A simulation approach to the thermal-hydraulic design of cored ceramic brick regenerative heat exchangers

by Gary Alan Upshaw

A thesis submitted in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in Mechanical Engineering
Montana State University

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Simulations using variations in tube roughness show minimal difference in total ceramic mass requirements. The model was used to determine the sensitivity of an air preheater performance to variation in flow and geometric parameters. Thermal operational characteristics of a cored brick heat exchanger test facility are predicted for several variations in flow rates and total cycle times.

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Date 5/17/77

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by

GARY ALAN UPSHAW

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of

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Nomenclature

a	=	inner radius of tube	[m]
A1	=	flow area	[m^2]
A2	=	area enclosed by adiabatic surface	[m^2]
BTI	=	blowdown time	[sec]
b	=	equivalent outside tube radius	[m]
C_p	=	specific heat	[J/kg-K]
D	=	hole diameter	[m]
D2	=	equivalent outside tube diameter	[m]
E	=	modulus of elasticity of ceramic	[Pa]
f	=	Darcy-Weisbach friction factor	
h	=	convective heat transfer coefficient between gas and ceramic surface	[W/m^2-K]
h.x.	=	heat exchanger (abbreviation)	
k	=	thermal conductivity	[W/m-K]
K_s	=	tube roughness	[m]
L	=	bed length	[m]
ln	=	logarithm to base e	
$\dot{m}$	=	mass flow rate	[kg/sec]
NB	=	number of heat exchangers in blowdown mode	
NR	=	number of heat exchangers in reheat mode	
Nu	=	local Nussult number (hD/k_g)	

Nomenclature

P1	=	inlet reheat gas pressure	[Pa]
P2	=	inlet blowdown air pressure	[Pa]
Pr	=	Prandtl number ($C_{pg}\ \mu_g/k_g$)	
Q	=	heat flux between gas and ceramic	[W/m^2]
R	=	gas constant	[J/kg-K]
Re	=	Reynolds number ($\mu_g V_g D/\mu_g$)	
RTI	=	reheat time	[sec]
S	=	hole to hole spacing	[m]
S/D	=	ratio of hole spacing to hole diameter	[m/m]
STI	=	switchover time	[sec]
T	=	temperature	[K]
T(J,I)	=	temperature of ceramic in lumped mass model at axial location J and time I. Time I=1 is present time, t, and time I=2 is time at t+Δt.	[K]
TEXIT	=	average exit gas temperature	[K]
TG(J)	=	gas temperature in lumped mass model at axial position J	[K]
TGI1	=	inlet reheat gas temperature	[K]
TGI2	=	inlet blowdown air temperature	[K]
t	=	time	[sec]
Δt	=	time increment	[sec]
V	=	local mean gas velocity	[m/sec]

Nomenclature

V1	=	inlet reheat gas velocity	[m/sec]
V2	=	inlet blowdown air velocity	[m/sec]
X	=	axial distance along tube	[m]
Z	=	axial direction coordinate	
ΔZ	=	length increment in axial direction	[m]
α	=	thermal diffusivity $(k_c/\rho_c C_{pc})$	
α_1	=	$\dfrac{2\ h\ a\ \Delta t}{\rho_c C_{pc}(b^2-a^2)}$	
α_2	=	$\dfrac{2\ h\ a\ \Delta Z}{\dot{m}\ C_{pg}}$	
α_3	=	$\dfrac{k\ \Delta t}{\rho_c C_{pc}\ \Delta Z^2}$	
α_6	=	$\dfrac{h\ a\ b^2}{k_c(b^2-a^2)}\left[\dfrac{a^2}{2b^2} - \dfrac{b^2+a^2}{4b^2} + \dfrac{b^2}{b^2-a^2}\ln\left(\dfrac{b}{a}\right) - \dfrac{1}{2}\right]$	
β	=	thermal coefficient of expansion for ceramic	[m/m-K]
μ	=	dynamic viscosity	[Pa-sec]
ν	=	Poisson's ratio for ceramic	
ρ	=	density	[kg/m^3]
σ	=	thermal stress	[Pa]
Δ	=	increment designation	

Subscripts

a	=	inner wall
avg	=	average
BD	=	blowdown
C	=	ceramic
conv	=	convection
g	=	gas
0	=	smooth pipe
RH	=	reheat
r	=	rough pipe

