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Super Boiler 2nd Generation Technology for Watertube Boilers

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Abstract

This report describes Phase I of a proposed two phase project to develop and demonstrate an advanced industrial watertube boiler system with the capability of reaching 94% (HHV) fuel-to-steam efficiency and emissions below 2 ppmv NO_x, 2 ppmv CO, and 1 ppmv VOC on natural gas fuel. The boiler design would have the capability to produce >1500°F, >1500 psig superheated steam, burn multiple fuels, and will be 50% smaller/lighter than currently available watertube boilers of similar capacity. This project is built upon the successful Super Boiler project at GTI. In that project that employed a unique two-staged intercooled combustion system and an innovative heat recovery system to reduce NO_x to below 5 ppmv and demonstrated fuel-to-steam efficiency of 94% (HHV).

This project was carried out under the leadership of GTI with project partners Cleaver-Brooks, Inc., Nebraska Boiler, a Division of Cleaver-Brooks, and Media and Process Technology Inc., and project advisors Georgia Institute of Technology, Alstom Power Inc., Pacific Northwest National Laboratory and Oak Ridge National Laboratory. Phase I of efforts focused on developing 2nd generation boiler concepts and performance modeling; incorporating multi-fuel (natural gas and oil) capabilities; assessing heat recovery, heat transfer and steam superheating approaches; and developing the overall conceptual engineering boiler design.

Based on our analysis, the 2nd generation Industrial Watertube Boiler when developed and commercialized, could potentially save 265 trillion Btu and $1.6 billion in fuel costs across U.S. industry through increased efficiency. Its ultra-clean combustion could eliminate 57,000 tons of NO_x, 460,000 tons of CO, and 8.8 million tons of CO_2 annually from the atmosphere. Reduction in boiler size will bring cost-effective package boilers into a size range previously dominated by more expensive field-erected boilers, benefiting manufacturers and end users through lower capital costs.

Executive Summary

This report describes Phase I of a proposed two phase project to develop and demonstrate an advanced industrial watertube boiler system with the capability of reaching 94% (HHV) fuel-to-steam efficiency and emissions below 2 ppmv NO_x, 2 ppmv CO, and 1 ppmv VOC on natural gas fuel. The boiler design would have the capability to produce >1500°F, >1500 psig superheated steam, burn multiple fuels, and will be 50% smaller/lighter than currently available watertube boilers of similar capacity.

This project is built upon the successful Super Boiler project at GTI. In that project, a firetube boiler was developed employing a unique two-staged intercooled combustion system that brings NO_x emissions below 5 ppmv without using flue gas recirculation, high excess air, or other efficiency-robbing measures. The system also employs an innovative heat recovery system based on the Transport Membrane Condenser (TMC) which removes moisture from the flue gas with full recovery of its latent heat and has demonstrated fuel-to-steam efficiency of 94% (HHV).

The project was carried out under the leadership of GTI with project partners Cleaver-Brooks, Inc., Nebraska Boiler, a Division of Cleaver-Brooks, and Media and Process Technology Inc. (MPT), and project advisors Georgia Institute of Technology, Alstom Power Inc. (Alstom), Pacific Northwest National Laboratory (PNNL) and Oak Ridge National Laboratory (ORNL). Phase I of efforts focused on developing 2^{nd} generation boiler concepts and performance modeling; incorporating multi-fuel (natural gas and oil) capabilities; assessing heat recovery, heat transfer and steam superheating approaches; and developing the overall conceptual engineering boiler design. Phase II, if implemented, would demonstrate the design developed in Phase I. The key project accomplishments are described below.

GTI completed Aspen Plus modeling of several versions of boilers with two-stage combustion systems. The variations include boiler load (40,000 to 80,000 lbs/hr), pressure (150 to 1,500 psig), saturated and superheated steam, steam temperature (up to 1200 F), and different design specifics, especially the performance of the intercooling section between the stages. Detailed performance results for selected versions would be used in the engineering design of the two-stage industrial watertube (IWT) boiler.

The team identified several concepts for vaporizing liquid fuel using energy from hot flue gases. An alternative dual fuel concept employing partial vaporization and atomization of liquid fuel at high pressure and high temperature involving distributed oil flames, i.e. an oil nozzle located in the center of each of the gas nozzle/spargers, was subsequently tested with No. 2 oil in the GTI laboratory on the two-stage firetube burner/boiler and achieved NO_x of 17 to 23 ppmv at 3% O_2 at 3 MMBtu/hr with low CO levels of less than 100 ppmv and without any soot generation. Tests were also carried out at GTI with fully vaporized liquid fuel (No. 2 oil) in the two-stage laboratory firetube Super Boiler at firing rates of up to 3 MMBtu/hr (22 gal/hr). NO_x measured in the primary zone was 4 ppmv, while NO_x in the stack varied from 20 to 23 ppmv at 3% O_2. An analysis of the No. 2 liquid oil fuel indicated a nitrogen content of 159 ppmw which is equivalent to 21 ppmv of NO_x in the flue gas if all bound nitrogen is converted to NO_x at 3% O_2. This indicates that practically no thermal NO_x was produced in the two-stage combustion process at this condition. Using the same approach, 100% biodiesel liquid fuel was vaporized and combusted in the laboratory two-stage burner/boiler with NO_x emissions of 20 to 25 ppmv at 3% O_2.

ORNL completed a study of high temperature materials for potential application in the high temperature/high pressure superheater. Test coupon samples of three high temperature alloy materials (Inoconel 617, Haynes 230 and Inconel 740) selected in conjunction with Alstom as potential materials for the superheater were subsequently prepared..

Nebraska Boiler initiated investigation of the modifications to their conventional watertube boiler design platform to incorporate the two-stage combustion system with inter-stage cooling. Two concepts for introducing secondary combustion air into an existing watertube boiler design were identified. Initial efforts focused on assessing application of two-stage combustion to a 40,000 lbs/hr steam capacity boiler while maintaining the physical dimensions comparable to their 20,000 lbs/hr conventional D type watertube boiler.

Based on our analysis, the 2nd generation IWT boiler when developed and commercialized, could potentially save 265 trillion Btu and $1.6 billion in fuel costs across U.S. industry through increased efficiency. Its ultra-clean combustion could eliminate 57,000 tons of NO_x, 460,000 tons of CO, and 8.8 million tons of CO_2 annually from the atmosphere. Reduction in boiler size will bring cost-effective package boilers into a size range previously dominated by more expensive field-erected boilers, benefiting manufacturers and end users through lower capital costs.

Introduction and Background

Gas Technology Institute (GTI), in partnership with Aqua-Chem Incorporated, a major industrial boiler manufacturer, developed a new boiler technology in mid-2000, based on firetube configuration that is capable of meeting DOE's goals for enhanced industrial boiler technology, including 94% fuel-to-steam efficiency, drastically reduced emission levels, and reduced boiler weight and footprint. The project described in this document has taken this Super Boiler technology, along with several new innovations, and developed designs for applying it to large multi-fuel watertube boilers capable of producing high-temperature and high-pressure superheated steam with 2 ppmv NO_x and CO, 1 ppmv VOC, and 50% size reduction.

Most of the installed industrial steam boilers in the U.S. are more than 25 years old and were built with pre-WWII technology. As a result, they tend to be large, relatively inefficient, with about 75% thermal efficiency on average, and are difficult to retrofit for compliance with current emissions regulations. These shortcomings were highlighted in the 1999 *Industrial Combustion Technology Roadmap*. In 2000, GTI began a project to apply innovative combustion, heat transfer, and heat recovery technologies to industrial steam generation as a first step towards developing the "Super Boiler" envisioned in the *Roadmap*. The technical goals of this project included 94% fuel-to-steam efficiency[1], NO_x and CO emissions[2] below 5 ppmv, VOC emissions below 1 ppmv, and substantial reduction in equipment footprint and weight when fired with natural gas. GTI and its industrial partner, Aqua-Chem, developed a natural gas fired firetube boiler system—the 1st generation Super Boiler— capable of meeting these goals. The system incorporates several core concepts which have been implemented in the current effort, along with new ideas, for larger, higher-pressure, higher-temperature watertube boilers capable of natural gas as well as liquid fuel firing.

Each of the following subsections explain how specific performance elements (thermal efficiency, emissions, ability to run on multiple fuel types, ability to produce high-pressure superheated steam, and reduced boiler size) were addressed, the modifications or new approaches that have been formulated for larger high-performance boilers, and initial plans to implement the selected concept(s) in the 2nd generation IWT Super Boiler design.

Thermal Efficiency

The thermal efficiency target for this boiler is 94%, same as for the earlier firetube Super Boiler. In the firetube Super Boiler, GTI addressed this goal by maximizing heat recovery from the flue gas. In particular, recovery of latent heat which, in a gas-fired system, accounts for about 66% of the total heat lost through flue gases was chosen as the best avenue for efficiency improvement. Cooling of flue gas to condensing conditions is already practiced on small hot water boilers but is not widely used in industrial steam boilers because of large surface area requirements and concerns about stack corrosion.

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1 Higher heating value (HHV) basis.
2 All emissions are corrected to 3% oxygen unless otherwise noted.

To overcome these barriers, GTI invented the TMC[3]. This device, shown in Figure 1, uses microporous membrane elements to selectively extract water vapor from flue gas by a combination of surface diffusion and capillary condensation. The extracted water vapor, after passing through the membrane, condenses in direct contact with boiler feed water, allowing full recovery of its latent heat while keeping flue gas humidity below 100%. In addition, all of the water removed from the flue gas enters the boiler feed water supply, an advantage in locations where water supply is limited. The membrane properties exclude contaminants such as CO_2, NO_x, and particulates. Table 1 shows TMC test data at 55% and 100% boiler load. The TMC performed in accordance with the specifications, cooling and dehumidifying the flue gas to a 41-46% moisture removal level, with a corresponding boiler efficiency of 93.5-94.1%.

