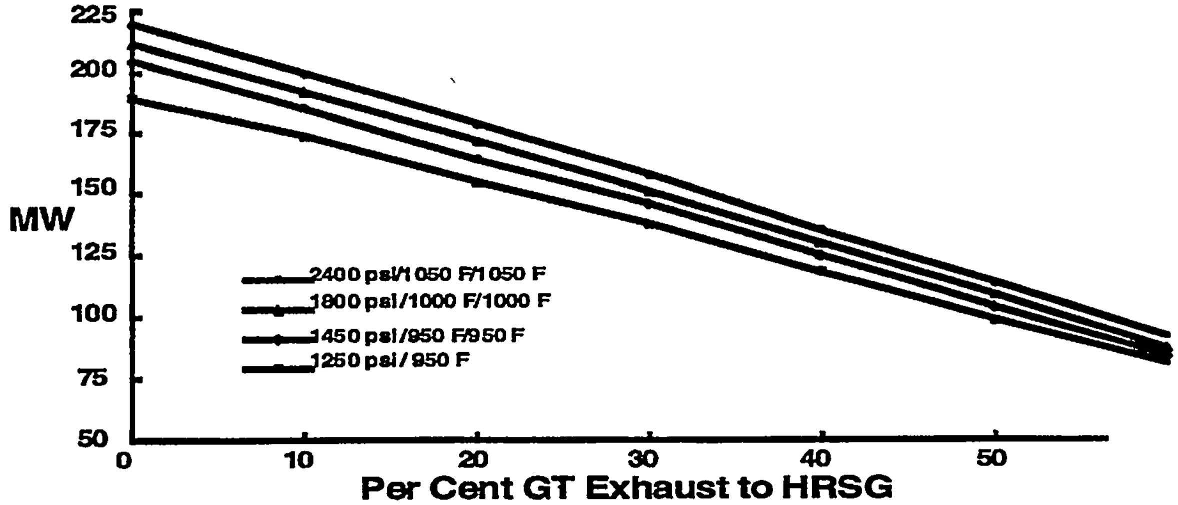

Figure 4. Steam Cycle Output.

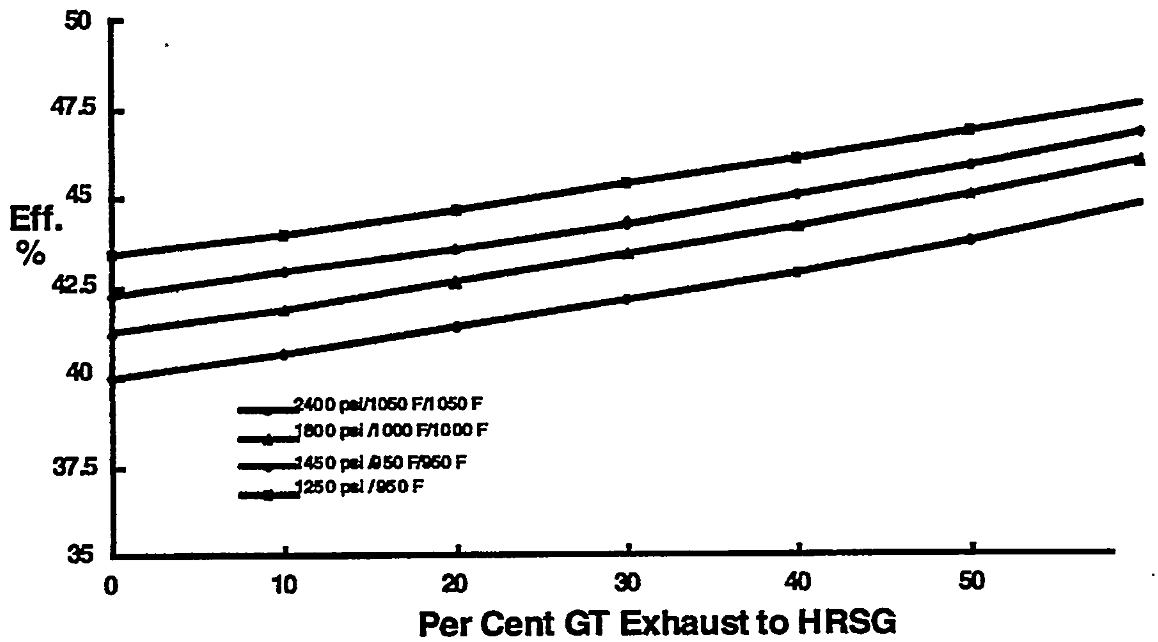

Figure 5. Overall Cycle Efficiency.

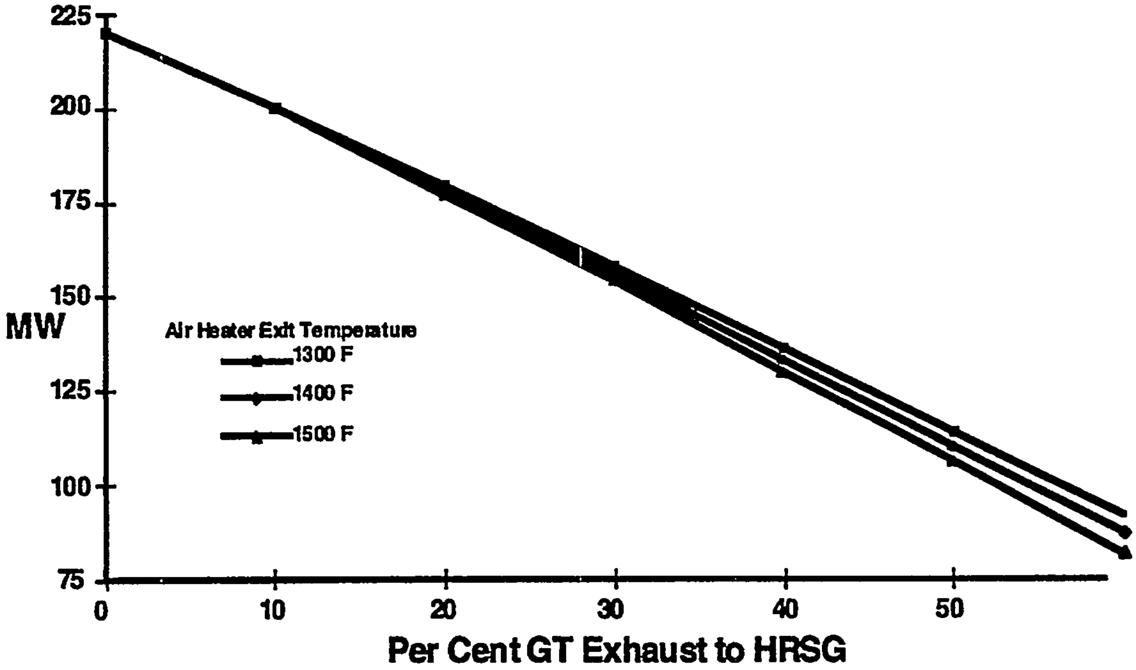

Figure 6. Effect of Air Heater Outlet Temperature.

The foregoing material was repeated in order to illustrate the configurations projected for repowering may exacerbate the operational problems discussed in Section 2. It is obvious that compromises between operational considerations and performance will have to be made. These compromises will be predicated on type of application, e.g., baseload, mid–range, etc. and also on the experience of the specific utility or other user. In a similar manner to economics, the configuration and operation will be very site (application) specific.

Site Specific Repowering

Near the close of Phase I, the UTRC Combustion 2000 Team received from DoE information on a potential repowering site in upper New York state. Additional information on the power plant was supplied by DoE subcontractors, SAIC and Gilbert Commonwealth. This information included a flow sheet for the original power plant (Fig. 7) and a simplified schematic for a repowering scheme based on the use of a GE Frame 6 gas turbine (Fig. 8).

Using the data on the original power plant, the UTRC Team identified a preliminary repowering configuration based on an aero–derivative engine. This engine is the latest version of the P&W FT4000 I/C, featuring significant changes from the earlier version used in the latter part of Phase I. At this time, proprietary concerns limit the scope of engine data available. The major differences, however, are: 1) flow rate, which is increased; 2) pressure ratio, which is increased; and 3) combustion exit temperature, which is decreased. The major effects are the higher flow rate and lower exhaust temperature (due to higher pressure ratio and lower combustion temperature). Subsequent reports will supply greater detail for this machine.

A preliminary flow sheet for the repowering of the NY site is shown in Fig. 9. To be consistent with the DoE–supplied flow sheet a RAH was used in the HITAF; future repowering designs will focus on use of a CAH only. The efficiency of 45%+ is some eight (8) points, or approximately 23% better than the original steam station. The power output has been increased from 122 MW to nearly 200 MW.

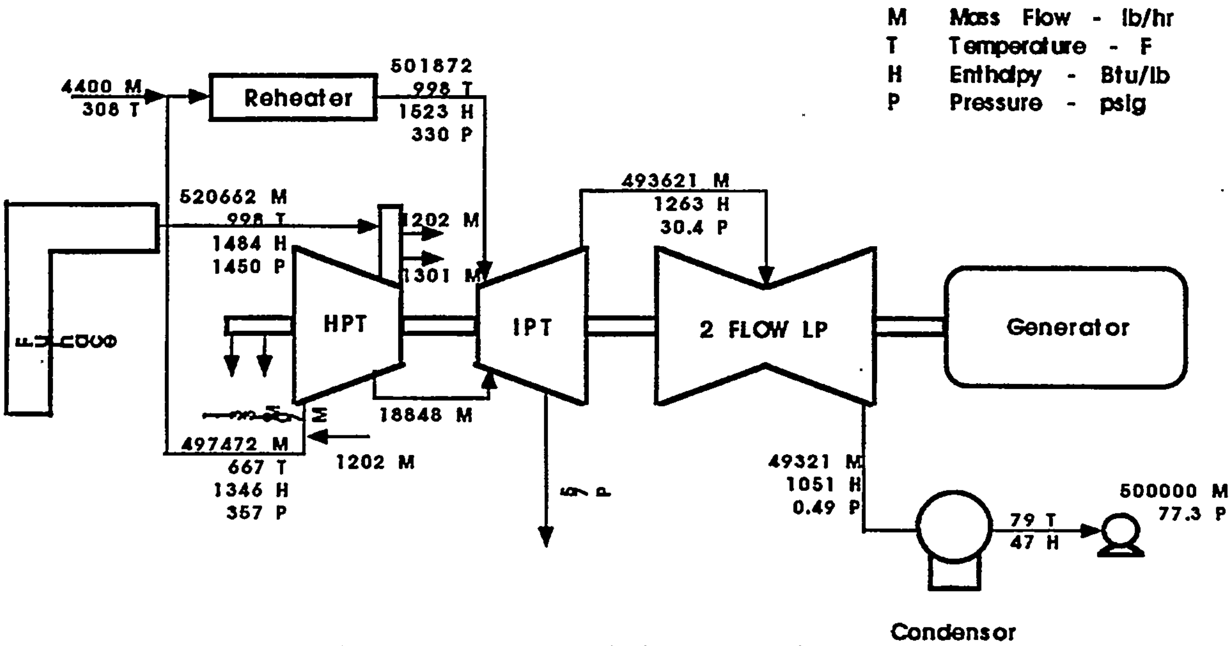

Figure 7. Steam Cycle for Repowering.

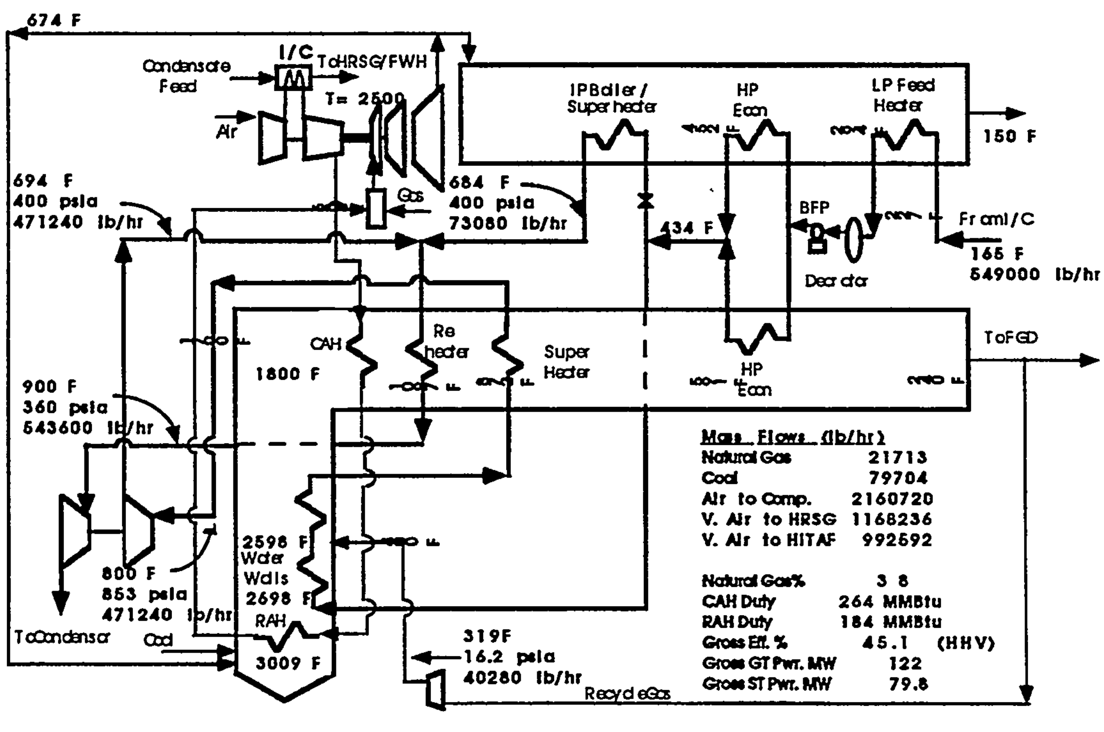

Figure 8. Case 4 with GE 6FA–Type Gas Turbine.

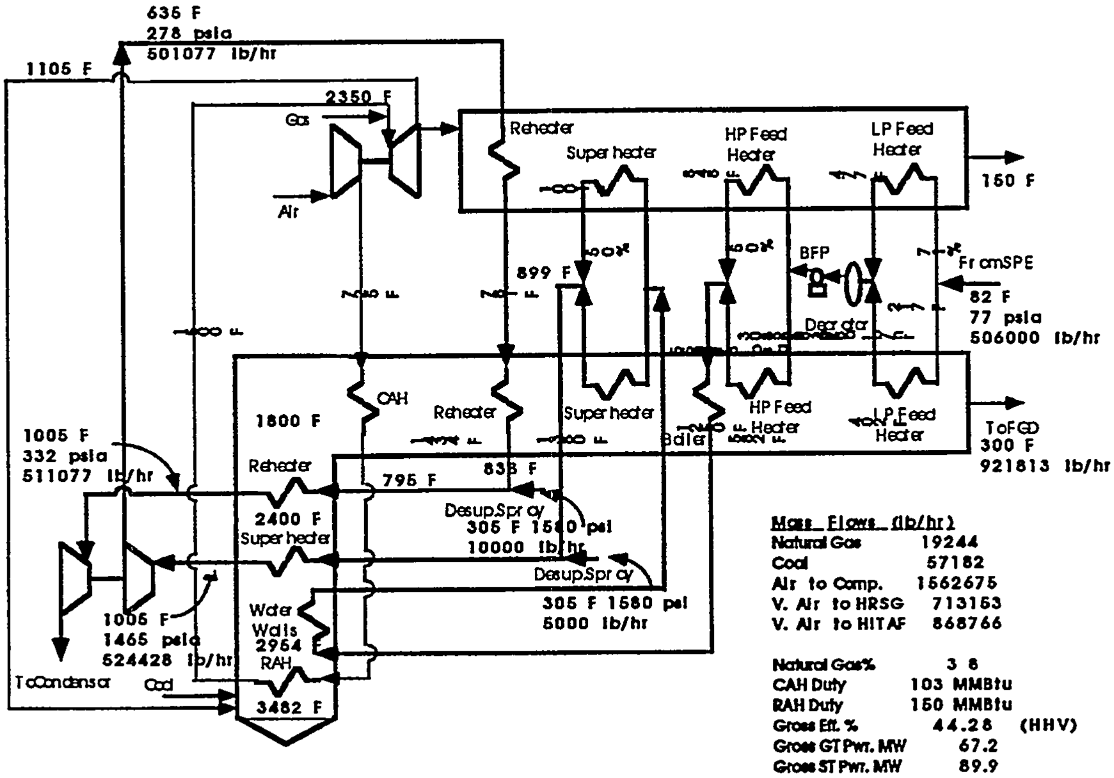

Figure 9. Repowering with Aero-Derivative Gas Turbine.