ABSTRACT

Using the cored brick regenerative heat exchanger as a basis, a method for designing heat exchangers to supply preheated air for open cycle magnetohydrodynamic power generation systems was developed. Design information and significant performance trends for multiple unit MHD air preheater systems were explored using a digital computer model which simulated operation of a cored brick regenerative heat exchanger. Thermal-hydraulic designs using numerous variations of core geometry and cycling parameters were completed to meet flow and temperature specifications of a typical full scale air preheater system. Unique values for flow rates, heat exchanger core sizing and ceramic mass requirements were calculated for each of these variations using criteria of limited thermally induced stress in the ceramic, specific discharge temperature for the preheated air stream and limited blowdown exit temperature droop.

Simulations using variations in tube roughness show minimal difference in total ceramic mass requirements. The model was used to determine the sensitivity of an air preheater performance to variation in flow and geometric parameters. Thermal operational characteristics of a cored brick heat exchanger test facility are predicted for several variations in flow rates and total cycle times.

Chapter I

INTRODUCTION

An important component of proposed coal-fired open-cycle magnetohydrodynamic (MHD) power plants is the high temperature air preheater. High cycle efficiencies associated with MHD depend on passing a sufficiently high temperature, ionized, combustion gas stream through the magnetic field of the MHD generator channel. The combustion of coal and ambient air, without the benefit of preheated air or enrichment with oxygen will not produce a gas stream hot enough to operate the MHD generator channel at high efficiencies.

Cored ceramic brick regenerative heat exchangers are presently being considered as a means of preheating air for combustion with coal in MHD systems. The air preheater system in a full scale, coal burning, open cycle MHD system will represent a significant portion of the over-all system costs. The design of an efficient cored-brick regenerative air preheater facility with emphasis on the economical use of the ceramic core of the regenerator is a main area of consideration in this thesis. Reliability of the air preheater system is incorporated into the design procedure by limiting maximum values of thermally induced stress within the bricks.

The operation of a regenerative heat exchanger includes two basic functions. First, hot fluid passes through some intermediate energy storage medium where energy in the form of heat is transferred from the hot fluid to the cooler storage medium. This phase is referred to as the reheat cycle. During the second operation, the blowdown cycle, heat which has been deposited in the storage medium is transferred to a cooler, counter-flow fluid stream. The cooler blowdown stream is heated and the regenerator storage matrix is cooled. The cycle is completed as the blowdown stream is terminated and the hot reheat stream once again flows into the opposite end to reheat the bed.

The energy storage medium can be of several types, but generally there are but two basic classifications. The first of these is the fixed bed or stationary type of regenerator, which operates on a cyclical pattern of alternate heating and cooling. The second method is the falling bed type regenerator, in which the heat storage medium consists of small particles of refractory material. The refractory particles or pebbles enter one of two vertical chambers where they are heated by a hot fluid. From this chamber the pebbles move into a second, usually lower, chamber where they give up their stored heat to a cooler gas. This heated gas leaves the second chamber at a higher temperature and the pebbles leave at a lower temperature than they entered. The pebbles then return to the upper chamber to again be heated and the process is repeated. The process operates continuously with no defined cycle periods, no reversals of flow and a constant discharge temperature of the blowdown gas. The falling bed heat exchanger has advantages of delivering a constant temperature blowdown gas, requiring less valve operation, and using a less expensive heat storage medium. Its disadvantages include the problem of point to point contact between the pebbles which can present dusting and/or spalling problems, difficulties in maintaining a pressure difference between the streams, and in applications where the hot gas contains coal slag, the particles are susceptible to sticking together after they are coated with the slag.

The storage medium in the fixed bed regenerator is typically a refractory material which is selected for the temperature ranges it will encounter and the type of atmosphere to which it will be exposed. The core or matrix geometry of the refractory fixed bed regenerative heat exchanger may have three general configurations:

1) The matrix may be built from rectangular refractory bricks arranged into a number of unique patterns.
2) The matrix may be composed of a bed of spherical pebbles.
3) The matrix may be formed from cored ceramic with the holes aligned to form the flow passages.