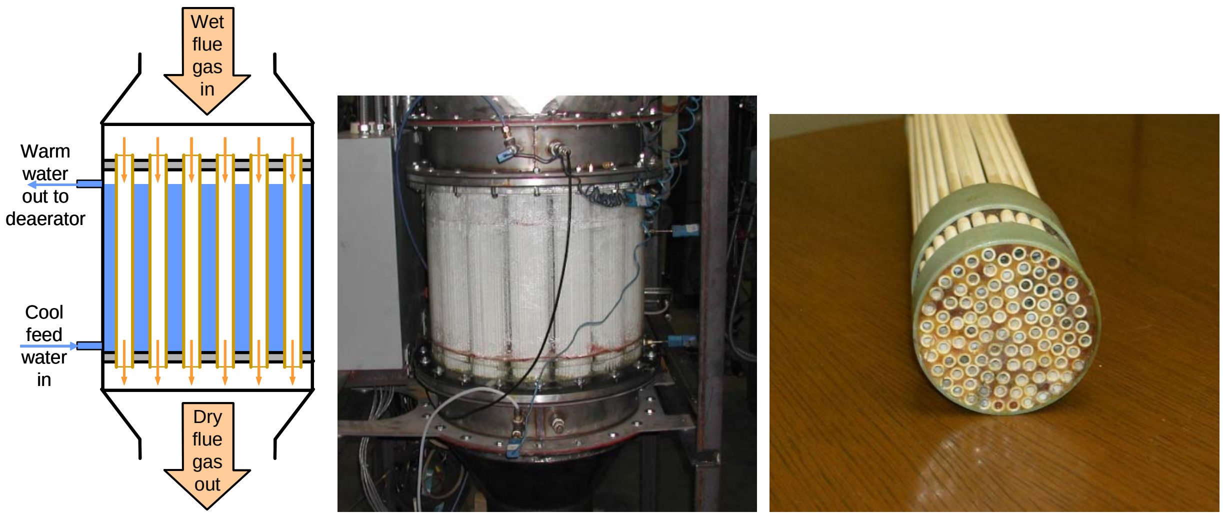

Figure 1. TMC Simplified Design, 3 MMBtu/h Prototype Unit, and Membrane Tube Bundle

Table 1. TMC Performance Data

Firing rate, MMBtu/h	1.6	3.0
Stack O_2, vol%	3.1	3.1
Flue gas temperature/dew point		
TMC inlet, °F	152/132	171/132
TMC outlet, °F	112/110	119/113
Water temperature		
TMC inlet, °F	68	68
TMC outlet, °F	131	132
Water vapor removed from flue gas, %	45.5	41.1
Calculated boiler efficiency, %	94.1	93.5

A complete heat recovery system, shown in Figure 2, was designed around the TMC to maximize its effectiveness.

3 U.S. Patent No. 6,517,607: "Method and Apparatus for Selective Removal of a Condensable Component from a Process Stream with Latent Heat Recovery" (11 Feb 2003).

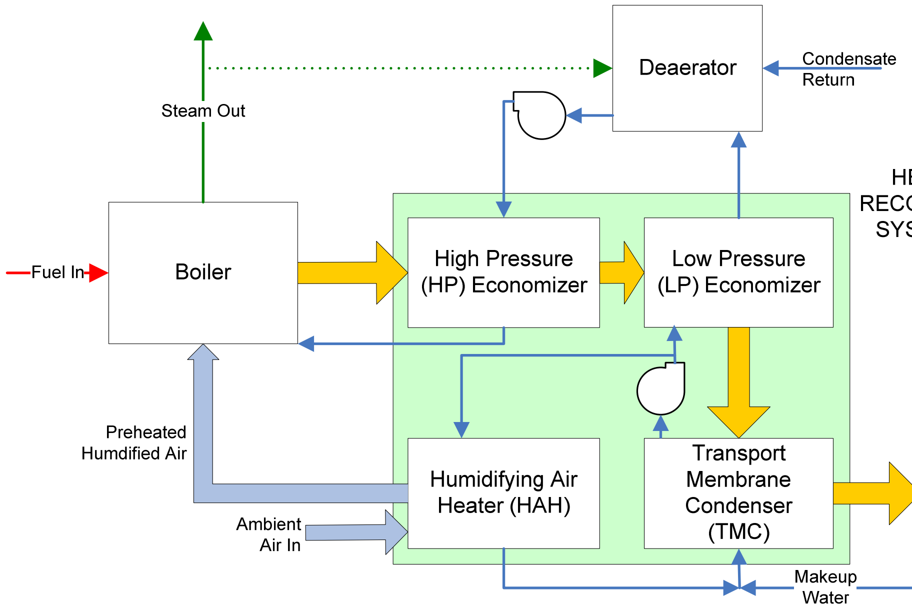

Figure 2. Super Boiler Heat Recovery System Schematic

High Pressure (HP) and Low Pressure (LP) economizers are used to condition the flue gas for optimal TMC performance. A Humidifying Air Heater (HAH), shown in Figure 3, is another key component, particularly for installations where substantial condensate return replaces cold makeup water. The HAH, like the TMC, uses microporous ceramic membrane tubes to transport water between streams, but in this case, liquid water is transported through the membrane tube and up to 5% is evaporated into the combustion air stream, warming and humidifying the air while cooling the exiting water stream. The cooled water exiting the HAH is then recycled back to the TMC to help remove heat from the extracted flue gas water vapor. Prototype tests at GTI have confirmed the ability of the HAH to perform properly with condensate return cases from zero to 75% of boiler feed water demand.

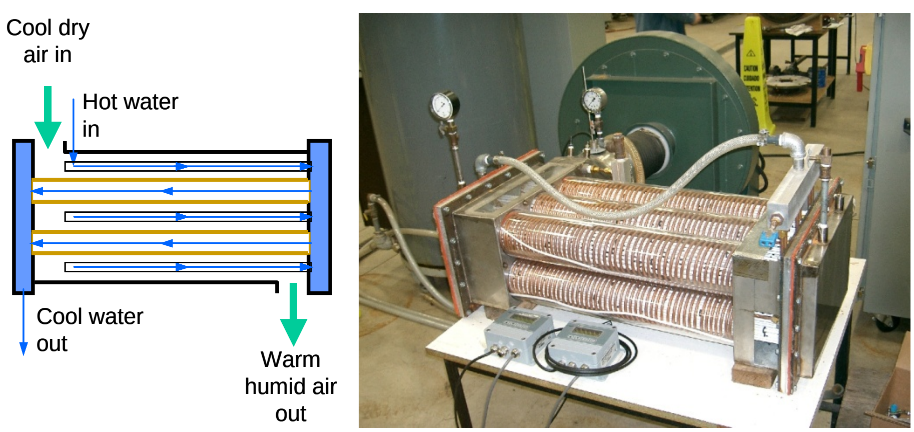

Figure 3. HAH Simplified Design and 3 MMBtu/h Prototype Unit

For the system described above and pictured in Figure 2, when properly tuned and controlled, the potential energy efficiency for a self-contained boiler system ranges from 90 to 95%, depending on condensate return, ambient air conditions, and makeup water temperature.

In the current project, the same general approach has been applied to larger high-pressure watertube boilers. The emphasis has been on scale-up and optimization with regard to equipment cost. The team also examined the option of applying a single TMC-based heat recovery system for multiple boilers in an industrial boiler house to save capital cost. Field testing results from GTI's earlier firetube Super Boiler project provided valuable performance data to support design improvement and optimization. Modifications that were considered included enhanced water distribution in the TMC vessel, improved membrane bundle designs, and more cost-effective membrane manufacturing procedures.

A two-stage TMC concept, where the first stage is located in the 500-700°F flue gas temperature range and the second stage in the 150-250°F region following the economizers was also considered.

Steam superheating has an impact on heat recovery with the TMC/HAH system. The effectiveness of the TMC for cooling and dehumidifying flue gas depends on the temperature and volumetric flow of water that passes through the TMC. This in turn is linked to the boiler feed water rate, which is lower for superheated steam than for saturated steam at the same heat input. The problem can be alleviated by increasing the recycle rate to the HAH to provide heat removal for water supplying the TMC, and 94% efficiency can be achieved in this way. However, this would require larger vessels, piping, and valves to accommodate the increased flow, plus a larger TMC circulation pump. For this reason, an additional component, the Flash Evaporation Cooler (FEC) was considered. The FEC uses evaporative cooling of hot water from the TMC outlet under vacuum conditions where a portion of the water flashes into vapor, reducing the temperature of the remaining liquid water. The vacuum can be provided by pump or a steam ejector, in which case the exit stream will be routed to the deaerator. Preliminary calculations indicated that 5% evaporation of the TMC discharge water stream at a 29-inch Hg vacuum would be adequate to achieve 94% boiler efficiency. The feasibility of this concept and the preferred way to integrate it with the existing heat recovery system was assessed.

An option was considered to help limit the size of the heat recovery equipment and its water recycle requirement - Direct-Fired Superheater (DFSH). This is a method of decoupling superheat from heat recovery, and is discussed in more detail in the section on High-Pressure Superheated Steam Capability.