CONCLUDING REMARKS

The changes in HITAF power plant configuration which could result from implementing the suggestions arising from operational considerations will impact power station performance and cost. This impact must be evaluated and trade-off studies made between added costs for redundancy, additional maintenance and/or downtime and increased fuel costs and/or reduced power output.

EERC LABORATORY AND BENCH-SCALE TESTING

Laboratory Activities

Although the focus of the refractory testing for this program at the Energy & Environmental Research Center (EERC) has shifted to brick materials, some limited testing on monolithic refractories is continuing because of their much lower cost. Strength testing at 1430°C was completed on two commercial SiC refractories initially studied for corrosion resistance under Phase I of the program. The first commercial refractory is a gunning mix (GC950) from Norton Company, with a composition of 82% SiC, 12.2% Al_2O_3, 2% SiO_2, 2.5% CaO, and 1.3% other constituents. The second is a castable refractory (11LI) from Carborundum, with a composition of 83% SiC, 12.5% Al_2O_3, 1.3% SiO_2, 2.8% CaO, 0.2% Fe_2O_3, 0.1% Na_2O, and 0.1% MgO.

The refractories were mixed in a Hobart Model N-50, 5-quart capacity mixer. Water was added to the dry material in the amounts of 7 wt% for GC950 and 9 wt% for 11LI. The mixtures were stirred until desired consistencies were achieved. The mixtures were then placed in molds and vibrated until the mixtures flowed to fill the molds. The GC950 refractory required some packing to fill in the corners of the molds. Then the refractories were dried in three stages. First, the molds were covered with plastic wrap and allowed to cure at room temperature (25°C) for 18-24 hours. Then the samples were removed from the molds and allowed to air dry at room temperature (25°C) for 24 hours. In the final drying stage, the refractory blocks were prefired in a Keith Model No. KKSK 999 3100 high-temperature furnace. Each refractory required a unique programmed ramping sequence. For both refractories, the furnace was slowly ramped to 1430°C, then slowly cooled to room temperature, at slightly different rates.

An Instron Model 8562 testing system was used to perform high-temperature compressive strength testing on the refractory blocks at 1430°C. The samples were 1 in. x 1 in. x 2 in. The furnace was programmed to ramp the temperature up to 1430°C at a rate of 5°C/minute. A total of five blocks of each refractory type were tested. Figure 9 shows the compressive strengths of the GC950 and 11LI refractories at 1430°C, as compared to the strengths measured for several experimental SiC-based monolithic refractories tested in Phase I of the project. The data for the experimental refractories were reported in the January through March 1995 quarterly technical progress report for Phase I of this project.

Figure 10 shows that the commercially available refractories are much weaker than the experimental refractories tested under Phase I. When the refractories were subjected to compressive stresses at 1430°C, they deformed plastically, never actually crushing. The lower strengths are most likely caused by the more glassy nature of the binder in the commercially available materials, as well their higher porosities. After testing, visual inspection of the blocks showed increased porosity due to gas evolution during furnace drying.

In addition to the high-temperature strength testing, static slag corrosion tests were repeated on these refractories, because new slags and procedures are now being used. The tests were done with

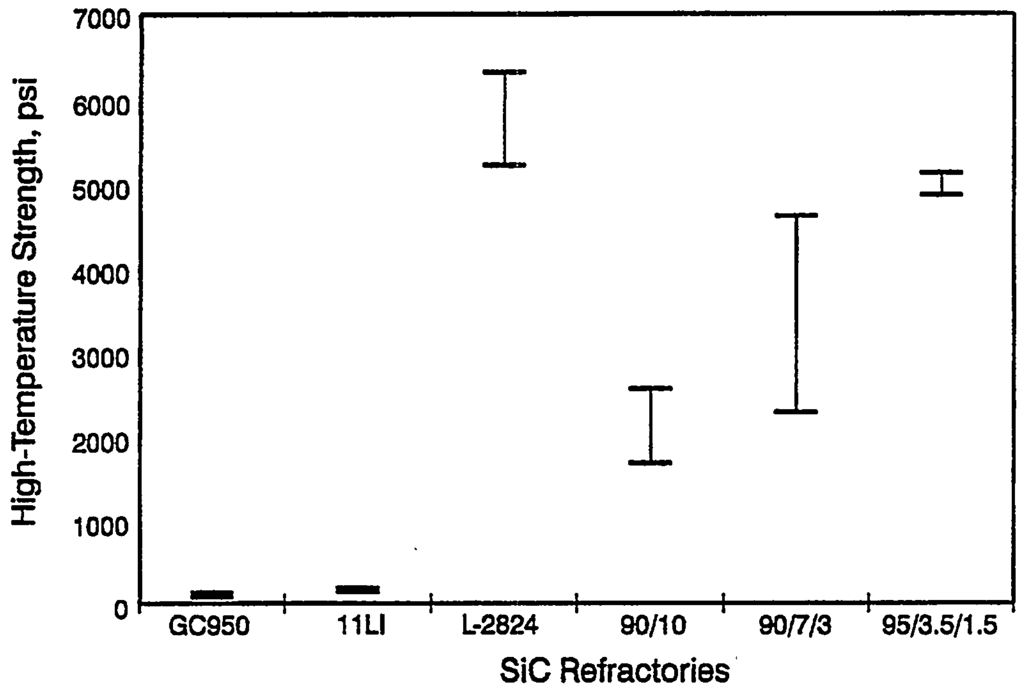

Figure 10. Compressive Strengths of GC950 and 11LI Refractories Compared to Experimental Refractories at 1430°C.

an Illinois No. 6 slag from the Illinois Power Baldwin Plant and a Powder River Basin coal slag formed while Rochelle mine coal was fired at the Northern States Power Company Riverside Plant. Samples of each refractory were exposed to each slag for 100 hours at 1430°C and then quenched. After the exposure, the blocks were cut in half vertically, and the refractory surface recession was measured at three places on each half of the two test blocks, for a total of twelve measurements per refractory per slag.

Figure 11 shows that the surface recession for the GC950 and 11LI exposed to the Illinois No. 6 slag are similar to the experimental refractories. A smooth, 4–5–mm–thick layer of slag remained in the slag well. The outer surfaces of the corrosion blocks had a dense glass coating, assumed to be the binder. Corrosion products of iron silicide are visible along the slag–refractory interface. Several vertical to nearly diagonal cracks extend outward from the corners of the slag wells on each of the blocks. These may have occurred during quenching (no slag has infiltrated the cracks), because of a thermal expansion mismatch between the slag and the refractory. The corrosion layer is relatively even with no visible slag penetration.

Figure 12 shows the average slag penetration into the commercially available refractories, as measured from the original surface by scanning electron microscope (SEM). This value is expected to be a truer indication of the resistance to slag corrosion than the recession given in Figure 11, since in a dynamic system with flowing slag, erosion will cause rapid surface recession once the slag has penetrated the refractory. The SEM analyses show that the castable 11LI has a greater amount of slag

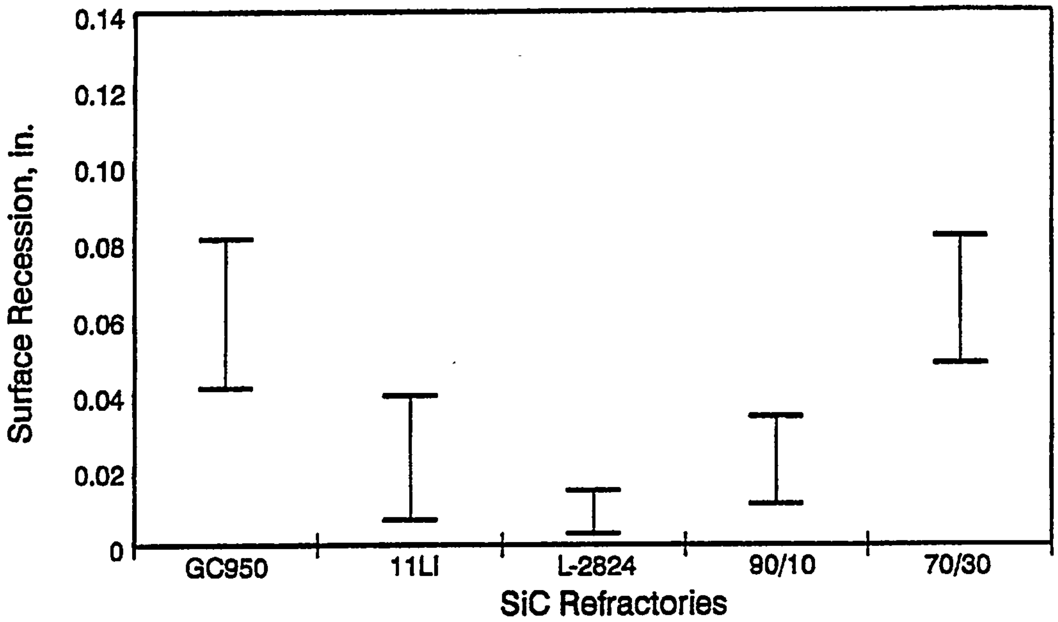

Figure 11. Surface Recession for the GC950 and 11LI Refractories Exposed to the Illinois No. 6 Baldwin Slag.

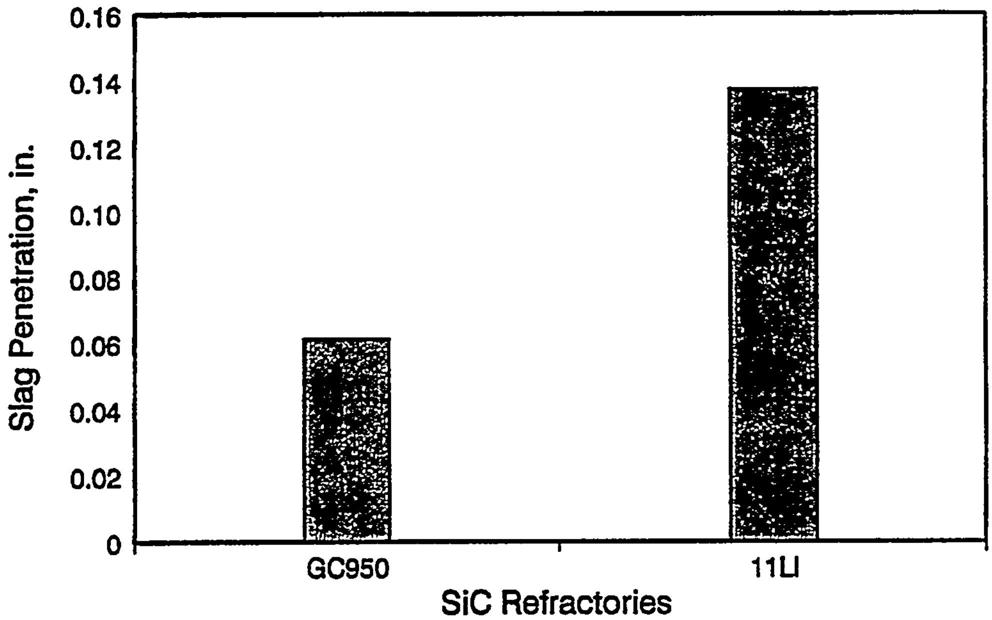

Figure 12. Slag Penetration for the GC950 and 11LI Refractories Exposed to the Illinois No. 6 Baldwin Slag.

penetration than the GC950. These slag penetration measurements should not be compared with Fig. 5 presented in the January through March 1995 quarterly report, as those measurements were done incorrectly. Those measurements will be redone in the next quarter.

Figures 13 and 14 show the surface recession and slag penetration results for the GC950 and 11LI exposed to the NSP Rochelle slag. The recession results are similar to the experimental refractories. A smooth, 1–2–mm–thick layer of slag with few vesicles remained in the slag well. The outer surfaces of the corrosion blocks had the same dense glass coating, and iron silicides are visible along the slag–refractory interface. Vertical to nearly diagonal cracks are also present at the slag–refractory–air interfaces in these blocks. The corrosion layer is more irregular than the corrosion layer from the Illinois No. 6 test, and some slag penetration is relatively deep. Therefore, the values given in the graph, which are averaged numbers, are somewhat misleading and do not indicate the maximum recession rate possible.

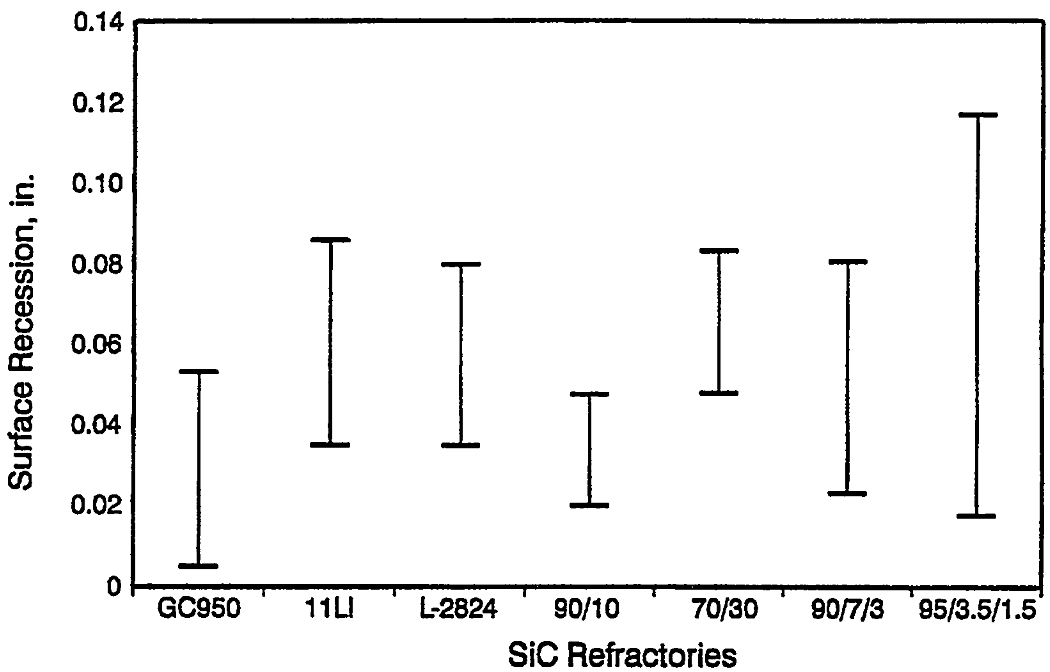

Figure 13. Surface Recession for the GC950 and 11LI Refractories Exposed to the NSP Rochelle Slag.

SEM analysis shows the slag penetration into the castable 11LI to be greater than that of the GC950 castable. As with the Baldwin slag, these slag penetration measurements should not be compared with Fig. 7 in the January through March 1995 quarterly report as the measurements were not done correctly. They will be performed at a later date.