This thesis considers only fixed bed regenerative heat exchangers with cored ceramic brick matrices. Typically the cored bricks have a hexagonal shape and the holes are arranged in a equilateral pattern as shown in Figure 1. The brick hole diameter, D, and the hole-to-hole spacing, S, are two important parameters describing a brick configuration. Various experiments [18,20] have used holes as small as 6.35 mm (.25in.) to test cored brick air heaters. However, these regenerators were heated by clean burning fuels and were not subject to adhesion of seed and slag on the tube walls as are coal fired regenerators. Because of the non-abundance of clean burning fuels, coal is considered as the primary source for fueling MHD power systems. Typically the combustion gas stream is seeded with from 1 to 5% with potassium or cesium to enhance the ionization potential of the gases passing through the MHD channel. It is this mixture of coal combustion products and seed that pass through the air preheater during reheat in direct fired MHD cycles.

The MHD cycle may be designed to use a direct fired or indirect fired air preheater. Direct firing utilizes the exhaust gas stream from the MHD generator as the hot fluid during reheat. The energy of this stream is used to heat the regenerator matrix as is shown in Figure 2a. Indirect fired air preheaters operate by heating the regenerator matrix with a clean fuel, such as natural gas or gasified coal. The important consideration for using indirect fired preheaters is the absence of problems attributed to contamination by seed and the products of coal combustion on the heater matrix, and the build up of seed and slag on the matrix walls. An operating diagram of a typical open cycle MHD system using indirect firing of air preheaters is shown in Figure 2b, with the steam bottoming cycle using the hot gases emerging from the MHD generator to superheat steam. The cooler gases then pass through a low temperature air preheater which has the advantage of being exposed to a cooler, less corrosive (because of condensation) seed and slag laden exhaust gas. The high temperature heater is not exposed to the coal combustion exhaust, but rather to the hot gases resulting from the residue free combustion of a clean fuel source. The main drawbacks of indirect fired preheater systems are associated with cycle efficiencies and fuel sources. Systems using direct fired preheaters, in theory, achieve an overall cycle efficiency 2 percent above the indirect fired system for a given preheat temperature [25]. Natural, clean burning fuel supplies are limited in quantity and gasification of coal requires a significant fraction of the total available heating value in a mass of coal.

With the exception of the interaction between the matrix material and the seed and slag laden products of high temperature coal combustion, open cycle MHD systems can effectively use a direct fired air preheater subsystem. Careful consideration must be given in the selection of materials which will withstand the combination of high temperatures and a corrosive atmosphere.

This thesis develops and analyzes thermal designs of cored ceramic brick heat exchangers suitable for open cycle MHD use.

Designs were completed which satisfy air preheater specifications for temperature and flow rates while operating below specified thermally induced stress levels in an effort to define core geometries providing efficient and economical use of the ceramic material.

Chapter II

Literature Review

The regenerative heat exchanger found its first use with Dr. Robert Stirling in 1816 who constructed a regenerator for a hot-air engine. Early analytical treatment of regenerator operation by Nusselt [12,13] produced solutions to the differential equations describing regenerators. His analyses have served as a basis for numerous analytical models simulating regenerators. Iliffe [11] simplified the solution of Nusselt somewhat, but the method involved the segmenting of the heater length into nodes, which, for each node required an hour of manual calculation to determine temperature changes and heat transfer.

Hausen [14,15,16] developed several methods for analyzing regenerative heat exchangers. In 1929 Hausen [14] developed the characteristic solution for equations describing an ideal regenerator which combined the heat transfer relations for the reheat and blowdown periods into a single relation which satisfied the reversal of flow direction and change in inlet temperature conditions. Hausen's "heat pole" method [15] for regenerators assumes a constant initial temperature distribution and utilizes a series of heat pole functions for describing transient temperature changes along an axially segmented bed. The heat pole functions are determined for both reheat and blowdown periods and a set of simultaneous equations results which is solvable for the regenerator axial temperature distribution in each cycle. In 1942 Hausen [16] derived an exact mathematical expression for relating the mean ceramic temperature at a point along a regenerator to the surface temperature and the fluid temperature at that point using a modified heat transfer coefficient. This model for regenerator operation assumed constant gas and ceramic properties and a constant axial temperature gradient in the gas stream.