NO_x, CO, and VOC Emissions

The emissions targets for the IWT boiler were NO_x emissions below 2 ppmv, CO emissions below 2 ppmv, and VOC emissions below 1 ppmv. In the previous firetube Super Boiler project, NO_x reduction to levels below 5 ppmv was achieved using staged combustion with two major innovations: (1) engineered internal recirculation and (2) combustion in two separate furnaces with intensive interstage cooling. Previous R&D at GTI, including field tests up to 60 MMBtu/h, had proven the effectiveness of staged combustion combined with Forced Internal Recirculation (FIR). In this type of burner, a recirculation insert, or sleeve, is mounted in the combustion chamber concentrically with an array of flame nozzles to recirculate products of partial combustion back to the flame root after giving up a portion of their heat by conduction to the recirculation sleeve. The recirculation sleeve then radiates to the cold water walls of the boiler, reducing peak flame temperatures. This approach eliminates the need for energy-robbing measures such as external Flue Gas Recirculation (FGR), high excess air firing, steam injection, or water injection to reduce NO_x. In the air-staged version, the first stage is fired at sub-stoichiometric conditions, yielding a CO and H_2-rich fuel gas mixture. Combustion is then completed in the secondary zone by injection of the remaining air. However, the effectiveness of NO_x reduction in the FIR burner at the time had been limited to about 7 ppmv, in part because of a lack of control over secondary zone combustion temperatures. In the 1st Generation Super Boiler project, researchers were able to address this problem by closer integration of the burner and boiler design, as shown in Figure 4. Burner, combustion chamber, and pressure vessel design are all integrated for optimal emissions, compactness, safety, and ease of operation.

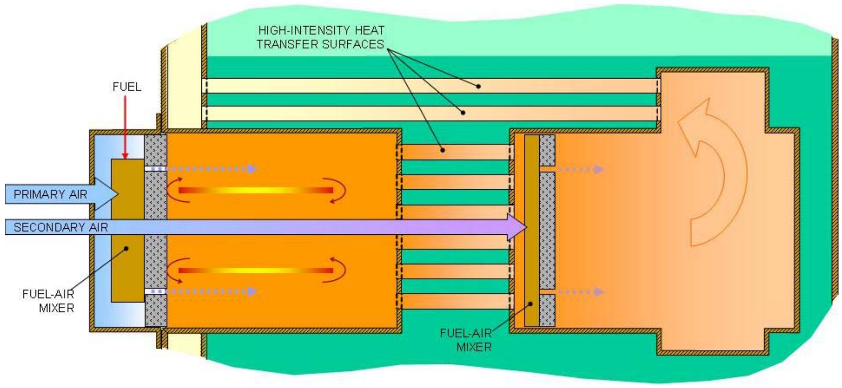

Figure 4. Simplified Diagram of Two-Stage Combustion System and Convective Pass

The first stage burner used an advanced nozzle design to obtain very uniform fuel-air mixing at the injection point. Engineered internal recirculation was applied as an improvement of the FIR approach using an optimized recirculation sleeve to simultaneously cool and stabilize the fuel-rich first stage flame, extending the range of stoichiometry over which the first stage can be fired. The interstage cooling pass reduces the temperature to allow premixing of the first-stage partially combusted fuel with secondary air, and significantly reduces the flame temperature of the secondary flame. Figure 5 shows an 80 HP laboratory boiler that was built by Cleaver-Brooks in January 2004 and tested over the last year at GTI, along with sample operating data that show NO_x emissions well below 5 ppmv with good burnout at only 9 to 17% excess air.

Firing rate, MMBtu/h	0.85 (light-off)	1.70	2.52	3.00	3.65
Staging	1	2	2	2	2
Steam pres, psig	atm	101	116	118	118
O_2, vol%	7.2	2.0	3.4	3.0	2.1
NOx, ppmv	8.2	2.2	3.3	2.6	3.0
CO, ppmv	14	6	5	7	7
THC, ppmv	2	0	7	0	0
Flue gas temp, °F	200	314	346	354	367
Furnace 1 temp, °F	1385	1222	N/a	1853	1941
Furnace 2 temp, °F	702	1912	N/a	2049	2146

Figure 5. 80 HP Laboratory Super Boiler and Test Data

Adaptation of this combustion system from a cylindrical firetube combustion chamber to a rectangular watertube furnace required both Computational Fluid Dynamics (CFD) modeling and laboratory evaluation. Additionally, reducing potential NO_x from the current levels (2.2-3.3 ppmv) to <2 ppmv required optimization of the combustion system. Approaches that the team considered included fuel preheating, to permit deeper staging (lower first-stage stoichiometry), and tuning the boiler wall heat extraction profile to further reduce peak flame temperatures through judicious layout of the water wall tubes. Flame stability, uniformity, recirculation profile, etc. were studied (CFD modeling and selective experimentation) to find the required solutions for watertube boilers.

As shown in Figure 5, CO emissions for the firetube version in staged mode were below 10 ppmv even though the secondary burner operated at low excess air (1 to 3% O_2) and the first-stage fuel gas contains up to 8% CO. This indicates excellent burnout with a very short flame, and potential for CO to be further reduced by minor improvements in the mixing nozzle design. The same approach applies to achieving <1 ppmv VOC.

Multiple Fuel Capability

The 2nd generation IWT boiler was targeted to be capable of operating on multiple fuels. Across the U.S., with the exception of California, most industrial boiler owners require backup fuel capability, with No. 2 fuel oil being most commonly specified. Although emissions regulations are generally relaxed for backup fuel, the team believed the market was very receptive to a combustion technology that can reduce NO_x for oil-fired systems. Initial development of oil firing capability with significantly reduced NO_x was investigated on the two-stage laboratory firetube Super Boiler, based on the concept of oil pre-vaporization. This general approach to oil combustion had been specifically called out in the 2002 revision of the *Roadmap* as a recommended avenue for boiler burner research.[4] Combustion of oil in vapor form, compared to the current practice of atomization, was believed to reduce the peak flame temperatures that produce thermal NO_x. This approach was also believed to improve burnout, reducing CO, VOC, and soot. The methods for oil vaporization assessed included direct heating with partial combustion gases and indirect heating with steam or combustion gases. Burner design for on-line fuel switching was also considered as an option in this effort.

It should be noted that in addition to liquid fuels, staged combustion with interstage cooling is amenable to firing gaseous fuels other than natural gas. These include Liquefied Natural Gas (LNG), syngas from gasifiers, coke oven gas, refinery gases, landfill gas, and a variety of other process-dependent industrial waste gases. Burner design to distribute fuel gases between first and second stages of combustion, depending on the heating value and ignition properties, is believed feasible and can be applied to large watertube boilers.

High-Pressure Superheated Steam Capability

The boiler system was targeted to be capable of producing high pressure/high temperature steam (greater than 1500°F/1500 psig). Firetube boilers cannot meet this condition because the highest pressure that is economically viable for a firetube design is about 600 psig. However, watertube

4 *2002 Industrial Combustion Technology Roadmap*, under "Priority Burner Research Needs, Advanced Combustion Stabilization Methods", p. 18.

boilers are capable of pressures up to 3000 psig or higher, depending on the application and its economics. The main industrial use for high-pressure steam (greater than 1000 psig) is power cogeneration via steam turbine, so economic evaluation was based on this application. Watertube boilers fall broadly into three design categories that have important implications for steam pressure and temperature: 1) natural circulation, 2) forced circulation, and 3) once-through boilers. Natural circulation systems are available up to 1800 psig and 1000°F and forced-circulation or once-through designs are capable of even higher pressures.

While natural circulation systems are self-regulating in terms of required circulation ratio for proper heat flux, forced-circulation boilers use a pump to circulate hot water between the drums. This allows more flexibility in the geometric design of the boiler because circulation is less dependent on the tube arrangement. Forced circulation boilers also typically have a lower water inventory and thus respond to steam pressure changes faster, but operational stability is more difficult to ensure. Once-through boilers are "drum-less" continuous tube heat exchangers in which preheating, evaporating, and superheating of the feed water take place sequentially. With its even smaller water inventory, the once-through boiler is suitable for high steam pressures, and has a thermal cycling advantage with fast startup from cold conditions. However, reliability issues require even more attention than with forced circulation boilers.

Application of two-stage combustion with interstage cooling to each of the three major boiler design types was considered. In a conventional industrial watertube boiler, the superheater is located at the end of the combustion box in the turning section, just before the convective section, but the staged IWT boiler approach offered unique advantages. In the substoichiometric primary zone, the high heat capacity of the partially combusted fuel gases promoted higher heat flux, and the reducing atmosphere was believed to be less destructive to the superheater fabrication materials. Partial superheat can be obtained in the secondary zone combustion chamber, where flue gas temperature during startup (single-stage firing) is less than 1000°F, but rise to 1700-1800°F after staged firing is established. The superheater can thus potentially be of a staged design, including integration with the intercooling section, which can comprise an array of steam generation and superheating tubes to regulate the temperature of primary zone gas to the second stage burner. Stage 2 superheating may also be integrated with the recirculation sleeve.

Identification and testing of suitable high-temperature materials was believed essential in order to implement steam superheating to >1500°F via indirect heat transfer. Superheater tubes in a boiler experience the most severe service conditions of any boiler component, and the tube material must satisfy demanding requirements with respect to fireside corrosion, steam side oxidation, creep rupture strength and fabricability. Depending on the thermal conductivity of the alloy, superheater tubes must be designed to operate at temperatures at least 60°F above the actual steam temperature

Another option considered, as discussed was DFSH, in which high-pressure steam that is partially superheated by conventional means is raised to the desired final superheat temperature by direct contact with a natural gas-oxygen flame. The high-pressure DFSH burner chamber can be steam-cooled, thereby reducing its material requirements. A small amount (less than 4% by volume) of CO_2 enters the steam supply, but for many end uses this may be acceptable.