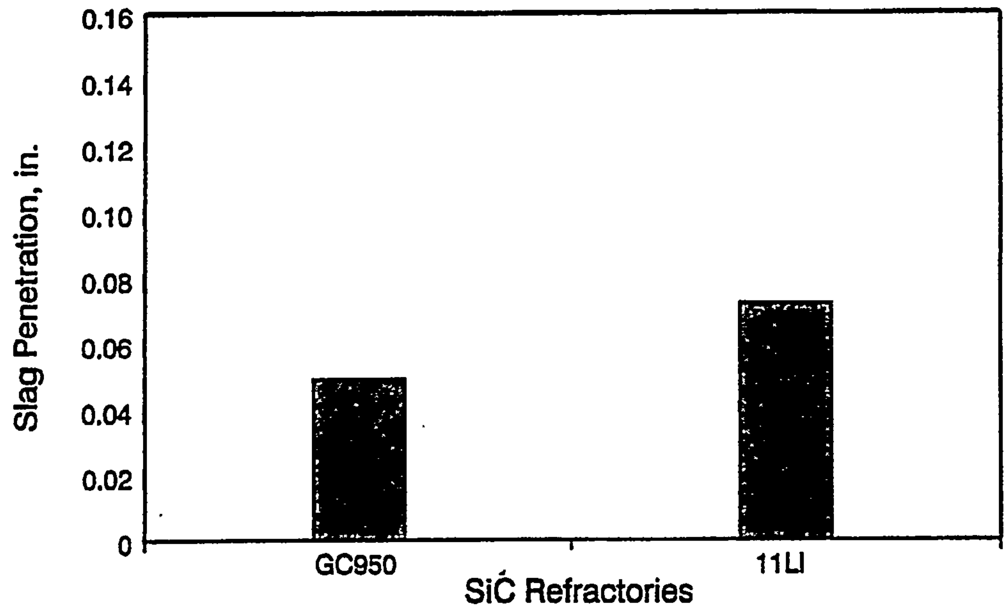

Figure 14. Slag Penetration for the GC950 and 11LI Refractories Exposed to the NSP Rochelle Slag.

Bench–Scale Activities

Background

An initial design of a bench–scale corrosion furnace to test refractories, alloys, and ceramics in a flowing slag environment has been prepared. To determine the lifetime of a refractory in flowing slag, the experiments need to be carried out for long periods under conditions as similar as possible to those in an operating HITAF (high–temperature advanced furnace) in terms of oxygen partial transport to the materials and slag erosion and corrosion. To get realistic slag erosion conditions, it will be necessary to have the slag flow over the materials under the influence of gravity which will best simulate realistic rates of oxygen transport to the material.

Figure 15 shows the recession rates of various ceramics exposed under static laboratory tests and dynamic pilot–scale conditions in a similar temperature range (1230° to 1260°C). Both tests were performed with an Illinois No. 6 slag, except the laboratory tests were performed for 300 hours, whereas the pilot–scale test was only for 100 hours. However, the recession rates for the dynamic, pilot–scale test are several magnitudes greater than those for the static laboratory tests. This emphasizes the need for a bench–scale dynamic corrosion reactor.

We propose to use a continuous flow of slag so that a true rate of corrosion can be estimated. However, it should be noted that the contributions from the complex competing variables involved in this test will make a generalized kinetic model very difficult to accomplish. Several of the variables are

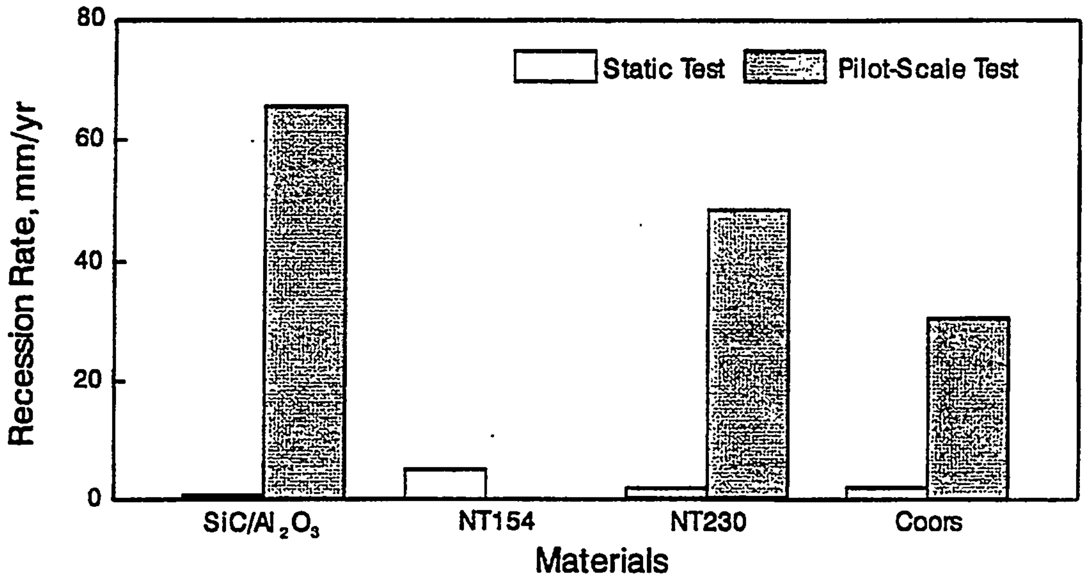

Figure 15. Recession Rates for Ceramics Exposed to Static Laboratory–Scale Corrosion Tests and Dynamic Pilot–Scale Test Using Illinois No. 6 Slag at Furnace Temperatures of 1230° to 1260°C.

related to the refractory's physical properties, such as density, porosity, and inhomogeneity. Secondly, the wettability of the slag on the refractory test piece is critical, since that would control the flow rate or pattern of the slag on the surface of the refractory. Also, the viscosity of the test slag will affect the corrosion kinetics. For example, if the test temperature is kept constant, different slag chemistries would have different flow behaviors which would, in turn, affect the corrosion kinetics. Lower viscosity melts would form thicker boundary layers at the slag–refractory interface, which could result in slow material transfer.

Variations in corrosion and/or erosion kinetics as the slag travels along the refractory test piece are also a concern. When the fresh slag makes contact with the refractory test piece, the corrosion kinetics may be different than farther along the flow path because the composition of the slag may change as it corrodes the refractory. A gradient in the corrosion kinetics would be observed. In order to characterize the changing compositions of the spent slag compared to the fresh slag, it would be necessary to collect this spent slag at the exit port and to freeze the phases on exit at some defined time periods. Freezing the phases can be questionable in the sense that only an artificial frozen phase can be accomplished by this method. The phase composition of the outer shell for a quenched slag bead will be different than that of the core because of basic heat– transfer phenomena.

Experimental

An existing laboratory–scale cyclone furnace would be modified for use in the dynamic slag corrosion tests. The current design will allow four different refractory test pieces to be evaluated under the same experimental conditions at the same time, without any cross contamination of the reactant products. The system is illustrated in Figure 16.

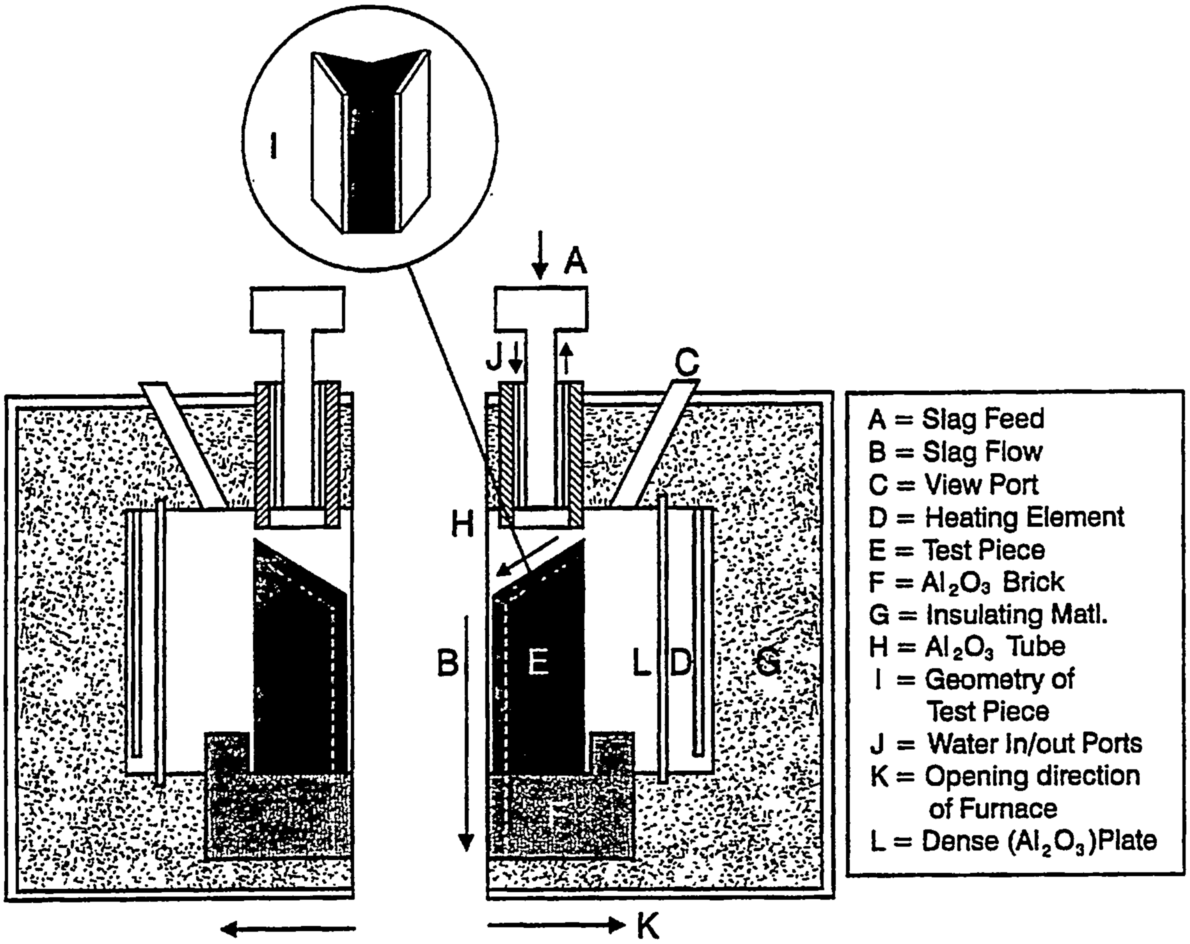

Figure 16. Cross–Sectional View of the Dynamic Slag Corrosion Reactor.

Thermocouples would be located at the point of slag contact with the refractory test piece and the middle and exit points along the refractory. View ports would be located at the slag feed and exit to check for any damming of the flow.

The furnace is electrically heated and has the capability of holding temperatures at 1550°C (2822°F). The furnace will be equipped with four entry ports. These ports will be fitted with alumina tubes, <99.5% Al_2O_3, wrapped with stainless water–cooled feed tubes. This should allow the slag to enter into the furnace as a powder but change to a melt on entry into the furnace. A screw feeder attached with a variable–speed drive will be used to feed the pulverized slag powder. The rate of slag feed will be determined through experimentation to achieve the optimum characteristics for the flow rate and the depth of the flowing slag.

At the end of each test, the slag flow will be stopped for a fixed time to allow for the drainage of any excess slag which might be adhering to the test sample. After cooling, the refractory test sample will be sectioned at various zones to determine the gradient corrosion and erosion kinetics as a function of position.

EERC PILOT–SCALE TESTING

An application for a permit to construct was submitted to the North Dakota State Department of Health, on October 13, 1995, for the pilot–scale slagging furnace to be constructed at the EERC. A response was received in the form of a letter dated November 14, 1995. In the response, the North Dakota State Department of Health stated that a Permit to Construct would not be necessary because of the small size of the proposed slagging furnace, but requested that it be notified upon unit installation. Although not specifically stated, the EERC anticipates that the new slagging furnace system will be added to the EERC permit to operate (Permit No. 078004). In addition, the letter stated that, based on initial dispersion modeling, it will be necessary to limit sulfur dioxide emissions to 16 lb/hr in order to avoid exceeding the ambient standard. Based on a nominal coal firing rate of 2.5 MMBtu/hr (Illinois No. 6, 11,100 Btu/lb and 3.3% sulfur), the 16 lb/hr sulfur dioxide limit should not require the implementation of a sulfur dioxide control system. If the coal firing rate were actually 3 MMBtu/hr, a small reduction (10%) in sulfur dioxide emissions may be required. An option may be to dilute the flue gas with the cooling air stream, effectively reducing the stack sulfur dioxide concentration.

The Uniform Building Code (UBC) requirements, along with the National Fire Protection Association's Life Safety Code (NFPA 101) for Building X, where the pilot–scale slagging combustor is to be built, were reviewed with Grand Forks City representatives in November 1995. An agreement has been reached concerning the use of open deck grating and installation of mid–height sprinkler protection (in addition to the existing ceiling–height sprinkler system), as a substitute for 1–hour fire–resistant construction and providing for a 300–foot maximum travel distance to exits. The midheight and ceiling–height sprinkler system combination will provide necessary column and beam cooling, as well as faster response in the event of a fire at lower elevations. A letter from a Grand Forks City representative, dated December 15, 1995, was received, documenting the review effort and agreement to the general approach.

A bid package was prepared and released for the purchase of an overhead crane in October 1995. Only one bid was received in response to the bid package. The bid was within the acceptable cost range, and procurement efforts are proceeding. Crane installation is expected in late February 1996. Therefore, the crane should be available for use in March 1996 to support erection of structural steel. The remainder of this section will discuss the status of the design and procurement activities in the past quarter.

Structural Steel Design, Procurement, Fabrication, and Erection

Structural steel design was begun this quarter because much of the design, fabrication, and erection of the structure are independent of the slagging furnace design. Load limits and steel orientation are largely dictated by the building. Certain elements of structural steel design and fabrication, such as load–bearing beams for the furnace and other large refractory–lined components, will be completed once the furnace design has been completed. Therefore, most of the structural steel design and fabrication drawings are in progress and will be completed in January 1996.

Initial vendor contacts indicate the cost for structural steel would be nearly $50,000. Much of the steel has been ordered; a small amount has arrived; and some limited fabrication has begun. The bulk of the structural steel is expected to arrive as partial shipments through the end of January 1996. Fabrication activities are expected to begin in earnest the first week of January 1996 and are expected to be mostly completed in the first quarter of 1996. Appendix A contains current structural steel fabrication and erection drawings.

Preliminary Design of the Pilot-Scale Slagging Furnace System

The primary purpose of the pilot-scale slagging furnace is to evaluate heat transfer and material and slag-ash issues related to the performance of the radiant air heater (RAH), slag screen, and convective air heater (CAH) components. An initial list of design questions concerning the pilot-scale slagging furnace was prepared and submitted to project team members in October 1995. Individuals receiving the list via fax included Dan Seery (United Technologies Research Center [UTRC]), Connie Senior (PSI), Mark Palkes (ABB), Phil Smith (Reaction Engineering International [REI]), and John Ruby (Bechtel). As of early November, responses were received from all five organizations. A summary of the design assumptions and conceptual system design based on recent EERC efforts and input from UTRC, PSI, ABB, REI, and Bechtel was to be prepared and submitted for review in late December, following a discussion of the assumptions and conceptual design with UTRC, ABB, and PSI personnel during a videoconference on December 20, 1995. However, because of a winter storm on the East Coast, the videoconference was postponed until January 3, 1996. Following the January 3, 1996, videoconference and general approval of the conceptual design, work will focus on completion of the preliminary system design, including the preparation of P&ID (piping and instrumentation diagrams) drawings, equipment and material lists, and equipment layout drawings (plan and elevation views). A formal review of the completed preliminary system design will be held in conjunction with the, as yet unscheduled, project kickoff meeting to be held in Pittsburgh. The remainder of this section presents the current design assumptions and preliminary system design.