Hausen's, Nusselt's and Iliffe's methods were adaptable to solution without the aid of a computer but involved extensive calculations and lengthy mathematical manipulations.

Butterfield, et al [10] verified Hausen's [16] method with a digital computer analysis and presented a modified Hausen procedure which accounted for variations in heat transfer coefficients, variations in flow rates and specific heats and included radiation effects.

Contemporary thermal simulations of regenerative heat exchangers have taken advantage of the digital computer to make evaluation of regenerators more feasible. Wilmott's [1] primary work expressed the differential equations governing regenerative heat exchangers in a difference equation form. An algorithm was used to successively simulate each cycle from an arbitrary beginning until an equilibrium condition was reached. Wilmott, as Hausen and Butterfield also did, relates the heat transfer coefficient used in convection to the mean solid temperature rather than the surface temperature. Later, Wilmott

[3] incorporated into his model a variable mass flow of the working fluids to achieve a constant temperature stream both entering and leaving the regenerator. Wilmott found that using a variable flow operation reduces the effectiveness of the regenerator to transfer heat from the hot stream to the cold stream. Wilmott [2] also considered a three dimensional analysis of the regenerator which included the transverse as well as the axial temperature gradients in the matrix. Heat transfer in the transverse direction is also included in the method presented by Handley and Heggs [6] for simulating regenerators with matrix material laying parallel to a fluid flow, as in a cored brick heater. Reihman, Townes, and Mozer [31] have shown that detailed simulation of the transverse temperature distribution is not required for modeling cored brick regenerative heat exchangers.

Razelos and Lazaridas [8] developed a method adaptable to computer solution for calculating the transverse transient temperature distribution in a regenerative heat storage medium, relating the mean temperature from that distribution to the convective surface temperature, and modifying the heat transfer coefficient to account for resistance to heat transfer at the convective surface.

The model developed in this thesis extends Wilmott's method [1] to account for changes in local gas and ceramic temperature dependent properties, and utilizes the quasi-steady portion of the transverse temperature distribution of Razelos and Lazaridas [8] to relate the surface temperature of the ceramic to the average ceramic temperature. Rather than modifying the heat transfer coefficient as Wilmott, and Razelos and Lazaridas did, the model used in this thesis relates the surface temperature to the average ceramic temperature and the local heat transfer coefficient for convection calculations.

Larsen et al [5] has proceeded with the solutions of Nusselt [12], Hausen [14], and Schumann [24] and developed a solution independent of initial ceramic temperatures and any time variation in inlet gas temperatures for finding fluid and solid temperatures at any time and location in a regenerative heat exchanger. Solution by this method involves approximating the arbitrary initial or inlet conditions as a number of linear segments and superimposing the results of the linear segments as a superposition type solution. The method uses tables and curves in the solution but does not require the use of a digital computer.

Equilibrium conditions are found without modeling successive cycles in the non-iterative technique presented by Edwards et al [4]. The method does not require complete digital simulation to determine temperature variations in regenerative heat exchangers; however, it does require considerable numerical approximation and matrix manipulation.

Regenerative heat exchangers have long been used in the iron and steel industry to economize the blast and open hearth furnace operations, However, these heaters were mostly the type with a bulky matrix of stacked rectangular bricks. More recently, cored bricks have been developed in the aerospace industry to provide high temperature air for hypersonic wind tunnels and propulsion facilities. Various studies and experimental work [9,18,23,24] on high temperature cored brick storage heaters provide design techniques, thermal analyses and material evaluations, and demonstrate the ability of cored brick air heaters to provide preheat temperatures in excess of 2400 K.

Heywood and Womack [25] discuss the feasibility of various types of air preheaters and provide an overview of air heaters as they apply to MHD. Chojnowski et al [22] designed a regenerative heat exchanger of stacked rectangular bricks for preheating air in an open cycle MHD system.

To date, no directly coal-fired, cored brick regenerative heat exchangers are used in a full scale operation. Research effort is presently being directed towards applying the cored brick heat exchanger to high temperature energy conversion systems with emphasis on MHD [17,19,20,21]. Pilot heaters have been built to determine contaminant levels of an argon stream for a closed cycle MHD study [17], to define the reliability of an air preheater operating on a continual cycling basis [20,21], and to test both the corrosion of ceramics and the plugging of a flow stream by seed and slag [19,20,21].