Currently, the only major end use of high-pressure, high-temperature steam is to drive a steam turbine, and the low levels of CO_2 created by a DFSH boost would not be a problem for the turbine, but may be a matter of concern with the outlet steam from the turbine for use by a downstream application. The matter of steam impurities was a concern and should be carefully evaluated. The DFSH approach has a significant advantage for energy efficiency as mentioned earlier, because virtually 100% of the added fuel energy is converted directly into steam enthalpy with no impact on the TMC/HAH-based heat recovery system. A preliminary analysis showed that for a boiler producing 1500-psig 1000°F conventionally superheated steam that is increased to 1500°F by DFSH, 16% of the fuel is fired in the DFSH and the thermal efficiency increases from 94.0 to 95.0%. Another potential advantage is reducing the pressure required across the superheater to get to the final steam pressure. The major questions considered were (1) energy costs of O_2 supply; (2) energy cost of oxidant and fuel compression; and (3) methods to minimize contaminants such as excess O_2 and unburned hydrocarbons in the steam product.

Reduced System Weight and Footprint

Based on the earlier work on firetube Super Boiler, the targeted system weight and footprint were 50% of currently available boilers with comparable performance. This was achieved on the firetube boiler through a convective pass system that uses a proprietary internal extended-surface design to intensify heat transfer up to 18 times compared to bare convective tubes. This design allowed the boiler length to be dictated not by the length of the convective tubes required to extract heat from the flue gases, but by the furnace length required to complete combustion and moderate the furnace exit temperatures. This is important because with the two-stage intercooled furnace design, the total combustion chamber length is less than with the conventional single burner and single furnace. With the advanced convective tube technology, stack temperature was only 10-20°F above the saturated steam temperature, resulting in a two-pass boiler that extracts heat as effectively as a much larger four-pass boiler. The approximate size reduction obtained from compact design and the use of enhanced heat transfer convective tubes is illustrated in Table 2, which compares the specifications for two commercially available 100-horsepower four-pass firetube boilers operating at similar steam output as the staged intercooled boiler pictured in Figure 5.

Table 2. Size Comparison of Conventional Boilers and Super Boiler

Description	Four-pass firetube boiler A	Four-pass firetube boiler B	Two-Stage Super Boiler
Overall footprint, ft^2	79.0	92.2	54.7
Dry weight, ton	6.4	6.2	3.8
Overall footprint, ft^2	79.0	92.2	54.7
Dry weight, ton	6.4	6.2	3.8
Shell length,-inch	118	109	94
Shell diameter,-inch	60	62	48
Overall length,-inch	144	168	113
Overall width,-inch	79	79	70
Stack height,-inch	104	84	70

A similar approach was considered for reducing size for watertube boilers because watertube combustion chambers have typically been even more oversized than firetubes. This size reduction for watertubes would allow not only reduced footprint, weight, and materials cost, but also the potential to design transportable shop-fabricated watertube boilers that can compete with field-erected boilers at the larger sizes (~125,000-200,000 lb/h steam). Typically, the cost of a field-erected boiler is about double the cost of a packaged boiler of similar capacity. Furthermore, with the staged intercooled design, including integration with superheating, a modular approach may be applicable to even larger boiler sizes (up to 300,000 lb/h steam). The cost savings to industrial users would be considerable, making replacement of aging field-erected boilers more attractive.

Another step in size reduction considered was the implementation of very compact economizers based on microchannel technology from PNNL. GTI and PNNL had performed extensive lab testing resulting in a design with heat transfer 65 times greater than conventional finned-tube economizer on a volume basis. Consequently, the microchannel economizer is easily integrated with the TMC.

Technical Approach

The technical approach for achieving the targeted specifications consisted of the following main elements:

- A high-pressure (>1500 psig capability) watertube boiler using air-staged combustion with intensive interstage cooling to simultaneously reduce emissions and reduce overall boiler size requirement. Options considered were natural circulation, forced circulation, and once-through designs for optimum technoeconomic value. Increased convective pass heat transfer via finned and/or dimpled tube configurations was also considered for integration into the design to minimize boiler dimensions, extract maximum heat, and thus reduce the size requirements of the downstream heat recovery system;
- Enhancements to two-stage intercooled combustion including primary zone flame optimization to bring NOx emissions down from current 2-4 ppmv to less than 2 ppmv: mechanical improvements to second stage burner were considered for implementation to bring CO below 2 ppmv and VOC below 1 ppmv;
- Steam superheating using advanced high-temperature alloys and a two-stage approach integrated with the two-stage intercooled combustion system for optimum thermal management: a DFSH concept was considered as an option for >1500°F steam temperature, including evaluation of industrial steam utilization with up to 4% CO2;
- Heat recovery from flue gases based on the TMC, HAH, and dual economizers, engineered and scaled up for the required high-pressure watertube boilers: a new concept, FEC, was considered and evaluated for optimizing the latent heat removal capacity of the TMC; optimizing microchannel economizer was considered for cost-effectiveness; solid-state power generation modules were considered as an option for "self-powered" boiler capability;

A simplified schematic of one version of the 2nd generation Super Boiler system is shown in Figure 6. This version includes two-stage indirect superheating and TMC/HAH heat recovery. Other versions considered include DFSH and/or FEC.

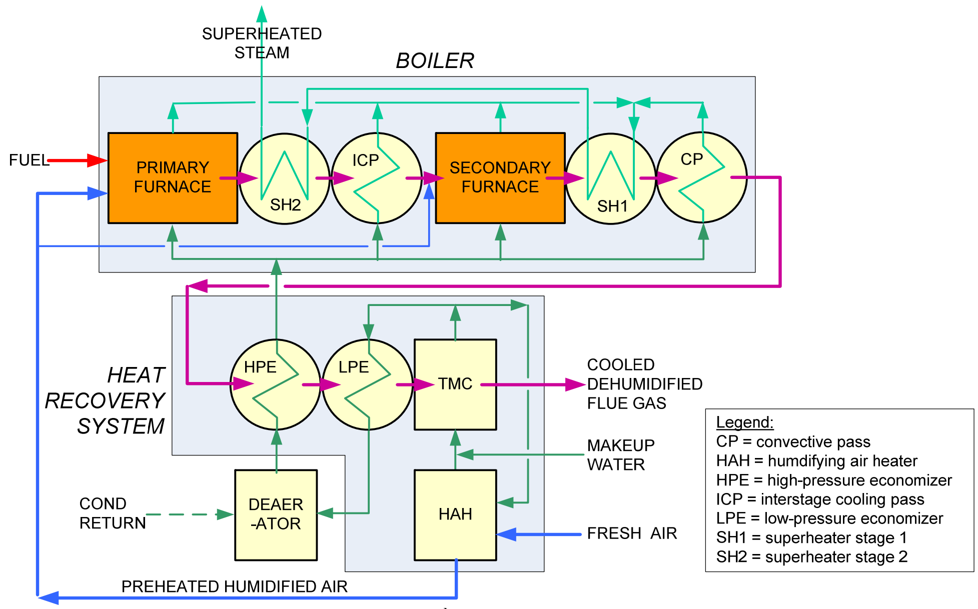

Figure 6. General Schematic of 2nd Generation Super Boiler System (Version 1)

The main objectives of the effort reported here were to develop, through modeling and experimental studies, concepts and conceptual designs for an industrial watertube boiler system that would have the technical capability of meeting the targeted requirements discussed above. This was accomplished through concept development and modeling, assessing key system elements consisting of heat recovery, heat transfer and superheating, and developing preliminary boiler design schemes.

Concepts and Modeling

The objective of this task was to develop and evaluate boiler design concepts for the 2nd Generation IWT boiler. The developed concepts include two-staged intercooled combustion for natural circulation, forced circulation, and/or once-through watertube boilers capable of producing steam at 1500 psig or higher. Engineering specifications for firing natural gas, LNG, propane, syngas, and fuel oil and new heat recovery concepts were assessed.

Laboratory Boiler Validation

The objective of this task was to conduct lab testing of IWT boiler design concepts specifically to establish a design basis for two-stage intercooled combustion in a watertube boiler platform. Approaches that were evaluated included preheating of fuel and optimized boiler wall heat extraction profile through CFD modeling. Integration of superheating with interstage cooling was investigated. Firing of backup fuel oil (No. 2 oil) in staged intercooled mode was investigated in the boiler simulator and the design elements transferred to the boiler design.

Heat Recovery

The objectives of this task were to incorporate improvements and manufacturing cost reduction procedures for the IWT boiler heat recovery system based on TMC and HAH that had previously been developed by GTI; evaluate two-stage TMC concept with high- and low-temperature stages; and (5) develop criteria to select the most appropriate design for watertube boilers. Approaches for TMC and HAH cost reduction examined included variant membrane materials, optimized membrane bundle manufacturing methods, and improved vessel design. The use of a single heat recovery system for multiple boilers in industrial settings was assessed.

Heat Transfer

The objective of this task was to evaluate methods of increasing heat transfer in radiative and convective sections of the IWT boiler with the ultimate goal of footprint and weight reduction. Available technologies were investigated for adding extended surfaces to convective pass tubes and incorporating such technologies into the boiler design concepts. Optimization of microchannel-based economizer design, focusing on cost reduction, was carried out by PNNL.