Pilot-Scale Slagging Furnace

The pilot-scale slagging furnace design is intended to be as fuel-flexible as possible, with furnace exit temperatures of at least 2700°F in order to maintain desired slag flow. It will have a nominal firing rate of 2.5 million Btu/hr and a range of 2.0 to 3.0 million Btu/hr using a single burner. The design is based on a bituminous coal (Illinois No. 6, 11,100 Btu/lb) and a nominal furnace residence time of 3.5 seconds. Resulting flue gas flow rates will range from roughly 425 to 640 scfm, with a nominal value of 530 scfm based on 20% excess air. Firing a subbituminous coal or lignite will increase the flue gas volume, decreasing residence time to roughly 3 seconds. However, the high volatility of the low-rank fuels will result in a high combustion efficiency (>99%). The EERC plans to orient the furnace vertically (downfired) and base the burner design on a swirl burner currently used on two EERC pilot-scale pulverized coal (pc)-fired units that are fired at 600,000 Btu/hr. Slagging furnace dimensions will be 48-in. inside diameter (ID) by roughly 16 ft in length. Combustion air preheat capabilities will range from 300° to 900°F. The EERC intends to make use of tube-and-shell heat exchangers to recover heat from the flue gas or cooling air stream to preheat combustion air.

However, it may also be necessary to include an electric air preheater to achieve adequate temperature control. The refractory walls will consist of three castable refractory layers: 2.75 in. of high–density (7 Btu–in./ft^{2}–°F–hr), slag–resistant material, 2.75 in. of an intermediate refractory (4.4 Btu–in./ft^{2}–°F–hr), and 5.5 in. of a low–density insulating refractory (1.8 Btu–in./ft^{2}–°F–hr). Three refractory layers will be necessary to avoid overheating the low–density insulating refractory.

The primary burner will be natural gas– and coal–capable, with coal particle size assumed to be a standard utility grind, 70% –200 mesh. Burner development and testing are not objectives within this activity. However, some burner turndown is desirable and has been factored into the burner design. Flame stability will be assessed by observation of the flame and its relation to the burner quarl as a function of secondary air swirl and operating conditions at full load and under turndown conditions. The basic burner design, an International Flame Research Foundation (IFRF)–type adjustable secondary air swirl generator, is illustrated in Figs. 17 and 18. An IFRF–type adjustable secondary air swirl generator uses primary and secondary air at approximately 15% and 85% of the total air, respectively, to adjust swirl between 0 and a maximum of 1.9.

"Swirl" is defined as the ratio of the radial (tangential) momentum to axial momentum imparted to the secondary air by movable blocks internal to the burner and is used to set up an internal recirculation zone (IRZ) within the flame that allows greater mixing of combustion air and coal. Swirl is imparted by moving blocks to set up alternate paths of radial flow and tangential flow, creating a spin on the secondary air stream that increases the turbulence in the near–burner zone. At the fully open position of the swirl block, the secondary air passes through the swirl burner unaffected, and the momentum of this stream has only an axial component (the air enters the combustion chamber as a jet). As the angle of the blocks changes, the air begins to spin or swirl, and the radial component to the momentum is established, creating the IRZ in the near–burner region. It is the ratio of this radial component of the momentum to the axial component that establishes the quantity defined as swirl.

The adjustable swirl burners, currently used by the EERC, consist of two annular plates and two series of interlocking wedge–shaped blocks, each attached to one of the plates. The two sets of blocks can form alternate radial and tangential flow channels, such that the air flow splits into an equal number of radial and tangential streams, which combine farther downstream into one swirling flow, as shown in Fig. 18. By a simple rotation of the movable plate, radial channels are progressively closed and tangential channels opened so that the resulting flux of angular momentum increases continuously, between zero and a maximum value. This maximum swirl depends on the total air flow rate and the geometry of the swirl generator. Swirl can be calculated from the dimensions of the movable blocks (the ratio of the tangential and radial openings of the blocks) or from the measurement of the velocity of the air stream (obtaining both radial and axial components). The following description of that calculation is provided by Beér and Chigier (Ref. 1):

When rotating motion is imparted to a fluid upstream of an orifice, the fluid emerging from the orifice has a tangential velocity component, in addition to the axial and radial components encountered in nonswirling jets. The presence of the swirl results in radial and axial pressure gradients

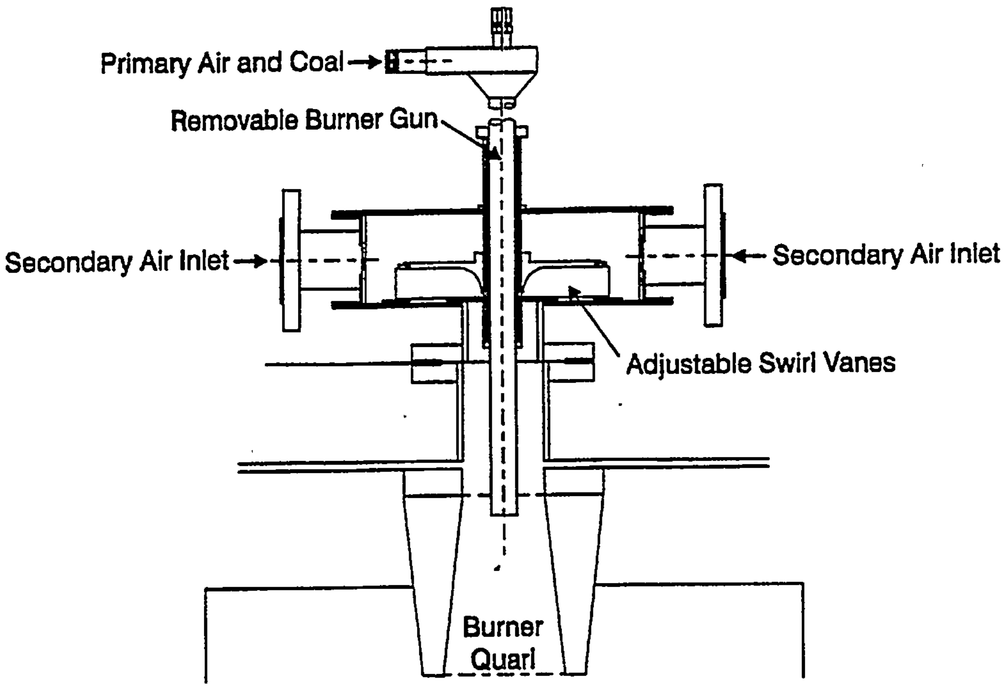

Figure 17. IFRF Adjustable Swirl Burner.

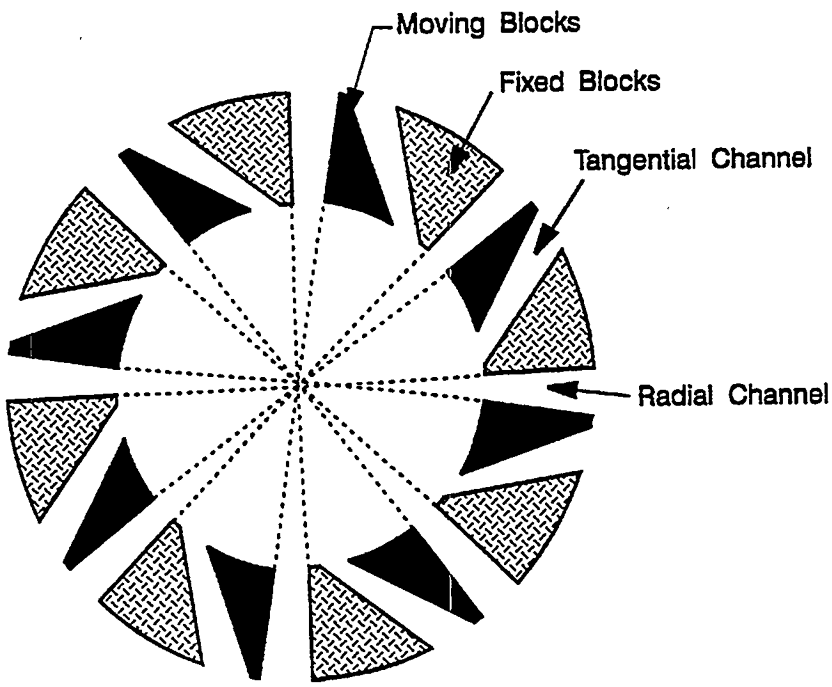

Figure 18. Cross Section of Movable Block Assembly.

which, in turn, influence the flow field. In the case of strong swirl, the adverse axial pressure gradient causes reverse flow along the axis, setting up the internal recirculation zone.

Secondary air swirl is used to stabilize the flame. In the absence of swirl, loss of flame may result, increasing the risk of dust explosion. As swirl is applied to the combustion air, coal particles are entrained in the IRZ, increasing the heating rate of the particles, leading to increased release of volatiles and char combustion. The flame becomes more compact and intense as swirl is increased to an optimum level, which is characterized in existing EERC pilot–scale test facilities as the point at which the flame makes contact with the burner quarl. Increasing swirl beyond this level can pull the flame into the burner region, unnecessarily exposing metal burner components to the intense heat of the flame and possible combustion in the coal pipe.

Increasing swirl to provide flame stability and increased carbon conversion can also affect the formation of NO_X. The high flame temperatures and increased coal–air mixing associated with increased swirl create an ideal situation under which NO_X may form. In full–scale burners with adjustable vanes, swirl is often increased to reach the optimum condition and then decreased slightly to reduce the production of NO_X. Although NO_X emissions are of interest, their control is not a key objective for the pilot–scale slagging furnace. Therefore, burner operational settings will be based on achieving desired furnace exit temperatures and slag conditions in the furnace.

Flame stability under turndown conditions will be characterized by firing the test fuel at reduced load (typically 66% to 85% of the full load rate), maintaining the same primary air flow and adjusting the secondary air flow to meet excess air requirements. At this time, the EERC intends to simply scale up the existing burner design based on increased combustion air volumetric flow rates. If desired, the final burner design may be reviewed with the project team organizations interested in those details.

Temperature measurement in the furnace will be extremely important. Furnace exit temperature will be measured using a minimum of two methods. Methods to characterize the furnace during shakedown will include Type S thermocouples, an optical method, and a high–velocity thermocouple (HVT). A minimum of four additional thermocouple locations will be designated for the furnace interior in the final design. Temperature measurements (two each) at the interface between the hard and intermediate refractory layers are planned, as well as between the intermediate and insulating refractory layers. Observation ports will be located in the furnace to permit visual observation of the primary burner flame, auxiliary burner flame, RAH panels, slag screen, and slag tap. With the exception of the furnace exit (inlet to the slag screen), there are no plans at this time to include additional sampling ports in the furnace walls. However, in order to adequately characterize the furnace during shakedown and since RAH test panels will not necessarily be available for all furnace operating periods, the first set of doors built for the RAH panel locations will have ports to permit the insertion of temperature and heat flux measurement probes.

An auxiliary gas burner (500,000 Btu/hr) will be located in the area of the furnace exit in order to ensure desired slag flow from the furnace and the slag screen. This auxiliary burner will compensate for heat losses through the furnace walls, site ports, and RAH test panels. The use of the auxiliary gas burner will be beneficial during start–up to reduce heatup time and to prevent the freezing of slag on the slag screen when initially switching to coal firing. The auxiliary gas burner will be fired at stoichiometric conditions to avoid high excess air levels in the system. Also, the EERC anticipates that the auxiliary gas burner will be fired at relatively low rates (<200,000 Btu/hr) once the furnace reaches thermal equilibrium. The overall pilot–scale slagging furnace system is illustrated in Figs. 19 and 20. Table 1 summarizes the furnace design parameters.

Radiant Air Heater Panels

A key furnace design feature will be accessibility for installation and testing of one large RAH panel and four small panels. Plans, at this time, call for the furnace design to accept a large RAH panel with a maximum active size of 1 ft × 6 ft. This size was selected to minimize furnace heat losses and based on manufacturing constraints identified by UTRC. Flame impingement on the RAH panels is not necessarily a problem. Cooling air for the large RAH panel will be provided by an existing EERC air compressor system, having a maximum delivery rate of 510 scfm and a maximum stable delivery pressure of 275 psig. Backup cooling air is available from a smaller compressor at a maximum delivery rate of 300 scfm and pressure of <100 psig. It will be necessary to heat the cooling air to achieve the 1300°F radiant panel cooling air inlet temperature desired. Outlet cooling air temperatures from the large RAH panel will range from 1375° to 1800°F, by adjusting cooling air flow rate. Cooling air pressure for the large radiant panel will be roughly 150 psig. Based on some limited heat–transfer calculations, the cooling air flow rate will be <200 scfm. The EERC anticipates using flue gas heat exchange and electrical heating to meet heated cooling air temperature requirements for the large RAH. The advantage of a combination of heat sources for the cooling air stream at this scale is greater flexibility and range of control. Figures 21 and 22 illustrate the RAH panels, including the approach to insulation and proposed location of thermocouples. Proper insulation and thermocouple location will be required to minimize and document edge effects, respectively, as well as adequately define surface temperature distributions, total heat absorption, and local heat flux.

Furnace design will also permit the installation of four 1–ft x 1–ft RAH panels. The actual size of the test brick for each of the small panels is 10 in. x 10 in., with an increased size to 12 in. x 12 in. on the front face of the alloy test panel. Cooling air for the small RAH panels will be provided by an existing EERC air compressor system, having a maximum delivery rate of 510 scfm and a maximum stable delivery pressure of 275 psig. Backup cooling air is available from a smaller compressor at a maximum delivery rate of 300 scfm and pressure of <100 psig. Cooling air temperature at the inlet of the small RAH panels will be low, facility–ambient temperature. At this time, the operating pressure of the cooling air system for the small RAH panels is assumed to be 150 psig, with a flow rate of <100 scfm. The outlet cooling air temperature for the small RAH panels has not been specified; however, it is assumed to be less than 1800°F. Again, actual outlet cooling air temperatures will be controlled as a function of cooling air flow rate. Figure 21 illustrates the small RAH panels, including the proposed location of thermocouples. Proper insulation and thermocouple location will be required to minimize and document edge effects, respectively.

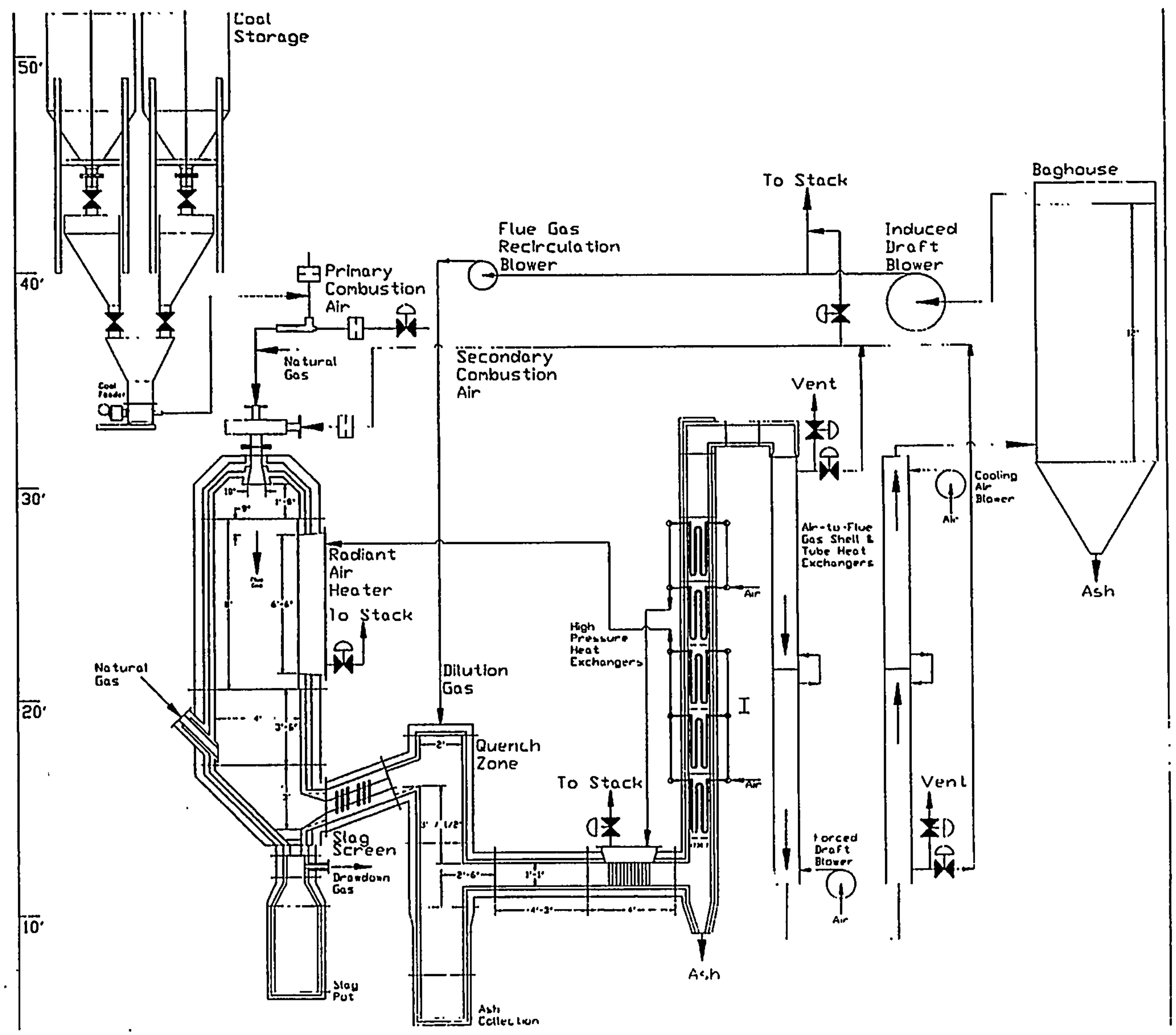

Figure 19. Combustion 2000 Furnace and Support Systems.