Chapter III

Methods

3.1 Cored Ceramic Brick Regenerative Heat Exchangers

In this section, operating characteristics of cored ceramic brick heat exchanger systems are discussed. Relations between cycling times and the number of heat exchangers in each mode, and the causes of temperature droop in the exit gas stream are described. Design constraints and effects of minimizing cored brick hole diameter and S/D ratios are also included.

3.1.1 Operation of Heat Exchanger Systems

The purpose of the high temperature heat exchanger in a MHD generating system is to provide a constant mass flow of air which has been heated to a specified temperature. Preheat air at temperatures less than those specified tends to decrease the combustion flame temperature and results in a reduction in the overall efficiency of the MHD system.

The fixed bed heat exchanger must regularly undergo a regeneration period to maintain continuous operation. The MHD combustor must also receive a constant flow rate of heated air. Continuous operation requires a specific number of heat exchangers in the blowdown mode and a specific number in reheat at any given time. Theoretically, the minimum number of heat exchangers required would be two; one carrying the reheat gas and heating the matrix; the other carrying blowdown air being heated by a hot matrix. However with an operation of this sort, maintaining a constant flow of constant temperature to the combustor would be difficult. Valving, pressurization and purging would have to take place instantaneously to insure continuous flow and the preheated air would be difficult to maintain at a constant temperature. The initial discharge air from the heat exchanger at the start of blowdown would be the highest temperature and would continually emerge at lower temperatures while the matrix energy was given up to the blowdown air. For precise control of flow and temperatures using a two heat exchanger system, auxiliary bypass streams would be required. Willmott [3] has shown that using a variable flow operation reduces the effectiveness of the regenerator to transfer that from the hot stream to the cold stream.

Regenerative heat exchangers experience changes in exit gas temperature with time. The magnitude of this variation for any cycle is called "temperature droop." The temperature droop from the beginning to the end of the blowdown cycle has three basic characteristics;

1. The longer the cycle time, the more droop there is.
2. The amount of droop is dependent upon the amount of ceramic heated and cooled.
   a) Large hole diameters increase ceramic mass and lead to less droop.
   b) Larger ratios of hole spacing to hole diameter (S/D) increase the mass of ceramic cooled and

heated and reduce the temperature droop.

3. The rate of heat removal from the ceramic bed affects the temperature droop. Higher blowdown velocities raise the heat transfer coefficient and the rate of heat removal. The ceramic is cooled faster and a greater temperature droop occurs.

The undesirable temperature droop can be offset by incorporating a larger number of heat exchangers in the preheater system. The amount of droop in each heat exchanger may be the same, but due to the number of heat exchangers cycling at sequenced times, the overall temperature droop can be reduced. This effect is shown in Figure 3.

A system of heat exchangers for preheating air would consist of a quantity, NR, of regenerators in the reheat mode, and a quantity, NB, in the blowdown mode at any given time. There will be a quantity, NS, of heat exchangers which are neither in the reheat nor the blowdown mode at a given time, but are in a transition period between the two. During this period, valving, purging, or pressurization may be occurring in the heat exchangers. The total number of heat exchangers in operation is then $N_{tot} = NR + NB + NS$. The value of NS will generally be two regenerators, one having finished the reheat mode and preparing for blowdown, the other following blowdown, before reheating.

The heat exchangers will undergo cycle changes progressively with a time delay equal to the switchover time STI seperating cycle changes. The number of heat exchangers NR will reheat for a specific time RTI, and NB will be in the blowdown mode for a time BTI, while NS preheaters are in the switchover mode for a time STI.