Superheating

The objective of this task was to develop and evaluate two main approaches for providing very high temperature superheated steam: (1) indirect superheating using high-temperature materials, and specific integration with the staged intercooled boiler design, and (2) DFSH with natural gas and oxygen. Superheating designs capable of >1500°F using indirect heat transfer from the fireside to the steam side in both primary and secondary combustion zones were conceptualized, including extended external (spiral fins, dimpled tubes, etc.) and internal (rifled, dimpled) tube surfaces. Materials for superheater tubes at this temperature and pressures above 1500 psig were identified. GTI coordinated this task with Alstom supporting superheater material specifications and superheater design, ORNL focusing on materials and Nebraska Boiler assisting in superheater integration with the boiler.

Design Evaluation, Selection, and Scale-Up

The objective of this task was to evaluate design elements studied in this effort and recommend a single overall design. The evaluation efforts was initiated with Nebraska Boiler based on projected ability to meet the performance targets, current regulatory framework at that time, and predicted market acceptance based on analysis of current markets. These include preliminary design development, scaled up, and review by the team members for a range of boiler sizes, steam pressures, and steam temperatures.

Description of Work Performed

Significant progress was made towards developing designs for the IWT boiler that can potentially meet the target specifications discussed earlier.

GTI completed Aspen Plus modeling of several versions of boilers with two-stage combustion systems. The variations include boiler load (40,000 to 80,000 lbs/hr), pressure (150 to 1,500 psig), saturated and superheated steam, steam temperature (up to 1200°F), and different design specifics, especially the performance of the intercooling section between the stages. Detailed performance results for selected versions would be used in the engineering design of the two-stage IWT boiler.

A concept for two-stage intercooled combustion for liquid fuel firing was developed, an existing burner and boiler were modified and testing was performed for liquid fuels using the two-stage laboratory firetube boiler at GTI. The team identified several concepts for vaporizing liquid fuel using energy from hot flue gases to allow firing of fully vaporized fuel. Tests were also carried out at GTI with fully vaporized No. 2 oil in the two-stage laboratory firetube boiler at firing rates of up to 3 MMBtu/hr (22 gal/hr oil). NO_x measured in the primary zone was 4 ppmv, while NO_x in the stack varied from 20 to 23 ppmv at 3 % O_2. An analysis of the No. 2 oil fuel indicated a nitrogen content of 159 ppmw which is equivalent to 21 ppmv of NO_x in the flue gas if all bound nitrogen is converted to NO_x at 3 % O_2. This indicates that practically no thermal NO_x was produced in the two-stage combustion process at this condition. Using the same approach, 100% biodiesel liquid fuel was vaporized and combusted in the laboratory two-stage burner/boiler with NO_x emissions of 20 to 25 ppmv at 3% O_2.

An alternative dual fuel concept employing partial vaporization and atomization of liquid fuel at high pressure and high temperature involving distributed oil flames, i.e. an oil nozzle located in the center of each of the gas nozzle/spargers was subsequently tested with No. 2 oil. Thgis approach achieved NO_x of 17 to 23 ppmv at 3% O_2 at 3 MMBtu/hr with low CO levels of less than 100 ppmv and without any soot generation.

ORNL completed a study of high temperature materials for potential application in the high temperature/high pressure superheater. Test coupon samples of three high temperature alloy materials (Inoconel 617, Haynes 230 and Inconel 740) selected in conjunction with Alstom as potential materials for the superheater were subsequently prepared..

Nebraska Boiler initiated investigation of the modifications to their conventional watertube boiler design platform to incorporate the two-stage combustion system with inter-stage cooling. Two concepts for introducing secondary combustion air into an existing watertube boiler design were identified. Initial efforts focused on assessing application of two-stage combustion to a 40,000 lbs/hr steam capacity boiler while maintaining the physical dimensions comparable to their 20,000 lbs/hr conventional D type watertube boiler.

Key Events and Accomplishments

The following is a summary of key events and accomplishments:

2006 Q3-4: DOE funding was delayed due to Congressional failure to pass a 2007 budget. Subcontracts and co-funding contracts with other sponsors were consequently also delayed.

2007 Q1: Discussions took place with DOE about restarting work and releasing some funds in 2007. Nevertheless, subcontracts and co-funding contracts with other sponsors were put on hold.

2007 Q2: DOE funding resumed in May 2007.

2007 Q3-4: Restarted efforts in developing IWT boiler and two-stage dual fuel burner concepts. Initiated development of revised plan for Phase I and Phase II based on long delay in the program restart. Work was also initiated at PNNL and ORNL to support IWT boiler development.

2008 Q1: Restarted efforts in application of Heat Recovery System (HRS) with TMC2. Initiated development of revised plan for Phase I and Phase II based on long delay in the program restart.

2008 Q2: Continued efforts in developing IWT boiler concepts, application of HRS with TMC2, dual fuel two-stage burner concept development and on revising Phase I and Phase II plans.

2008 Q3- 4: Continued efforts in developing IWT boiler concepts, application of HRS with TMC2, dual fuel two-stage burner concept development and on revising Phase I and Phase II plans. Hot oil testing of a flash liquid evaporation concept was conducted with a specially designed oil atomizing nozzle incorporated into the natural gas sparger of the two-stage burner design. Variations of the atomizer design to give the narrowest atomizing oil jet with the smallest oil droplet size were tested with positive results. A full scale test (40 MMBtu/hr input) of the two-stage combustion system in the D type (CB WT Model D-34) industrial watertube boiler (20,000 lbs/hr output) in the laboratory and then the deployment of the combustion system in the field was assessed for inclusion in the revised project plan. There are 20 to 40 D type watertube boilers sold per year in the range of 23 to 40 MMBtu/hr and a number of these are sold in the state of California where the demonstration site was planned. Also, a number of this size range of watertube boilers were identified in California as part of the existing base of 6000 watertube boilers, some of which are in the growing food industry applications.

2009 Q1: Continued efforts in developing IWT boiler concepts, application of HRS with TMC2, dual fuel two-stage burner concept development and on revising Phase I and Phase II plans. The first commercial prototype atomizer/vaporizer unit from Spraying Systems, for integrating oil atomizer/vaporizer into existing natural gas sparger design for the primary stage of two-stage super boiler, was successfully hot tested a single nozzle setup. The fabrication drawings for the modification of the dual fuel burner design for the laboratory two-stage firetube boiler, to include the new integrated design of oil/natural gas sparger, were completed. As a starting point for a compact two-stage watertube boiler, Nebraska Boiler provided a quote per GTI request for a 40,000 lbs/hr D type 250 psig design (150 psig operating) boiler (without burner), of membrane wall construction with some rows of external finned tubes in the convective section. The furnace cross section of the boiler is the same as Nebraska Boiler's 20,000 lbs/hr boiler and just 2 ft longer. The revised plans for Phase I and Phase II for the project were completed and submitted to DOE for consideration for funding. In addition to continued testing of the oil atomization/vaporization development for low NO_x dual fuel burners, and the development of the two-stage watertube boiler design, and testing of a 36 MMBtu/hr two-stage burner in the laboratory Watertube Test Boiler, the revised plan included development and testing of 34-inch long TMC modules for larger firetube and watertube boiler applications.

2009 Q3-4: Efforts were focused on installation and testing of the modified laboratory dual fuel burner on No. 2 oil with the prototype atomizer vaporizer nozzles in the laboratory two-stage firetube boiler. The 9 module TMC HRS, including integrated LPE, internal by-pass damper and microchannel HPE, was tested using flue gases from GTI's watertube test boiler. The development of the two-stage watertube boiler continued with extensive discussions with Nebraska Boiler. A proposal was requested from Nebraska Boiler for the preliminary engineering for the two-stage IWT boiler concept for either the D type or O type industrial watertube boilers. Request was made to DOE for extension of Phase I development through March 31, 2010 to allow for Nebraska Boiler to prepare preliminary engineering drawings for the two-stage IWT boiler concept and GTI to complete the report for Phase I.

2010 Q1-4: Discussions where held with both Nebraska Boiler and Siemens in regard to a high pressure/high temperature two-stage boiler and backpressure steam turbine for a high efficiency steam driven CHP system, which is one of ideal applications for the IWT boilers. Siemens provided several proposals for available backpressure turbines. One backpressure steam turbine for 80,000 lbs/hr 150 psig steam output and about 5 Megawatt electricity production was selected. Nebraska Boiler provided a proposal for 1500 psig design IWT boiler with economizer for 1300 psig operating and 950^{o}F superheat for 80,000 lbs/hr steam. GTI secured additional funding from UTD to continue working with Nebraska Boiler for the design study of the two-stage industrial watertube boiler as well as pursuing potential host sites.

2011 Q1-2: Work was initiated at Nebraska Boiler on developing IWT boiler conceptual design

2011 Q3: Work was initiated at Nebraska Boiler on developing a full IWT boiler design package

Results and Discussions

As discussed, this project was carried out under the leadership of GTI, and included Cleaver-Brooks, Inc., Nebraska Boiler, a Division of Cleaver-Brooks, and MPT. Also, project advisors include Georgia Institute of Technology for substoichiometric combustion, Alstom Power for HP/HT superheater, PNNL for microchannel economizer, and ORNL for HP/HT materials. Project efforts focused on developing 2nd generation boiler concepts and modeling; incorporating multi-fuel (natural gas and oil) capabilities; assessing heat recovery, heat transfer and steam superheating approaches; and developing and assessing the overall boiler design.

Concepts and Modeling

To facilitate the development effort, GTI conducted Aspen modeling of the two-stage IWT boiler. Detailed schematics and model parameters for the IWT boiler with two-stage combustion system are presented in Appendix A. Two approaches were modeled, one for saturated steam production (sheet 40K lb-h^Sat-v2-1) and another for superheated steam production (sheet 40K lb-h^HP-SSH-1). Both cases are for 40,000 lb/hr steam flow rate. Each case consists of several main blocks and a few auxiliary blocks. The inlet and outlet parameters of each block are presented in the tables located under model schematics. Each stream, before and after the blocks is named and all parameters of the stream are shown in the column with stream name in the top cell.