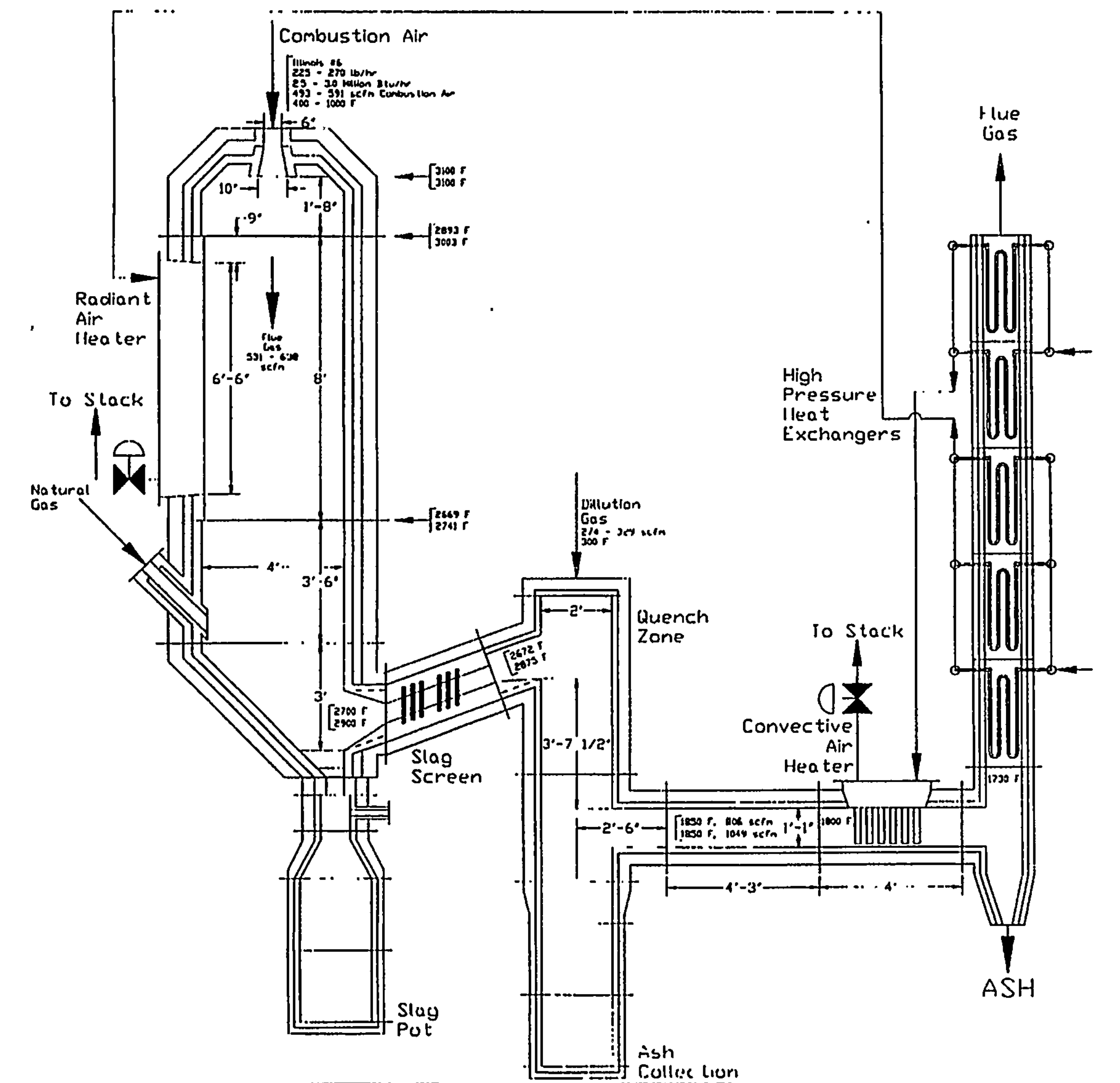

Figure 20. Combustion 2000 Furnace and Refractory Components.

Table 1. Flow and Heat–Transfer Calculations for Combustion 2000 Slagging Furnace and Refractory Ducts

Illinois No. 6 Bituminous					
Furnace			Furnace		
Furnace ID, in.	48	48	Furnace ID, in.	48	48
Firing Rate, MMBtu/hr	2.500	3.000	Firing Rate, MMBtu/hr	2.500	3.000
Coal Feed Rate, lb/hr	225	270			
Air Flow Rate, scfm	493	592	Furnace Exit Duct Out		
Flue Gas Flow Rate, scfm	531	637	Gas Temp., °F	2650	2850
Flue Gas Flow Rate, acfm	3433	4242	Flue Gas Flow Rate, acfm	3250	4269
Furnace Gas Velocity, ft/s	4.6	5.5	Gas Velocity, ft/s	54	71
Exit Gas Velocity, ft/s	57	71			
Flue Gas Residence Time, s	3.5	2.9	Dilution Gas Requirements		
			Gas Velocity in, ft/s	13.5	17.8
Auxiliary Burner, MMBtu/hr	0.070	0.190	Exit Gas Temp., °F	1850	1850
Wall Losses, MMBtu/hr	0.170	0.173	Dilution Gas Temp., °F	300	300
Other Losses, MMBtu/hr	0.200	0.200	Calc. Dilution Gas, scfm	274	411
			Total Flue Gas Flow, scfm	806	1049
Furnace Sect. Length, ft	16	16	Flue Gas Flow Rate out, acfm	3579	4660
Refract. 1 Thickness, in.	2.75	2.75	Gas Velocity out, ft/s	14.9	19.4
Refract. 2 Thickness, in.	2.75	2.75			
Refract. 3 Thickness, in.	5.25	5.25	Dilution Gas Nozzles		
Furnace Weight, tons	12.6	12.6	Nozzle Diameter, in.	1.9	1.9
			No. of Nozzles	3	3
Inlet Gas Temp., °F	3100	3100	Dilution Gas Flow, acfm	401	601
Avg. Gas Temp., °F	2900	3000	Dilution Gas Velocity, ft/s	113	170
Exit Gas Temp., °F	2700	2900			
Refract. 1 Surf. Temp., °F	2869	2888	Convective Air Heater		
Refract. 2 Surf. Temp., °F	2554	2570	Gas Temp., °F	1800	1800
Refract. 3 Surf. Temp., °F	2102	2115	Flue Gas Flow Rate, acfm	3501	4559
Furnace Skin Temp., °F	252	254	Gas Velocity, ft/s	50	65

Assumptions:

Excess air is 20%.

First layer of Narco Cast 60 refractory has thermal conductivity of 7 Btu–in./ft^2–°F–hr.
Second layer of Harbison–Walker Lightweight Castable 30 refractory has thermal conductivity of 4.4 Btu–in./ft^2–°F–hr.
Third layer of Harbison–Walker Lightweight Castable 26 refractory has thermal conductivity of 1.8 Btu–in/ft^2–°F–hr.

Furnace Slag Tap

The slag tap design will make use of EERC experience and, possibly, some existing components used in a slagging gasifier. The design is intended to be as simple and functional as possible. To that end, the current slag tap design is a simple refractory–lined hole in the bottom of the furnace. The diameter of the slag tap will be 6 in. initially, with the potential to reduce the diameter to 2 in. by simply repouring refractory. To minimize heat losses associated with the slag tap, slag will be collected in an

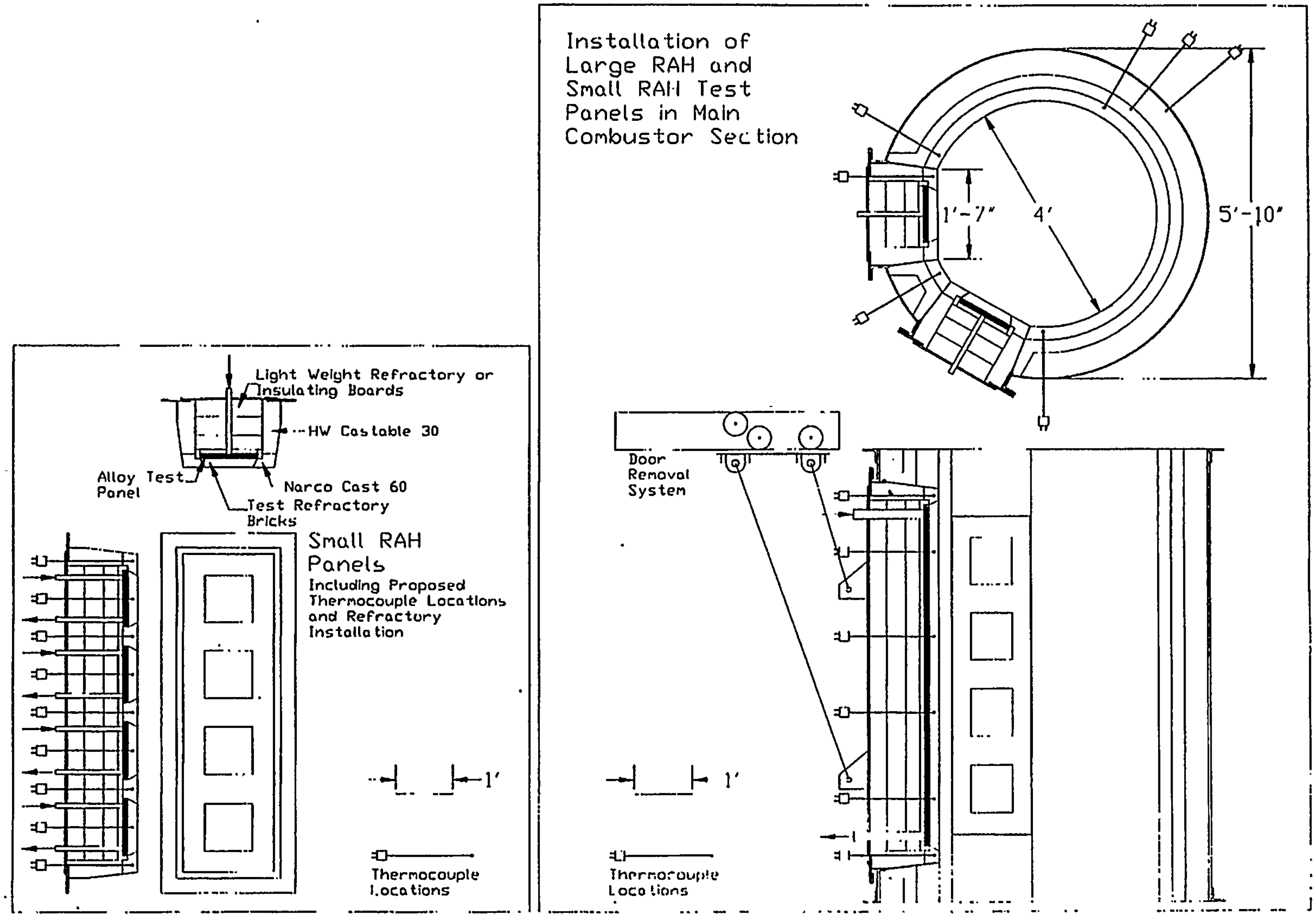

Figure 21. Radiant Air Heater (RAH) Test Panels and Installation.

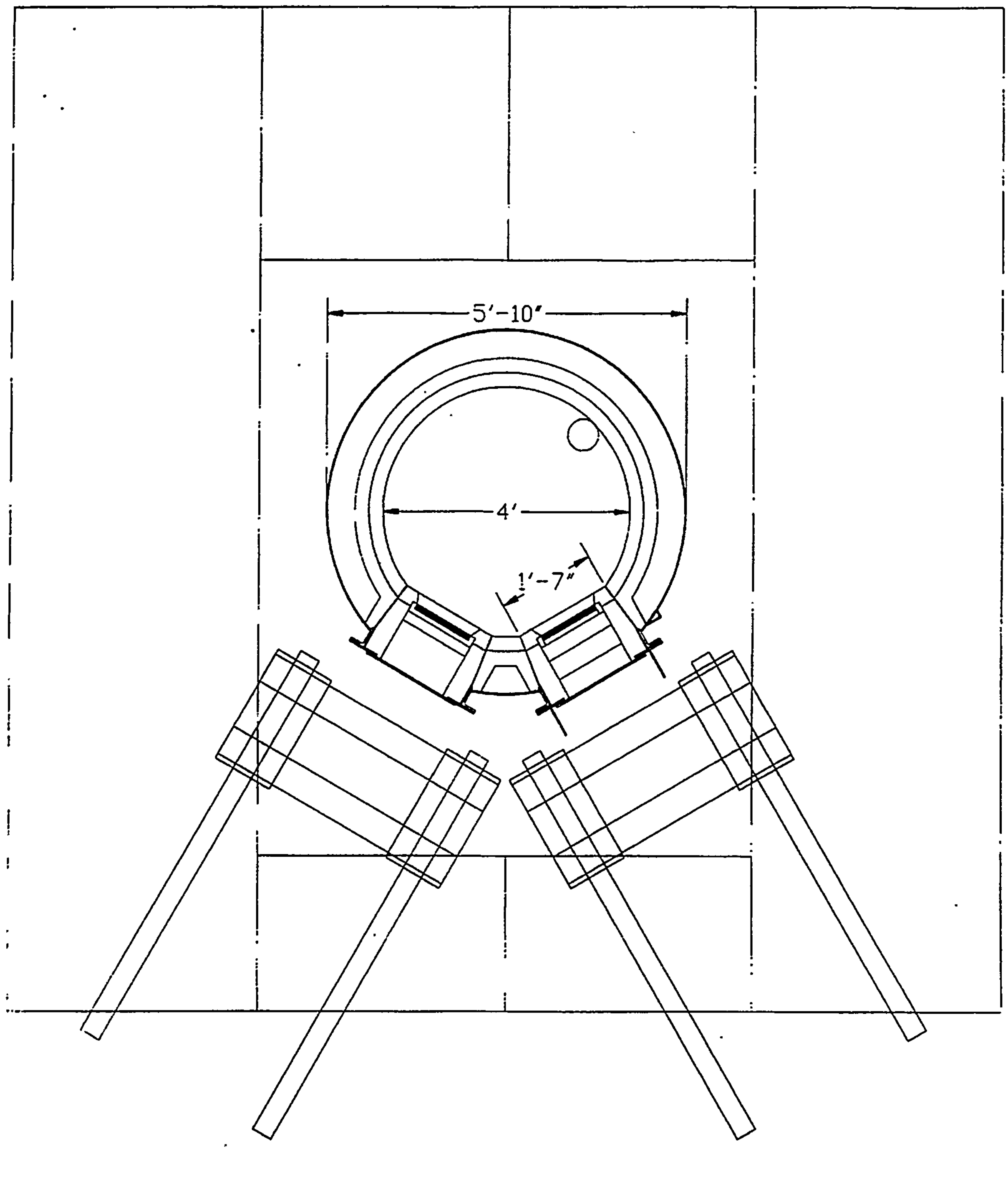

Figure 22. Door Removal System and Centerline Locations of Support Beams.

uncooled dry container with refractory walls. A design feature being considered would involve drawing a small amount of flue gas (<10%) through the slag tap to maintain temperature and promote slag flow. The flue gas would reenter the system in the dilution/quench zone or downstream of the convective section. Alternative design options have also been considered. The most complicated option would involve the use of water-cooled hearth plates, with a slag tap burner originally designed for use on a ton/hr slagging gasifier.