As an illustrative example, assume there are 6 heat exchangers in reheat, 4 in blowdown and 2 in switchover modes. The ratio of reheat time to blowdown time must be the same ratio as NR/NB;

$$\frac{RTI}{BTI} = \frac{NR}{NB} = \frac{6}{4}$$

In this example, use RTI = 600 seconds and BTI = 400 seconds. In order to maintain 6 heat exchangers in the reheat mode and 4 in the blowdown mode it is necessary for there to be 6 switchovers during a reheat period or 4 during a blowdown period. Then;

$$STI = \frac{RTI}{NR} = \frac{BTI}{NB}$$

or in this example, STI = 600/6 = 100 seconds. An operational diagram for this system is shown in Figure 4. As the figure shows, for our example, there are 12 total operational heat exchangers, cycling at staggered times with a ratio of RTI/BTI = 6/4 and NR/NB = 6/4 at any time. At the time, $\underline{t}$ shown in the figure, heat exchangers 2,3,4,5 are in the blowdown phase, 7,8,9,10,11 and 12 are reheating while 1 and 6 are undergoing a switchover process. It is seen that the time lag between consecutive heat exchangers is equivalent to the switchover time, STI. To maintain flow continuity and insure equivalent conditions, the heat exchangers must be of identical size, with the same number of flow channels in each cored matrix. Equations for determining flow areas, number of flow channels and velocity relations for reheat and blowdown are found in Appendix I.

3.1.2 Design Constraints

The cored ceramic bricks comprising the heater matrix of a direct fired cored brick regenerative heat exchanger in an open cycle-coal burning MHD power plant will be subjected to the hot, corrosive, seed and slag laden exhaust gases emerging from the MHD generator channel. High temperatures and a temperature accelerated corrosive attack of the seed and slag combine to make ceramic material selection for the cored bricks a principal task when designing MHD air preheaters. This thesis does not devote itself to material choice, but rather to specifying operational parameters and determining heat exchanger sizing and flow criteria. Numerous studies have been made which test different candidate materials for application in a coal burning, MHD plant air preheater system [26,27,28,29,30].

The presence of coal slag from the combustor process and the addition of a seed material to the MHD stream to increase the gas ionization introduce the probability of seed and slag deposition on the air preheater flow hole walls. To date, there are no coal fired cored brick regenerative heat exchangers in operation, so information concerning buildup of matter on tube walls is primarily from small scale experimental results [19] and simulations [7]. There is general agreement that small diameter holes (near 20 mm) should be avoided when using a reheat gas laden with coal combustion products and seed, but a lower limit has not yet been determined and this thesis does not attempt to define a minimum hole diameter. However, the probability of plugging of holes or at least flow restriction by the adherance of seed or slag on the the tube walls is recognized and no extreme reductions in hole size are considered in the design process to reduce heat exchanger mass.

Nevertheless, the important design criteria of minimum possible ceramic mass is approached by using smaller diameter holes as is shown below. The major resistance to heat transfer is located at the convective surface, so by raising the convective surface area, a more effective heat exchanger results. The ratio of convective surface area to ceramic volume in the cored brick is inversely proportional to the hole diameter, or

$$\frac{A_{conv}}{Vol} = \left(\frac{1}{D}\right) \frac{1}{(S/D)^2 \cos(\pi/6) - 1/4}$$

From this equation it is seen that cored bricks with smaller diameter holes have a larger convective area to volume ratio and can transfer heat between the ceramic and the gas streams more effectively than large hole bricks, yet the plugging problem exists and is more severe with the smaller holes. Cored brick regenerators with large hole diameters can also provide high temperature preheated air, have a lower expectancy of plugging and exhibit less pressure drop along the flow stream. Although some pressure drop is acceptable, a minimum pressure drop has the advantage of reducing auxiliary compressor and pumping requirements.

The equation in the previous paragraph also shows the dependence of the convection surface area to volume ratio upon the hole spacing to diameter ratio (S/D). Reducing the S/D ratio decreases the ceramic volume surrounding each hole and thereby increases the A_{conv}/Vol ratio. Problems resulting from a reduction in S/D include an increase in the blowdown exit temperature droop over a specified cycle time, a decrease in the average exit temperature for a specific cycle time, and anticipated increased fabrication difficulties arising from the reduction in web thicknesses. Advantages of smaller S/D are a more effective heat-transfer mechanism, a smaller total ceramic-mass requirement, and smaller thermal stress levels for equal mass flow rates (this occurs as a result of smaller temperature differences between the convective surface and average ceramic temperatures).

Design trends then suggest that reductions in heat exchanger masses and a more effective mass use are obtained by decreases in hole diameter and/or decreases in the S/D ratio. The extent of these decreases is limited by the problems of slag buildup, pressure drops, fabrication difficulties and excessive blowdown exit temperature droop.