Both cases, saturated steam boiler and superheated steam boiler, include primary combustion chamber, secondary combustion chamber, convective pass HP economizer. Primary combustion chamber is modeled by a Partial Oxidation Reactor (POR), a primary evaporation section (PR-EV1) and a cooling section (COL-EV2) located before the secondary combustor. The secondary combustion chamber is modeled by a secondary combustor (COMB-2), cooling walls around the combustor (COL-EV3), and a cooling section (SEC-EV4) located between the combustor and the convective pass. The convective section of the boiler is modeled by one block CONV-EV5, and the HP economizer by block ECON-HP.

The primary combustion chamber was modeled based on experimental data from 75 HP and 300 HP two-stage firetube boilers. Primary stoichiometry (air to fuel ratio) was selected as 0.6, exit temperature from primary zone about 1600 to 1700°F, and temperature entering secondary combustion chamber about 1200°F. Fuel gas composition from the primary zone is also close to the actual experimental data, as illustrated in the tables presented in Appendix A.

Total flow of natural gas (NG-POR) is fed to the POR. Total air flow (AIR-IN) is split into primary air (AIR-PR) and secondary air (AIR-SEC). AIR-PR is fed to POR, and AIR-SEC is fed to COMB-2. Combustion products and flue gases are fed from block to block starting from POR and ending in the STACK.

Water and steam flows are modeled respectively starting with feed water inlet (FW-IN) and ending with steam outlet (STM-OUT).

The model for superheated steam boiler, in addition to previous saturated steam case includes a superheating section which consists of a first superheater (SSH-1ST) and a second superheater (SSH-2nd). SSH-1st is located in the primary combustion chamber, and SSH-2nd is located in the secondary combustion chamber, before convective pass.

All parameters for each of the streams are presented in the tables located below of each model schematics. The parameters include mass flow rates, total stream flow rates for each stream component, temperatures and pressures, as well as mole fractions for each of the gas components. Major stream parameters (temperature, pressure, mass and volumetric flow rates) are also shown on the model schematics for each of the streams. All parameters were calculated by Aspen code and present the results of model conversion for mass and heat (energy) balances and chemical reactions after up to 30 iterations.

Two concepts for incorporating two-stage combustion in a modified D type watertube boiler were investigated. One concept is similar to the secondary zone burner arrangement for the two-stage firetube super boiler. This is limited to boiler sizes of less than 100,000 lbs/hr steam flow rate, because of the size of the secondary air duct. The second concept is for larger size watertube boilers of 100,000 lbs/hr to 500,000 lbs/hr and is based on bringing in secondary air through the side of the boiler.

A 34,000 lbs/hr IWT boiler concept based on D-type watertube boilers (physical dimensions for 20,000 lbs/hr conventional D type) with microchannel heat exchangers for high pressure and low pressure economizers and a 20 module TMC2 based on 36-inch long membrane tubes was prepared to support CFD modeling efforts. The size of the 30,000 lbs/hr IWT boiler is based on sizing of a two-stage combustion system, a required interstage cooling section and a convection section from previous firetube super boilers (laboratory and 300 HP or 10,000 lbs/hr).

Layout of a 35,000 lbs/hr IWT boiler with TMC HRS was prepared using conventional D type watertube boiler platform (physical dimensions of existing 20,000 lbs/hr capacity).

For laboratory testing of the IWT boiler concept for a D type watertube boiler, a two-stage burner for 36 MMBtu/hr was preliminarily sized with 18 nozzles. The interstage cooling section was simulated using plate type heat exchangers. A vertical waterwall or refractory section was installed in the furnace section of the boiler to form the two distinct combustions volumes. Secondary combustion air was supplied from the rear of the boiler through a stainless steel pipe into the secondary burner section where the fuel gas would mix with the secondary combustion air in 18 nozzle arrangement.

Nebraska Boiler prepared a proposal for engineering design of a two-stage combustion system for the IWT boiler based on preliminary specifications developed by GTI for the two-stage combustion chamber. It includes primary zone stoichiometry and temperature estimates and interstage cooling and temperature profiles for the primary and the secondary stages. Heat inputs and required surface area were identified with the ASPEN model results. Nebraska Boiler has done some preliminary investigation for the design concept of the two-stage combustion chamber cooling surface for both saturated and superheated steam boilers.

GTI and Nebraska Boiler selected 40,000 lbs/hr, HP/HT superheated steam generator, rated at P=1300 psig and T=950°F for developing the engineering design of IWT boiler. Detailed boiler parameters were calculated by both GTI and Nebraska Boiler. The design basis includes fuel and air flow rates, temperatures and compositions for primary and secondary combustion chambers, as well as for the steam superheaters and the convective pass. Also, a preliminary engineering design of a two-stage 40,000 lbs/hr, 150 psig saturated steam generator was initiated in close communication with Nebraska Boiler.

A conceptual engineering design for a once through 40,000 lbs/hr IWT boiler was developed. The total length of the conceptual boiler is 20 feet and it has a width of 5 feet. There is a convection section between the primary and secondary zones as well as after the secondary zone. Figure 7 is a drawing of the concept.

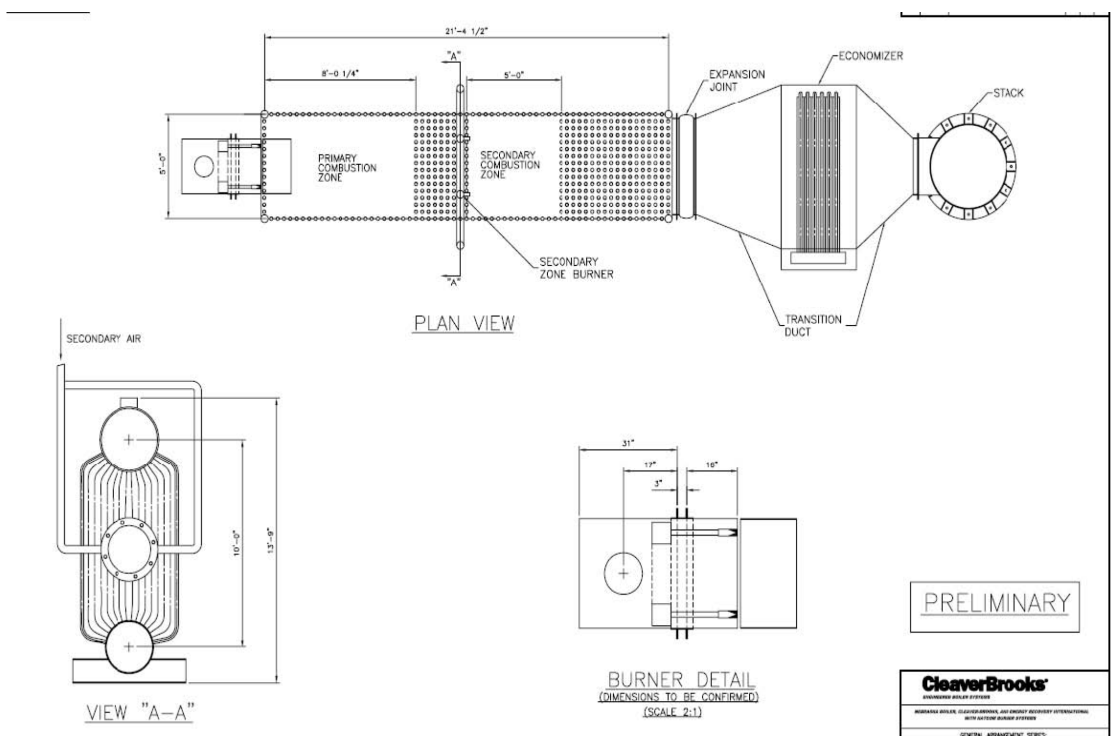

Figure 7. Conceptual Engineering Design for a 40,000 lbs/hr IWT Boiler

Laboratory Boiler Validation

An existing natural gas burner at GTI was modified for dual-fuel firing using a proprietary concept that allows air-staged combustion with oil and rapid switching between natural gas and oil fuel. The modified design includes preheating of oil with superheated steam, which for the laboratory testing was provided by a separate boiler. An 18-inch round burner was installed on a 20-inch-ID boiler simulator for initial testing, and setting up the test rig, and shakedown were completed. Initial tests demonstrated the ability to achieve stable, uniform blue flame combustion firing No.2 oil in single-stage mode from <1 to 5% O_2 flue gas and firing rates up to 4 MMBtu/hr.

The burner was then tested in the GTI 20-inch boiler simulator under fuel-rich conditions to determine flame stability and combustion chemistry. Fluctuations were experienced in oil preheating system temperature that created some flame pulsations and difficulties in providing uniform fuel flow rates to all nozzles. These problems were corrected and testing was continued to acquire detailed data on combustion products at substoichiometric conditions up to 4.3 MMBtu/hr. As expected, NO_x content in the combustion products decreased with decreasing air to fuel ratio, with the lowest level measured at 9.0 ppmv. Flame uniformity to the nozzles was still not satisfactory, suggesting that NO_x could be reduced further with improvements.