The refractory in the slag pot will be sacrificial when slag samples are collected for analysis and maintenance performed after a test. The actual size and number of slag collection vessels to be constructed have not been determined and will depend on the slag quantity estimated from a 1-week operating period and procedural options for changing slag containers during furnace operation. The use of a single slag pot for a week of operation is desirable.

Slag Screen

The slag screen design will be provided by PSI. Recent discussions indicate a desired flue gas approach velocity of 60 ft/s and an inlet flue gas temperature of 2700°F. The flue gas outlet temperature from the slag screen must be >2450°F to minimize the potential for slag freezing in the slag screen. The slag screen will consist of six rows of four 1.5-in.-diameter vertical tubes mounted in a sloped duct to facilitate slag flow from the slag screen into the furnace slag tap. The centerline-to-centerline tube spacing in each row is 4.5 in. Centerline-to-centerline spacing between individual rows in each set is 4.5 in., with a 6-in. centerline spacing between each set of three rows. Duct dimensions for the slag screen are proposed to be 12 in. x 12 in. x 4 ft, with the height of the duct in the area of the tubes reduced to roughly 6 in. The resulting flue gas velocity through the slag screen will be roughly 120 ft/s. Mullite tubes have been proposed for use in the slag screen without cooling to minimize heat loss and avoid slag freezing in the last few rows of tubes. Routine on-line cleaning of the slag screen is not anticipated to be a requirement. However, the availability of one or more ports permitting the use of an air or steam lance for periodic cleaning is desirable. The availability of ports for collecting gas and solid samples at the inlet and outlet of the slag screen is also desirable, along with pressure taps for measuring slag screen differential pressure.

Furnace exit design will involve the use of a refractory-lined spool piece that can be interchanged with the slag screen. This will permit furnace operation with and without the slag screen.

Dilution/Quench Zone

The dilution/quench zone will be located immediately downstream of the slag screen. Flue gas recirculation will be used to cool the flue gas in the dilution/quench zone and freeze remaining slag particles. A centrifugal flue gas recirculation fan will remove flue gas from the system immediately downstream of the induced-draft (ID) fan. Dilution gas will be injected into the quench zone through multiple nozzles. The temperature of the dilution gas is assumed to be 300°F. Therefore, assuming a furnace firing rate of 2 to 3 MMBtu/hr, and a flue gas exit temperature of 1850°F from the quench zone, the dilution gas flow rate would range from 220 to 330 scfm. A flow control valve will be used to control the flue gas recirculation rate, based on the flue gas exit temperature from the dilution/quench zone. The piping that transports the dilution gas will be insulated to avoid condensation problems.

The evaluation of selective noncatalytic reduction (SNCR) is not a priority at this scale. Therefore, no consideration will be given to SNCR evaluation when finalizing the design of the dilution/quench zone. However, ports will be located in the dilution/quench zone to permit the addition of NH_3 or other additives, if desired. Flue gas sampling ports will be located in and downstream of the dilution/quench zone. These options were requested by PSI to permit evaluation of new measurement techniques for NO_x, NH_3, and other species in the dilution/quench zone.

Convective Air Heater

The CAH design will be provided by UTRC. The flue gas flow rate to the CAH section will be 645 to 970 scfm (2800 to 4800 acfm at 1800°F). A square duct dimension of 1.17 ft^2 should result in a flue gas approach velocity of 50 to 60 ft/s to the CAH. Based on recent discussions, the CAH will consist of three or four rows of 2-in.-diameter tubes. Tube spacing will be a minimum of 4 in. on center, with an overall CAH section dimension of 13 in. x 13 in. x 20 in. Cooling air for the convective air heater will be provided by an existing EERC air compressor system, having a maximum delivery rate of 510 scfm and a maximum stable delivery pressure of 275 psig. Backup cooling air is available from a smaller compressor at a maximum delivery rate of 300 scfm and pressure of <100 psig. Cooling air heating and flow rate control will be necessary to achieve the 700°F minimum (1000°F maximum) inlet cooling air temperature desired, effectively controlling surface temperatures, and provide for an inside heat-transfer coefficient similar to the commercial design. At this time, the operating pressure of the cooling air system is assumed to be 150 psig. The EERC anticipates using flue gas heat exchange and electrical heating to meet heated cooling air temperature requirements for the CAH. The advantage of a combination of heat sources for the cooling air stream at this scale is greater flexibility and range of control. The cooling air exit temperature from the CAH should not exceed 1300°F.

Observation ports are planned for the CAH section. The number and location of the observation ports will depend on the CAH design developed by UTRC. In addition, ports for inserting air or steam lances for intermittent manual cleaning of the CAH will be included. Again, the number and location of these ports will depend on the CAH design developed by UTRC. Critical measurements relative to the CAH will include accurate surface temperatures, cooling air temperatures, cooling air flow rates, and pressure. CAH testing to evaluate heat-transfer performance is a high priority, requiring extensive instrumentation to monitor performance adequately. An important secondary priority is length of material life in relation to operating conditions and ash deposition.

Emissions Control

A pulse-jet baghouse will be used for final particulate control on the pilot-scale system. The pulse-jet baghouse will be constructed of stainless steel and electrically heated and insulated to provide adequate temperature control to avoid condensation problems on start-up and shutdown. The intent is that the baghouse can be operated at both cold-side (250° to 400°F) and hot-side (600 to 750°F) temperatures. Because of the planned operating conditions, materials of construction will be primarily stainless steel (14-gauge sheet metal and 2 x 2-, 3 x 3-, and 4 x 4-in. angle iron). A figure illustrating the pilot-scale pulse-jet baghouse is not yet available.

Flue gas flow rates to the baghouse are expected to range from a low of 665 scfm (1040 acfm at 350°F) to a maximum of 1060 scfm (2365 acfm at 700°F). Therefore, the baghouse design will be based on an average flue gas flow rate of 850 scfm (1325 acfm at 350°F or 1900 acfm at 700°F). The baghouse will be sized to accommodate a maximum of 36 bags mounted on wire cages. Bag dimensions will be 6 in. in diameter by 10 ft in length, providing a total filtration area of 565 ft^2. Arranging the bags in six rows of six bags each allows the number of bags on–line to be changed by installing different tube sheets. For example, when the baghouse is operated at 350°F and 850 scfm (1325 acfm), only 24 bags will be required to achieve a filter face velocity of 3.5 ft/min. If all 36 bags are installed, the filter face velocity would be roughly 2.3 ft/min. In the event that the baghouse is operated at a hot–side condition (1900 acfm at 700°F), 30 bags would result in a filter face velocity of 4 ft/min, and 36 bags would decrease the face velocity to 3.4 ft/min. At a maximum potential flow rate of 2365 acfm (1060 scfm at 700°F), 36 bags would result in a filter face velocity of 4.2 ft/min.

Initially only one tube sheet will be constructed, permitting the installation of 36 bags (565 ft^2 of filtration area) arranged in a six–by–six array. Installing the maximum number of bags will permit the overall system to be evaluated over the broadest potential range of operating conditions during shakedown, while minimizing the potential impact of the baghouse on overall system performance. Each filter bag will be secured to the tube sheet using a snap band sewn into the top cuff. Stainless steel wire cages with 20 vertical wires and 6–in. ring spacing will provide bag support. The pulse–jet baghouse will be a single compartment capable of either on– or off–line cleaning. Flue gas will enter the baghouse in an area just below the bottom of the cage–supported bags and above the ash hopper. Access to the filter bags and stainless steel wire cages will be gained by removing the clean air plenum at the top of the baghouse. The baghouse will be supported at an elevation on one of the upper decks to facilitate inspection, maintenance, installation, and removal of filter bags and cages.

Low–watt density heat tape made for conductive surfaces will be used to preheat the baghouse to prevent moisture condensation during start–up and to control baghouse temperature over the range of interest (350°B700°F); 1–in. heat tape will be run vertically on each of the four baghouse walls spaced on 6–in. centers. The individual elements will be wired in parallel to minimize the impact of single element failures. The baghouse ash hopper will be heated in a similar manner. Five temperature controllers will be used to control electrical resistance heating on the pulse–jet baghouse.

Pulse–jet cleaning will be triggered as a function of baghouse differential pressure or as a function of time. The baghouse pulse–jet cleaning system will be operated by a controller that permits adjustment of cleaning frequency and pulse duration. Timers will be used to set pulse duration and off time, while counters total baghouse operating time and test time in hours and total and test cleaning cycles. Filter bag cleaning will occur when the controller opens the solenoid–operated valves between the pulse–air reservoir and the six pulse–air manifold lines. Each manifold line will provide pulse air to six filter bags. Six filter bags will be cleaned simultaneously, with a short delay between each set of filter bags to allow air pressure to recover in the pulse–air reservoir.

Air for bag cleaning will be provided by an existing EERC air compressor system having a maximum delivery rate of 510 scfm and a maximum stable delivery pressure of 275 psig. Backup air is available from a smaller compressor at a maximum delivery rate of 300 scfm and pressure of <100 psig. High-pressure/low-volume and low-pressure/high-volume cleaning options will be included in the design of the pulse-air system. In order to operate at a low-pressure/high-volume condition in the pulse-jet baghouse, a pulse volume of 0.03 ft^3/ft^2 of fabric surface or roughly 0.5 ft^3/bag will be necessary at a pulse-air reservoir pressure of <40 psig. The pulse volume for each set of six bags would be 3 ft^3. For a high-pressure/low volume case, 0.01 to 0.02 ft^3/ft^2 of fabric surface is common and the pulse-air reservoir pressure will be 40 to 100 psig. Therefore, a maximum pulse volume of 1.9 ft^3 will be required to clean one set of six bags. The required pulse duration to achieve a desired pulse-air volume, as a function of pulse pressure, will be determined during system shakedown.

Flue gas sample ports will be installed at the inlet and outlet of the baghouse to permit flue gas flow rate measurements and sampling for gaseous/vapor-phase constituents as well as fly ash. Specific routine measurements to be made will include flue gas oxygen, sulfur dioxide, and nitrogen oxide concentrations, as well as fly ash particle-size distribution and mass loading. Hazardous air pollutants (HAPs) will be measured on a limited basis. Thermocouples will be installed at the inlet and outlet, as well as at four locations along the length of the chamber. Differential pressure across the chamber and static pressure at the outlet of the chamber will be monitored continuously with pressure transducers. Gauges will also be used to visually monitor baghouse differential and static pressure at the main combustor control panel. Baghouse thermocouple and pressure transducer data will be automatically logged on the data acquisition system. As a backup, baghouse data will be recorded manually on data sheets on a periodic basis.

Baghouse ash removal will be accomplished by opening a 6-in. knife valve at the bottom of the hopper, allowing ash material to drain through a stainless steel pipe into a 55-gallon drum on the main floor level. An ash container of this size facilitates handling and is adequate for ash accumulation during 100-hr and longer test periods.

The EERC is not planning to install a sulfur dioxide control system. Initial dispersion modeling data, developed by the North Dakota State Department of Health, indicate that sulfur dioxide emissions would have to be limited to 16 lb/hr to avoid exceeding the ambient standard. Based on a 2.5-MMBtu/hr coal-firing rate, the EERC is not concerned about the potential 16-lb/hr sulfur dioxide limit or the need for a sulfur dioxide control system. If the coal-firing rate were actually 3 MMBtu/hr, a small reduction (10%) in sulfur dioxide emissions might be required. One option may be to dilute the flue gas with the cooling air stream, effectively reducing the stack sulfur dioxide concentration.

Instrumentation and Data Acquisition

The instrumentation and data acquisition components for the pilot-scale slagging furnace system will address combustion air, flue gas, cooling air, cooling water, and other appropriate measurements (temperatures, static and differential pressures, and flow rates). Flue gas will be monitored for oxygen, sulfur dioxide, carbon monoxide, carbon dioxide, and total nitrogen species (nitric oxide and nitrogen dioxide) on a continuous basis at the furnace exit. A set of existing gas analyzers for oxygen, sulfur dioxide, and nitrogen species will be available to monitor a second system location simultaneously. Surface temperature measurements are anticipated for the RAH panels and CAH section. In addition, heat flux measurements for the furnace, RAH panels, and CAH section have been recommended. Orifice plates or other on–line devices (monitored with pressure transducers and gauges on the main combustor control panel) will be installed at various points in the system to measure combustion air flow rate, flue gas flow rate, flue gas recirculation rate, and cooling air flow rates. The following is a preliminary list of planned temperature, pressure, and flow rate measurements for the pilot–scale slagging furnace system:

- **Thermocouple Measurements**
  - Combustion Air System
    - Facility Ambient Air Temperature, Type K, data logger
    - Combustion Air Inlet to Heat Exchanger, Type K, ambient to 100°F, data logger
    - Primary Burner, Primary Air, Type K, ambient to 100°F, data logger
    - Primary Burner, Secondary Air, Type K, ambient to 900°F, data logger
    - Combustion Air Inlet to Auxiliary Burner, Type K, ambient to 100°F, data logger
  - Slagging Furnace
    - Furnace Interior Temperature No. 1, Type S, ambient to 3000°F, data logger
    - Furnace Interior Temperature No. 2, Type S, ambient to 3000°F, data logger
    - Furnace Interior Temperature No. 3, Type S, ambient to 3000°F, data logger
    - Furnace Interior Temperature No. 4, Type S, ambient to 3000°F, data logger
    - Furnace Wall/Refractory Temperature No. 1, Type S, ambient to 3000°F, data logger
    - Furnace Wall/Refractory Temperature No. 2, Type S, ambient to 3000°F, data logger
    - Furnace Wall/Refractory Temperature No. 3, Type S, ambient to 3000°F, data logger
    - Furnace Wall/Refractory Temperature No. 4, Type S, ambient to 3000°F, data logger
    - Furnace Exit Temperature No. 1, Type S, ambient to 3000°F, data logger
    - Furnace Exit Temperature No. 2, Type S HVT, ambient to 3000°F, data logger
    - Furnace Exit Temperature No. 3, optical method, ambient to 3000°F, data logger
    - Slag Tap/Slag Pot Flue Gas Exit Temperature, ambient to 2600°F, data logger
  - Radiant Air Heater Panels

- Large Panel Cooling Air Inlet Temperature, Type K, ambient to 1300°F, data logger
- Large Panel Cooling Air Outlet Temperature, Type K, ambient to 1800°F, data logger
- Large Panel Surface Temperature No. 1, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 2, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 3, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 4, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 5, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 6, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 7, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 8, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 9, Type K, ambient to 2000°F, data logger
- Large Panel Surface Temperature No. 10, Type K, ambient to 2000°F, data logger
- Small Panel No. 1 Cooling Air Inlet Temperature, Type K, ambient to 100°F, data logger
- Small Panel No. 1 Cooling Air Outlet Temperature, Type K, ambient to 1000°F, data logger
- Small Panel No. 1 Surface Temperature No. 1, Type K, ambient to 2000°F, data logger
- Small Panel No. 1 Surface Temperature No. 2, Type K, ambient to 2000°F, data logger
- Small Panel No. 1 Surface Temperature No. 3, Type K, ambient to 2000°F, data logger
- Small Panel No. 1 Surface Temperature No. 4, Type K, ambient to 2000°F, data logger
- Small Panel No. 2 Cooling Air Inlet Temperature, Type K, ambient to 100°F, data logger
- Small Panel No. 2 Cooling Air Outlet Temperature, Type K, ambient to 1000°F, data logger
- Small Panel No. 2 Surface Temperature No. 1, Type K, ambient to 2000°F, data logger
- Small Panel No. 2 Surface Temperature No. 2, Type K, ambient to 2000°F, data logger

- Small Panel No. 2 Surface Temperature No. 3, Type K, ambient to 2000°F, data logger
- Small Panel No. 2 Surface Temperature No. 4, Type K, ambient to 2000°F, data logger
- Small Panel No. 3 Cooling Air Inlet Temperature, Type K, ambient to 100°F, data logger
- Small Panel No. 3 Cooling Air Outlet Temperature, Type K, ambient to 1000°F, data logger
- Small Panel No. 3 Surface Temperature No. 1, Type K, ambient to 2000°F, data logger
- Small Panel No. 3 Surface Temperature No. 2, Type K, ambient to 2000°F, data logger
- Small Panel No. 3 Surface Temperature No. 3, Type K, ambient to 2000°F, data logger
- Small Panel No. 3 Surface Temperature No. 4, Type K, ambient to 2000°F, data logger
- Small Panel No. 4 Cooling Air Inlet Temperature, Type K, ambient to 100°F, data logger
- Small Panel No. 4 Cooling Air Outlet Temperature, Type K, ambient to 1000°F, data logger
- Small Panel No. 4 Surface Temperature No. 1, Type K, ambient to 2000°F, data logger
- Small Panel No. 4 Surface Temperature No. 2, Type K, ambient to 2000°F, data logger
- Small Panel No. 4 Surface Temperature No. 3, Type K, ambient to 2000°F, data logger
- Small Panel No. 4 Surface Temperature No. 4, Type K, ambient to 2000°F, data logger

- Slag Screen
  - Inlet Flue Gas Temperature (Furnace Exit No. 4), Type S, ambient to 3000°F, data logger
  - Outlet Flue Gas Temperature, Type S, ambient to 2700°F, data logger
- Dilution/Quench Zone
  - Dilution/Quench Gas Temperature, Type K, ambient to 500°F, data logger
  - Flue Gas Exit Temperature, Type K, ambient to 2000°F, data logger
- Convective Air Heater Section

- Flue Gas Inlet Temperature, Type K, ambient to 1800°F, data logger
- Flue Gas Exit Temperature, Type K, ambient to 1700°F, data logger
- Cooling Air Inlet Temperature, Type K, ambient to 1000°F, data logger
- Cooling Air Outlet Temperature, Type K, ambient to 1300°F, data logger

- Tube Surface Temperature No. 1, Type K, ambient to 1800°F, data logger
- Tube Surface Temperature No. 2, Type K, ambient to 1800°F, data logger
- Tube Surface Temperature No. 3, Type K, ambient to 1800°F, data logger
- Tube Surface Temperature No. 4, Type K, ambient to 1800°F, data logger
- Tube Surface Temperature No. 5, Type K, ambient to 1800°F, data logger
- Tube Surface Temperature No. 6, Type K, ambient to 1800°F, data logger

- Flue Gas Heat Exchangers
  - Heat Exchanger No. 1, Flue Gas Inlet, Type K, ambient to 1600°F, data logger
  - Heat Exchanger No. 1, Flue Gas Outlet, Type K, ambient to 1200°F, data logger
  - Heat Exchanger No. 1, Cooling Air Inlet, Type K, ambient, data logger
  - Heat Exchanger No. 1, Cooling Air Outlet, Type K, ambient to 1000°F, data logger

  - Heat Exchanger No. 2, Flue Gas Inlet, Type K, ambient to 1200°F, data logger
  - Heat Exchanger No. 2, Flue Gas Outlet, Type K, ambient to 800°F, data logger
  - Heat Exchanger No. 2, Cooling Air Inlet, Type K, ambient, data logger
  - Heat Exchanger No. 2, Cooling Air Outlet, Type K, ambient to 500°F, data logger

  - Heat Exchanger No. 3, Flue Gas Inlet, Type K, ambient to 800°F, data logger
  - Heat Exchanger No. 3, Flue Gas Outlet, Type K, ambient to 400°F, data logger
  - Heat Exchanger No. 3, Cooling Air Inlet, Type K, ambient, data logger
  - Heat Exchanger No. 3, Cooling Air Outlet, Type K, ambient to 400°F, data logger

  - Heat Exchanger No. 4, Flue Gas Inlet, Type K, ambient to ?°F, data logger
  - Heat Exchanger No. 4, Flue Gas Outlet, Type K, ambient to ?°F, data logger
  - Heat Exchanger No. 4, Cooling Air Inlet, Type K, ambient to ?°F, data logger
  - Heat Exchanger No. 4, Cooling Air Outlet, Type K, ambient to ?°F, data logger

- Heat Exchanger No. 5, Flue Gas Inlet, Type K, ambient to ?°F, data logger
- Heat Exchanger No. 5, Flue Gas Outlet, Type K, ambient to ?°F, data logger
- Heat Exchanger No. 5, Cooling Air Inlet, Type K, ambient to ?°F, data logger
- Heat Exchanger No. 5, Cooling Air Outlet, Type K, ambient to ?°F, data logger

- Heat Exchanger No. 6, Flue Gas Inlet, Type K, ambient to ?°F, data logger
- Heat Exchanger No. 6, Flue Gas Outlet, Type K, ambient to ?°F, data logger
- Heat Exchanger No. 6, Cooling Air Inlet, Type K, ambient to ?°F, data logger
- Heat Exchanger No. 6, Cooling Air Outlet, Type K, ambient to ?°F, data logger

- Baghouse
  - Flue Gas Inlet Temperature, Type K, ambient to 750°F, data logger
  - Flue Gas Outlet Temperature, Type K, ambient to 750°F, data logger

  - Baghouse Flue Gas Temperature No. 1, Type K, ambient to 750°F, data logger
  - Baghouse Flue Gas Temperature No. 2, Type K, ambient to 750°F, data logger
  - Baghouse Flue Gas Temperature No. 3, Type K, ambient to 750°F, data logger
  - Baghouse Flue Gas Temperature No. 4, Type K, ambient to 750°F, data logger
  - Baghouse Wall Temperature No. 1, Type K, ambient to 750°F, data logger
  - Baghouse Wall Temperature No. 2, Type K, ambient to 750°F, data logger
  - Baghouse Wall Temperature No. 3, Type K, ambient to 750°F, data logger
  - Baghouse Wall Temperature No. 4, Type K, ambient to 750°F, data logger
  - Baghouse Hopper Wall Temperature, Type K, ambient to 750°F, data logger
- ID Fan/Stack
  - ID Fan Flue Gas Inlet Temperature, Type K, ambient to 500°F, data logger
  - Stack Flue Gas Temperature, Type K, ambient to 500°F, data logger

- **Static and Differential Pressure Measurements**
  - Combustion Air System
    - Total Air Static and Differential Pressure for Primary Burner, differential pressure (dP) data logger and control panel
    - Primary Air Static and Differential Pressure for Primary Burner, control panel
    - Secondary Air Static and Differential Pressure for Primary Burner, control panel
    - Total Air Static and Differential Pressure for Auxiliary Burner, control panel

- Primary Air Static and Differential Pressure for Auxiliary Burner, control panel
- Secondary Air Static and Differential Pressure for Auxiliary Burner, control panel

- Slagging Furnace
  - Furnace Static Pressure, data logger and control panel
- Radiant Air Heater Panels
  - Large Panel Cooling Air Inlet Pressure, data logger and control panel
  - Large Panel Cooling Air Outlet Pressure, data logger and control panel
  - Large Panel Cooling Air Differential Pressure (inlet flow rate), data logger and control panel

  - Small Panel No. 1 Cooling Air Inlet Pressure, data logger and control panel
  - Small Panel No. 1 Cooling Air Outlet Pressure, data logger and control panel
  - Small Panel No. 1 Cooling Air Differential Pressure (inlet flow rate), data logger and control panel

  - Small Panel No. 2 Cooling Air Inlet Pressure, data logger and control panel
  - Small Panel No. 2 Cooling Air Outlet Pressure, data logger and control panel
  - Small Panel No. 2 Cooling Air Differential Pressure (inlet flow rate), data logger and control panel

  - Small Panel No. 3 Cooling Air Inlet Pressure, data logger and control panel
  - Small Panel No. 3 Cooling Air Outlet Pressure, data logger and control panel
  - Small Panel No. 3 Cooling Air Differential Pressure (inlet flow rate), data logger and control panel

  - Small Panel No. 4 Cooling Air Inlet Pressure, data logger and control panel
  - Small Panel No. 4 Cooling Air Outlet Pressure, data logger and control panel
  - Small Panel No. 4 Cooling Air Differential Pressure (inlet flow rate), data logger and control panel
- Slag Screen
  - Differential Pressure, control panel and data logger
- Dilution/Quench Zone
  - Flue Gas Static and Differential Pressure, control panel

    - Dilution/Quench Gas Static and Differential Pressure (flow rate), control panel, dP data logger
  - Convective Air Heater Section
    - Flue Gas Differential Pressure, data logger and control panel
    - Flue Gas Static Pressure, control panel
    - Cooling Air Inlet Pressure, data logger and control panel
    - Cooling Air Outlet Pressure, data logger and control panel
    - Cooling Air Differential Pressure (inlet flow rate), data logger and control panel
  - Flue Gas Heat Exchangers
    - Differential and Static Pressure for Each Heat Exchanger (Nos. 1 through 6), control panel
  - Baghouse
    - Differential Pressure, data logger and control panel
    - Outlet Static Pressure, control panel
  - ID Fan/Stack
    - Inlet Static Pressure, control panel
    - Inlet Differential Pressure (flue gas flow rate), data logger and control panel
    - Stack Static Pressure, control panel
- Flow Rate Measurements
  - Combustion Air System
    - Total Combustion Air Flow Rate, primary burner, data logger and control panel
    - Primary Air Flow Rate, primary burner, control panel
    - Secondary Air Flow Rate, primary burner, control panel
    - Total Combustion Air Flow Rate, auxiliary burner, data logger, and control panel
    - Primary Air Flow Rate, auxiliary burner, control panel
    - Secondary Air Flow Rate, auxiliary burner, control panel
  - Slagging Furnace
    - None
  - Radiant Air Heater Panels
    - Large Panel Cooling Air Flow Rate, data logger, and control panel

- Small Panel No. 1 Cooling Air Flow Rate, data logger, and control panel
- Small Panel No. 2 Cooling Air Flow Rate, data logger, and control panel
- Small Panel No. 3 Cooling Air Flow Rate, data logger, and control panel
- Small Panel No. 4 Cooling Air Flow Rate, data logger, and control panel

- Slag Screen
  - None
- Dilution/Quench Zone
  - Flue Gas Recirculation Flow Rate, data logger, and control panel
- Convective Air Heater Section
  - Cooling Air Flow Rate, data logger, and control panel
- Flue Gas Heat Exchangers
  - None required; install pitot ports for periodic measurements.
- Baghouse
  - None required; install pitot ports for periodic measurements.
- ID Fan/Stack
  - Flue Gas Flow Rate, ID fan inlet, data logger, and control panel

The data acquisition system will be based on a Genesis software package and a personal computer. This type of data acquisition system is currently used at the EERC on a number of pilot–scale units. All process data points will be logged on the data acquisition system. However, the EERC is uncertain, at this time, of how much integrated system control will be required or desired. Decisions concerning the level of integrated control implemented will be made based on relative cost.

ASH REMOVAL

Proper slag flow and slag corrosion are major operational concerns for the HITAF. Since the viscosity of the slag controls its flow characteristics, as well as the rate of mass transport of corrodents, it is important to delineate the factors affecting slag viscosity, so that accurate predictions of flow and corrosion in different regions of the HITAF can be made. The factors affecting viscosity include slag composition, local atmosphere, and slag temperature. By controlling these factors in the HITAF, proper slag flow can be assured while minimizing the corrosion and erosion of critical subsystems. In order to ascertain the effects of slag composition, local atmosphere, and slag temperature on slag viscosity, laboratory measurements are being performed. For this quarter, we measured the changes in viscosity versus temperature curves of two slags as slag composition and local atmosphere were varied.

Experimental

The slags were collected under another project from the taps of cyclone–fired utility boilers. Subbituminous coal slag was provided by the Northern States Power Company Riverside Plant, which was burning a coal from the Rochelle Mine of the Powder River Basin in Wyoming. In addition, slag was prepared from ash made by burning the Rochelle Mine coal used in Phase I in a laboratory furnace. The laboratory ash tests will provide a direct comparison between the current tests and those performed under Phase I. A paper describing the results of the Rochelle coal ash slag viscosity tests was prepared for presentation at the American Chemical Society National Meeting in March 1996. Entitled "The Effects of Atmosphere and Additives on Coal Slag Viscosity," it is included as an appendix to this quarterly report.

Illinois No. 6 bituminous coal slag was provided by the Central Illinois Public Service Coffeen Plant and by the Illinois Power Company Baldwin Plant. Two Illinois No. 6 slags were collected and tested because the limestone was added to the Coffeen Plant coal to reduce slag viscosity, so it is abnormally high in calcium. The power plant slags will be used throughout the laboratory testing as standard materials. They are also available to other researchers for use in standardized corrosion testing.

The slags were prepared by melting either the ash or existing slag at 1500°C in air in a platinum crucible and then quenching them on a brass plate at room temperature. The slags were crushed, placed in an alumina crucible, and reheated to 1500°C for viscosity testing. Viscosities were measured with a Haake RV–2 Rotovisco system with a DMK 50/500 dual measuring head, which is a rotating bob viscometer. The bob is immersed until the slag just covers its top and then is rotated at 64 rpm. Testing in an oxidizing atmosphere requires the use of a shaft and bob fabricated from platinum–rhodium (90/10) alloy. The bob is 9 mm in diameter and 20 mm long and has a 45° taper at each end. For testing in a reducing atmosphere, the shaft and bob were fabricated from molybdenum and had dimensions of 11.6 mm in diameter and 23.4 mm long and a 45° taper at each end. The torque applied to the viscometer is read and converted to an electrical signal, which is then sent to a computer with a data acquisition–manipulation program that determines the viscosity of the slag. The viscosity of the slags were measured over the range of 10 to as high as 4000 poise, unless crystallization was seen to occur along the walls of the alumina container, at which point the measurements were terminated. Measurements were started at least 100°C above the temperature of critical viscosity (T_c) because above this point the slag viscosity is relatively low and decreases linearly with increasing temperature. The temperature was decreased in 20°C intervals as readings were taken. The viscometer was calibrated with both National Bureau of Standards silicate glasses 710 and 717 for each type of bob. The accuracy of the viscosity tests is approximately ± 5%.

The viscosities of the slags were measured under three atmospheres: air, air + 10% water vapor, and $H_2/CO/CO_2$ + 10% water vapor. Air was used because it is a standard atmosphere and so is included so that comparisons can be drawn between these results and those in other research. However, we believe that the air + 10% moisture tests better simulate the atmosphere in oxidizing portions of the HITAF. To simulate the regions of the HITAF near the flame where gas compositions may be reducing, an atmosphere of 31% H_2, 45% CO, and 24% CO_2 to which 10% water vapor was added was used. Because of equilibrium reactions at this temperature, we have calculated that the concentrations of CO may have risen as high as 49%, H_2O as high as 18%, CO_2 as low as 13%, and H_2 as low as 20% at 1400°C. To test the effects of additives on the Rochelle coal ash slag viscosity, reagent–grade magnesium oxide (45 wt%) or aluminum oxide (20 wt%) was added to the slag under air + 10% water vapor conditions. Compositions of the original ashes are listed in Table 2.