Tests were then carried out using the two-stage combustion system on the firetube laboratory boiler at firing rates up to 3 MMBtu/hr (22 gal/hr). Results showed that vaporized No. 2 oil could be combusted in the two-stage mode with a stoichiometric ratio of 0.67 in the primary zone and overall excess air level 15% at the boiler exit. The NO_x level measured in the primary zone was 4 ppmv, while NO_x levels in the stack varied from 20 to 23 ppmv at 3% O_2. An analysis of the No. 2 oil fuel indicated a nitrogen content of 159 ppmw which is equivalent to 21 ppmdv of NO_x in the flue gas assuming all bound nitrogen is converted to NO_x at 3% O_2. This indicates that minimal, if any, thermal NO_x is produced in the two-stage combustion process at these conditions.

In parallel, evaluation of various concepts for vaporization of liquid fuels which were previously identified continued. Biodiesel was successfully vaporized in an external vaporizer and fired in the laboratory two-stage firetube boiler with NO_x less than 20 ppmv. A meeting was held at GTI with experts at Brookhaven National Laboratory (Brookhaven) to discuss biodiesel and other liquid fuel vaporization and combustion in two-stage Super Boiler.

In addition, propane was investigated as a backup fuel for natural gas. It is worth mentioning that the two-stage burner design does not require changing the mixing nozzles for different gaseous fuels, i.e. one mixing nozzle design can handle any gaseous fuel such as natural gas or propane. Although not demonstrated on propane, this feature was previously demonstrated on vaporized liquid fuels such as No. 2 oil and biodiesel.

An alternative concept to full oil vaporization was also assessed. It employs conventional atomization of liquid fuel oil to create distributed oil flames by using multiple oil atomizing nozzles - one for every gas nozzle. The oil atomizing nozzles were located in the center of each of the gas nozzle/spargers. The multiplicity of atomizing nozzles is believed to provide better mixing of atomized oil and air when compared to a single central nozzle. This approach was not expected to prove deep staging that could be achieved with vaporized oil, but allow sufficiently sub-stoichiometric combustion in the first stage to achieve reduced NO_x on No. 2 Oil. It was believed that this alternative approach may allow quicker development and deployment of dual fuel burner while the lower NO_x emission and longer range vaporization concepts were being developed.

To assess the quality of atomization achievable, cold flow testing was conducted with water in a Plexiglas setup using a single primary zone mixing nozzle equipped with commercially available liquid atomizing nozzles. Atomizing particle size and spray angle from various atomizing nozzles were investigated to minimize wetting of the inside of the mixing nozzle. Encouraging results were obtained with a dual fluid atomizer and discussions were held with a leading atomizing nozzle supplier to design and fabricate an atomizer that would meet the temperature and pressure requirements for the 3 MMBtu/hr laboratory burner.

The atomizer was subsequently designed to be incorporated into the natural gas nozzle sparger mixing system of the two-stage burner on the laboratory firetube boiler. Cold flow atomizing tests were conducted at the atomizer vendor's R&D facilities with water to witness atomizing jet spray angle and spray mist particle size, for various water flow rates and other adjustable parameters. Cold flow testing was then conducted at GTI in the Plexiglas setup on the single mixing nozzle. Spray angle from the specially designed atomizing nozzle as well as a commercially available long snout atomizing nozzle were investigated to minimize wetting of the inside of the mixing nozzle.

Hot atomization testing of specially designed atomizer and the long snout atomizer were then conducted with No. 2 oil at 115 psig pressure and 500 F in a modified Plexiglas setup to view the wetting on the mixing section of the nozzle as well as the vaporization rate during flash evaporation. The hot liquid oil was obtained by using existing indirect heat exchangers with steam as the heating medium. For the two atomizers, the spray angle and flash evaporation were studied at No 2 oil flow rates of 0.5 gph to 4 gph at various air atomization flow rates in the single nozzle setup. The long snout atomizer provided very encouraging results. There was little wetting of the mixing section of the single nozzle chamber because of the long snout, which locates the origin point of atomizer mist further downstream in the mixing section and also generates a narrow spray angle. The flash evaporation was estimated to be as high as 50% based on the amount of liquid droplets collected.

A commercial prototype design of the atomizer/vaporizer nozzle was then developed and tested at 80 to 125 psig oil pressure and a temperature of 550°F with No. 2 oil. The design incorporated features to address thermal expansion and provided successful operation during tests with no degradation of internal seals. Both air atomized and steam atomized approaches were tested at oil flow rates over the full range expected for turndown. Results showed acceptable spray angles and no wetting in the mixing section of the burner nozzle. A smaller liquid cap was tested to reduce air or steam atomizing pressure requirements with good results. Eight additional atomizer/vaporizer nozzles with the smaller liquid cap and nine new spargers were procured for the laboratory dual fuel firetube Super Boiler burner.

A nine nozzle laboratory dual fuel burner was subsequently assembled with the prototype atomizer/vaporizer, fabricated by Spraying Systems, along with the new gas plenum and a central oil and air plenum that was designed and fabricated for oil. New gas spargers were fabricated so that the atomizer/vaporizer nozzle could be installed in the center of the primary air/fuel nozzles. The dual fuel burner was then tested with No. 2 fuel oil at a firing rate of 3 MMBtu/hr with 100 psig oil at 550 F under staged conditions with about 7/1 air to fuel ratio in the primary zone. Measurements made in the stack showed NO_x in the range of 17 to 23 ppmv, CO less than 100 ppmv and no soot. Ignition of the oil flame was very smooth and transition to rich combustion in the primary zone with the secondary zone flame was very quick. Both primary and secondary zone flames for the two-stage oil combustion were very stable.

Additional funding was secured from Department of Defense's United Sates Army Construction Engineering Research laboratory (DOD USACERL) for continued development of the multi-fuel (natural gas and liquid fuel, including biofuels) capability of the two-stage combustion process with demonstration targeted for the Rock Island Arsenal. The main emphasis would be on development of a system to preheat the liquid fuel under pressure up to 600°F.

Heat Recovery

Preliminary conceptual designs for an improved TMC geometry for higher flue gas contact at low pressure drop, reduce assembly labor, and reduce manufacturing cost of the TMC module were developed in coordination with Cleaver-Brooks and MPT.

In parallel, PNNL investigated cost effective mircrochannel heat exchanger designs, previously tested at GTI for compact high and low pressure economizers, based on manufacturing techniques developed at PNNL and Battelle. Initially, the design was targeted at a 200 HP boiler; however, subsequently it was decided to target a 300 HP boiler at ORNL as the basis for the microchannel heat exchanger study. Both HP economizer and LP heat exchanger were considered. Issues and benefits of using TMC2 modules versus non-module TMC design were assessed.

It was decided that the size and configuration of the microchannel heat exchanger should closely match the present modular TMC to provide for a more compact arrangement. A draw type modular configuration of the microchannel heat exchanger was considered for integration into the TMC flue gas housing. This configuration involves external distributed water headers for both inlet and outlet. Ideally each module would be a single “microchannel panel”. The layout for the 35,000 lbs/hr watertube boiler has a 36-inch long modular TMC which is possible with the ceramic membrane tubes. Ideally it was envisioned that the microchannel heat exchanger/economizer would be in flat modules for a low profile, although other arrangements such as Chevron configuration, to allow more open area for the flue gas, would conceivably be acceptable depending on its profile. These configurations still involve external distributed water headers for both inlet and outlet. Ideally, it is envisioned that each module would be a single “microchannel panel”.

Another design of the TMC modules considered was based on a two pass configuration on the water side within the TMC module, with an internal water turnaround on one end cap of the module, to allow a back to back arrangement of the TMC modules in the TMC housing. This arrangement would allow extending the modular approach to larger capacity boilers with a more square open area, as opposed to a long and narrow rectangular open area, thereby achieving a closer match to the stack and reducing its footprint. This design was studied with the TMC computer model to explore any other potential benefits (i.e. a two pass design on the water side within a given module may also lead to increased effectiveness of the TMC module and may further reduce the overall number of modules in the vertical plane).

Heat Recovery system designs for integrated HPE/LPE/TMC unit (that is one housing to eliminate transitions to provide a lower profile heat recovery and economically packaged unit) for larger boiler applications were considered. Several ideas were discussed with economizer suppliers. The microchannel heat exchanger was considered in this approach for the LPE. Cost of 34-inch long ceramic membrane tubes in quantities of 7200 tubes, enough for 18 modules for a 1500 HP size boiler was obtained from MPT. The cost is about 15% less than two 17-inch long tubes and accordingly, since the tube sheets and end caps are the same as for the 17-inch long module, the cost of the 34-inch long bundle that has twice the capacity, will be less than 1.5 times the cost of the 17-inch module. Therefore it was determined that there were additional savings in going to the longer modules.

A nine Module TMC HRS system for Baxter Chemical was assembled in the laboratory and flue gases from the laboratory 20 MMBtu/hr industrial watertube boiler were employed to test the TMC HRS prior to its shipment to Baxter in Thousand Oaks, CA. The laboratory setup employed microchannel heat exchangers for the HPE. The two microchannel heat exchangers, previously purchased as part of the firetube Super Boiler program from PNNL, were setup in a parallel configuration per discussions with experts at PNNL. Also an available ID fan with a Variable Frequency Drive (VFD) on the outlet of the TMC was employed to pull the flue gases through the TMC HRS system without putting any additional backpressure on the watertube burner/boiler.

Flue gases from the watertube boiler, equivalent to those from a 10.4 MMBtu/hr burner, were cooled by the mircochannel heat exchanger from 450°F (outlet of the watertube boiler) to less than 300°F at inlet to the LPE of the TMC HRS. The Baxter control system, developed by GTI, was also tested as part of the laboratory setup allowing a check on the operation of the TMC HRS including the preliminary tuning of control loops, TMC pump, and recirculation flow to the Air Heater at design condition. The TMC HRS setup tested in the lab with the microchannel HPE and the ID fan was equivalent to a compact skip mounted unit for watertube boiler application. One of the microchannel heat exchanger developed some minor leaks during the test that were repaired by PNNL.