Table 2. Compositions of Coal Ashes, wt% Oxide Basis

	Coal Ash Rochelle	NSP Rochelle	Coffeen Ill. No. 6	Baldwin Ill. No. 6
SiO_2	38.8	47.1	52.5	53.4
Al_2O_3	20.8	18.6	16.3	18.6
Fe_2O_3	6.1	5.3	13.6	17.6
TiO_2	1.5	1.4	0.7	0.7
P_2O_5	1.1	0.6	0.2	0.0
CaO	23.2	19.8	13.1	7.1
MgO	6.5	5.7	1.2	0.9
Na_2O	1.4	0.9	0.8	0.0
K_2O	0.3	0.4	1.6	1.7
SO_3	ND[1]	0.3	0.1	ND[1]

[1] Not detected.

Results and Discussion

Figure 23 shows the viscosity–versus–temperature curves for the Rochelle coal ash slag in air, air + 10% moisture, and reducing conditions. The curves are very similar for both the air and air + moisture tests, with T_c approximately 1250°C and the viscosity of each slag around 20 poise at 1400°C. However, under the reducing conditions, T_c dropped considerably to 1180°C, and the viscosity of the slag at 1400°C was only around 5 poise. These changes cannot be explained merely by the reduction of iron in the slag from a network former (Fe^{3+} ion) to a network modifier (Fe^{2+} ion) since there is so little iron in the slag.

Figure 24 shows the effects of the additions of alumina and magnesia to the Rochelle coal ash slag when measured in air + 10 moisture. As was true for the measurements in air, adding alumina raised the viscosity and T_c of the slag, but much more substantially, with the viscosity increased by 60 poise at 1400°C while T_c rose by 140°C. This increase in viscosity would reduce the corrosivity of the slag to energy system materials, dramatically reducing the corrosivity of the slag at temperatures below 1400°C. The additions of magnesia also created similar changes as when measured in air. The viscosity of the slag at 1400°C dropped by approximately 10 poise, but T_c rose by only 50°C to 1310°C.

Figure 25 shows the curves for the NSP plant slag produced while burning Rochelle coal. There was much less difference in the viscosity curves produced under the different atmospheres than was measured for the coal ash slag, although generally the viscosity was somewhat higher and T_c was much higher for the NSP plant slag. These increases are likely caused by the higher silica content of the NSP slag.

Figure 26 shows the curves for the high–calcium Illinois No. 6 slag from the Coffeen plant. Like the NSP slag, the atmosphere did not have a large effect on T_c or the viscosity above that temperature. Like the Rochelle coal ash slag, however, the slags in air + moisture and in reducing conditions remained liquid to lower temperatures than did the slag in air. However, in general, this slag had a much higher viscosity and somewhat higher T_c than the NSP Rochelle slag.

Figure 27 shows the curves for the Illinois No. 6 slag from the Baldwin Plant. As was true for the NSP Rochelle and Coffeen slags, atmosphere did not have a large effect on viscosity or T_c, although the slag generally had a higher viscosity and somewhat higher T_c than the higher calcium Coffeen slag.

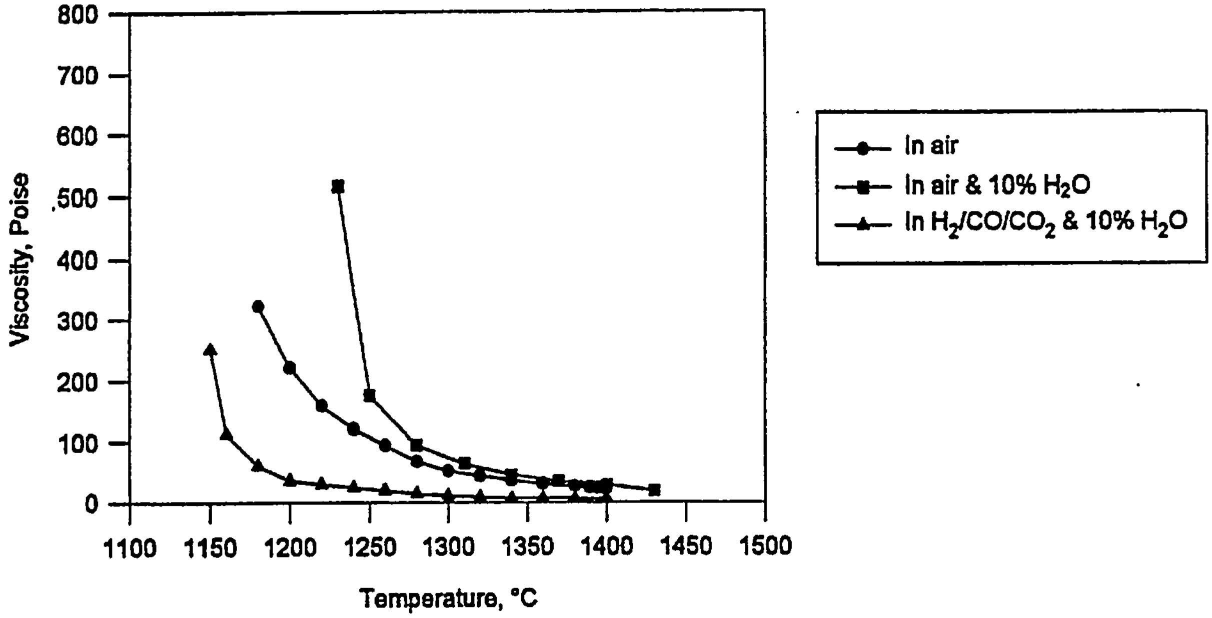

Figure 23. Viscosity–Versus–Temperature Curves for the Rochelle Coal Ash Slag in Air, Air + 10% Moisture, and Reducing Conditions.

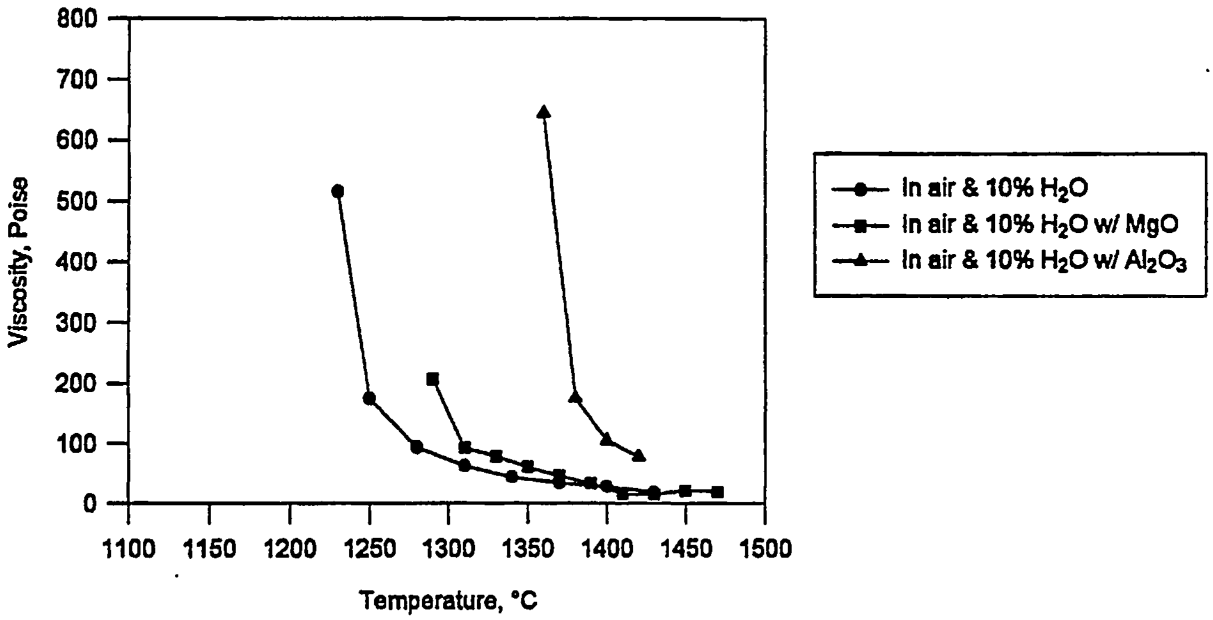

Figure 24. Viscosity–Versus–Temperature Curves Measured in Air + 10% Moisture for the Rochelle Coal Ash Slag, Slag Plus Alumina, and Slag Plus Magnesia.

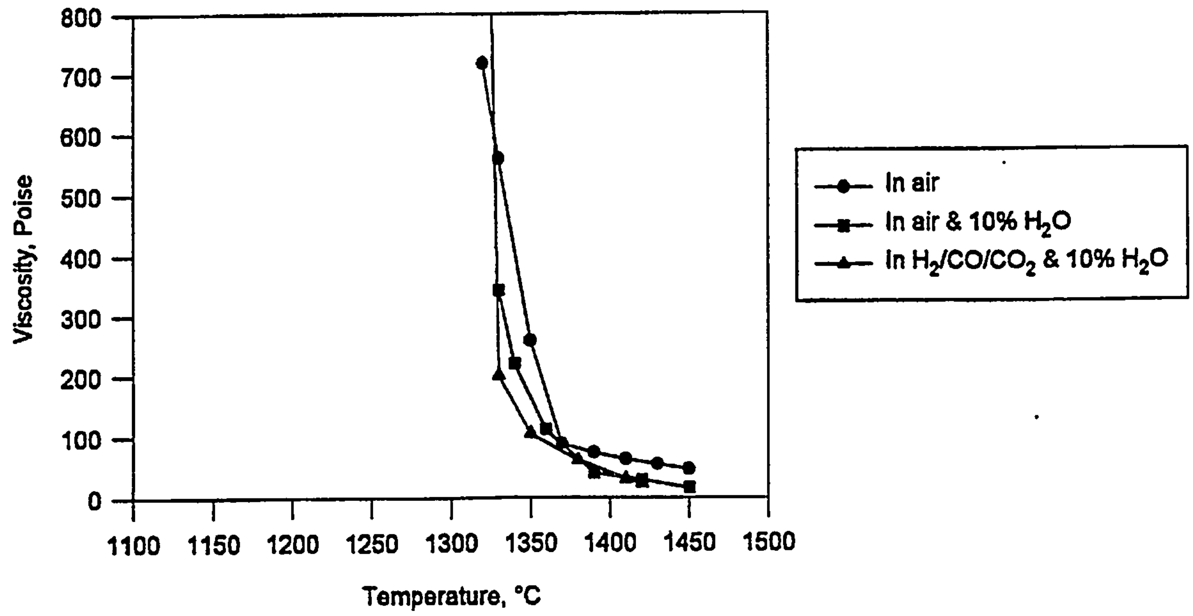

Figure 25. Viscosity–Versus–Temperature Curves for the NSP Plant Rochelle Slag in Air, Air + 10% Moisture, and Reducing Conditions.

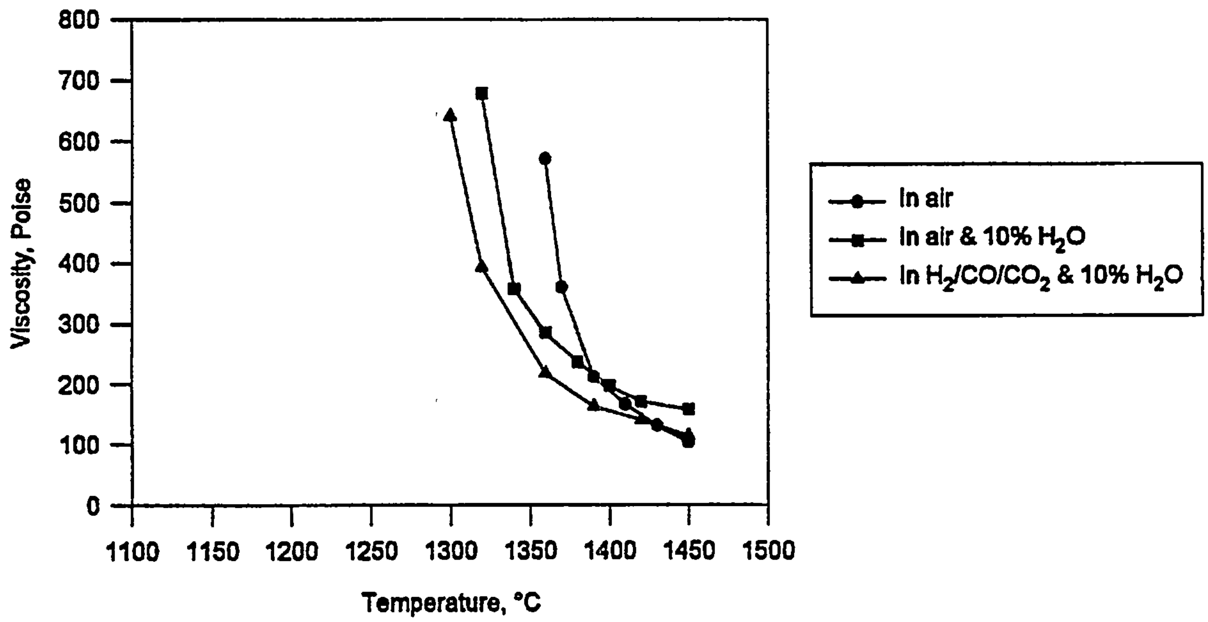

Figure 26. Viscosity–Versus–Temperature Curves for the Coffeen Plant High–Calcium Illinois No. 6 Slag in Air, Air + 10% Moisture, and Reducing Conditions.

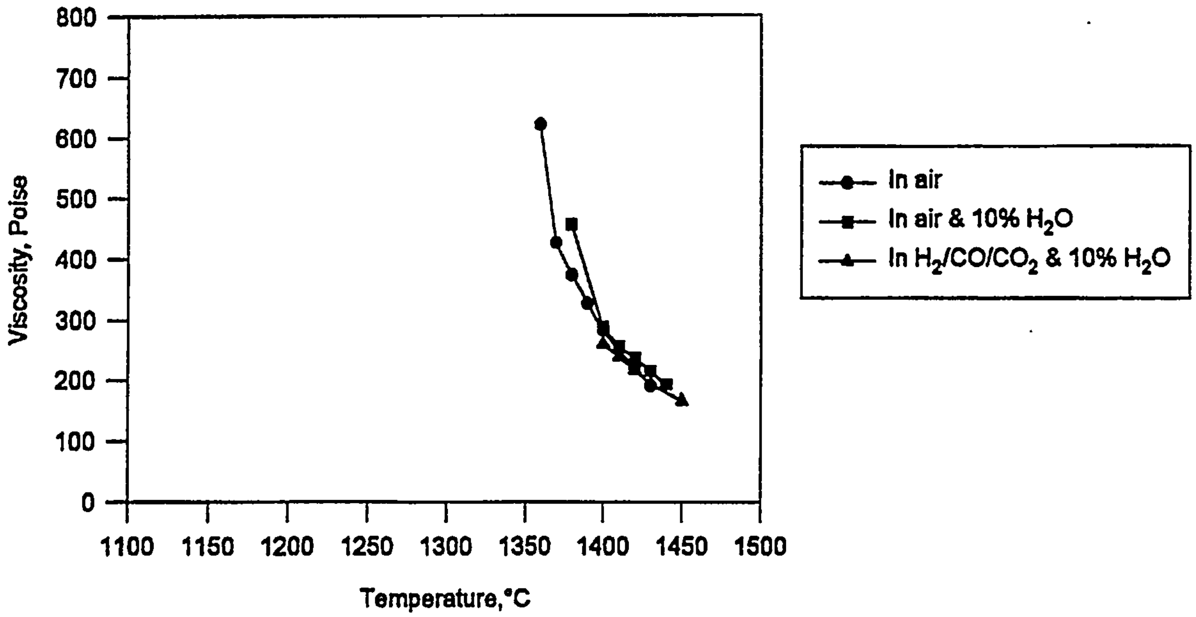

Figure 27. Viscosity–Versus–Temperature Curves for the Baldwin Plant Illinois No. 6 Slag in Air, Air + 10% Moisture, and Reducing Conditions.

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