Heat Transfer

Nebraska Boiler modeled the compact boiler with finned tubes in the convective section with an additional 8 rows of finned tubes and an extended watertube drum. The results show that the length for a 20,000 lbs/hr furnace cross section boiler needed to be only 2 ft longer to achieve 40,000 lbs/hr output. This was a first step in the design of the compact two-stage watertube boiler.

Nebraska Boiler has developed additional concepts for incorporating their finned watertubes to improve the heat transfer in the convective zone of the D or O type watertube boilers.

GTI has developed some novel heat transfer enhancement concepts for significantly reducing the convective paths of boilers.

Superheating

ORNL completed literature study of high temperature materials for potential application for high temperature/high pressure superheater, and prepared test coupon samples of three high temperature alloy materials (Inoconel 617, Haynes 230 and Inconel 740) selected in conjunction with Alstom as potential materials for the superheater for the watertube super boiler concept.

Design Evaluation, Selection, and Scale-Up

As discussed earlier, Nebraska Boiler prepared a proposal for engineering design of a two-stage combustion system for the IWT boiler based on preliminary specifications developed by GTI for the two-stage combustion chamber. It includes primary zone stoichiometry and temperature estimates and interstage cooling and temperature profiles for the primary and the secondary stages. Heat inputs and required surface area were identified with the ASPEN model results. Nebraska Boiler has done some preliminary investigation for the design concept of the two-stage combustion chamber cooling surface for both saturated and superheated steam boilers. This work is expected to be funded by UTD.

GTI and Nebraska Boiler selected 40,000 lbs/hr, HP/HT superheated steam generator, rated at P=1300 psig and T=950°F for developing the engineering design of IWT boiler. Detailed boiler parameters were calculated by both GTI and Nebraska Boiler. The design basis includes fuel and air flow rates, temperatures and compositions for primary and secondary combustion chambers, as well as for the steam superheaters and the convective pass. Also, a preliminary engineering design of a two-stage 40,000 lbs/hr, 150 psig saturated steam generator was initiated in close communication with Nebraska Boiler.

A conceptual design for a once through 40,000 lbs/hr boiler was developed. The total length of the conceptual boiler is 20 feet and it has a width of 5 feet. There is a convection section between the primary and secondary zones as well as after the secondary zone.

Recommendations

Considerable progress was made in the current project on developing and validating key concepts and design schemes for the 2nd generation IWT boiler. Further development, validation and demonstration of key components and subsequently the complete boiler system would require significant additional time and resources. The successful development and commercialization of this technology offers large energy, economic and cost benefits illustrated in Table 3.

Table 3. U.S. Energy Benefits Estimated for 2nd Generation Boiler Technology

Total estimated replacement market (7,716 units)	555,130 MMBtu/h
Annual fuel requirement for existing boilers (27% capacity factor)	1,313 TBtu/year
Annual fuel input for Super Boiler to supply equivalent steam	1,078 TBtu/year
Annual fuel input for electrical equipment at 35% power generation efficiency	46 TBtu/year
Additional fuel used for self-generated power alternative	16 TBtu/year
Annual energy savings with Super Boilers Fuel Cost Savings at $6/MMBtu Average	265 TBtu/year (0.265 Quad) $1.6 billion

Economic Viability

The major market for industrial watertube boilers is replacement of aging boilers. The life of an industrial watertube boiler is about 20 years, and many boilers now operating in the major industrial sectors of the paper products, chemical, food and petroleum industries are well past this expected lifetime. This takes its toll in poor efficiency, high maintenance, and high emissions. A more compact boiler with ultra-high efficiency and ultra-low emissions is very attractive for the replacement boiler as well as for new equipment, particularly where it can compete effectively with field-erected boilers. With the 2nd generation Super Boiler technology, the capacity range of shop-fabricated, transportable packaged boilers—which are typically lower in initial cost and simpler to install—can potentially be extended from 150,000 lb/h to as much as 300,000 lb/h steam capacity. Also, the concept of multiple boilers or modules has gained acceptance in the industrial sector, and if the boiler modules can be more compact and lower in initial cost, less costly to install, and less costly to operate, the market will be even more receptive to the modular approach.

The energy savings enabled by 94% efficiency translate to a two-year payback period for the premium heat recovery system investment. This would accelerate replacement of aging boilers and create business opportunities for major boiler manufacturers, which would in turn justify the capital improvements required in the industry's manufacturing infrastructure to support new boiler design. This would also create industrial sector jobs for manufacturing and installation/construction. Creating an environment of maximum efficiency in the industrial sectors allows them to save money for infrastructure investment. This creates opportunities for increasing capacity, just-in-time products, and higher productivity.

Finally, the extension of the industrial watertube boiler to higher steam pressure and temperature offers the opportunity for high-efficiency power cogeneration with advanced steam turbines. Many large factories and processing facilities currently use steam to drive power generation turbine. Development of a compact ultra-efficient industrial steam generator can provide the impetus for the development of suitable steam turbines to increase efficiency of distributed power. This will help the industrial sector to become more self-sufficient and less vulnerable to grid power failures.

The boiler industry, driven by its customers, is a very conservative one that values safety and reliability above all else. The stepwise implementation of this technology—first to traditional lower-pressure (<250 psig) steam applications, then to higher steam pressure/superheat ranges for which applications already exist (600-900 psig), and finally to 1500°F/1500 psig for advanced cogeneration—will provide a basis for a successful product.

Environmental Benefits

The reduction in emissions resulting from the replacement of current boilers with Super Boilers is shown below. The estimated average U.S. NO_x level from conventional boilers is 63 ppmv[5], and the estimated CO emissions are 400 ppmv[6]. The reductions indicated in **Error! Reference source not found.** amount to an annual decrease of 97% in NO_x emissions and 99% in CO emissions. With the "self-powered" boiler option, it is worth noting that this power could be generated with 2-ppmv NO_x/CO emissions, rather than the typical power plant emissions of 70-200 ppmv NO_x (average ~0.20 lb/MMBtu) and 400-1000 ppmv CO (average ~1.0 lb/MMBtu), depending on the power plant fuel. Also, as a consequence of the higher efficiency of the Super Boiler, CO_2 emissions can potentially be reduced by the amount shown in**Error! Reference source not found.**.

Table 4. U.S. Environmental Benefits from 2nd Generation Super Boiler Technology

Current boiler population emissions	
NO_x (63 ppmv = 0.083 lb/MMBtu)	54,450 ton/year
NO_x additional from grid power required	4,600 ton/year
CO (400 ppmv = 0.32 lb/MMBtu)	441,900 ton/year
CO additional from grid power required	23,000 ton/year
Super Boiler emissions for same steam output	
NO_x (2 ppmv = 0.003 lb/MMBtu)	1,425 ton/year
CO (2 ppmv = 0.003 lb/MMBtu)	1,830 ton/year
Annual NO_x reduction	57,625 tons
Annual CO reduction	463,070 tons
Annual CO_2 reduction	8.8 million tons

5 Basis: 30 ppmv in California and in 40% of non-California jurisdictions, 70 ppmv in unregulated jurisdictions.
6 CO emissions limit in nearly all U.S. jurisdictions

List of Acronyms

Acronym	Description
AIR-IN	Total Air In
AIR-PR	Primary Air In
AIR-SEC	Secondary Air In
CFD	Computer Fluid Dynamics
CHP	Combined Heat and Power
COL-EV2	Cooling Section
COL-EV3	Cooling Walls Around Combustor
COMB-2	Secondary Combustor
CONV-EV5	Convective Section of Boiler
DFSH	Direct-Fired Superheating
DOD	Department of Defense
DOE	Department of Energy
ECON-HP	High Pressure Economizer Section
FEC	Flash Evaporation Cooler
FGR	Flue Gas Recirculation
FW-IN	Feed Water In
GTI	Gas Technology Institute
HAH	Humidifying Air Heater
HHV	Higher Heating Value
HP	High Pressure
HP	High Pressure
HPE	High Pressure Economizer
HRS	Heat Recovery System
HT	High Temperature
ID	Internal Diameter
IWT	Intercooled Watertube Boiler
LNG	Liquefied Natural Gas
LP	Low Pressure
LPE	Low Pressure Economizer
MPT	Media and Process Technology
NG-POR	Natural Gas Flaw Rate to Partial Oxidation Reactor
ORNL	Oakridge National Laboratory
PNNL	Pacific Northwest National Laboratory
POR	Partial Oxidation Reactor
PR-EV1	Primary Evaporation Section
SEC-EV4	Cooling Section
SSH-1st	First Superheater
SSH-2nd	Second Superheater
STACK	Boiler Stack
STM-OUT	Steam Outlet
TE	Thermoelectric
TI	Thermionic
TMC	Transport Membrane Condenser
USACERL	United States Army Construction Engineering Research Laboratory
UTD	Utilization Technology Development
VFD	Variable Frequency Drive
VOC	Volatile Organic Compound

Appendix A. ASPEN PLUS Modeling of 2nd Generation IWT Boiler

Error! Reference source not found. and **Error! Reference source not found.**, and (A1). Detailed Modeling Results for 2nd Generation IWT Boiler, 150 psig, 40,000 lb/hr Saturated Steam**Error! Reference source not found.**, **Error! Reference source not found.**, and Table 4A show detailed schematic drawings and results of Aspen modeling.

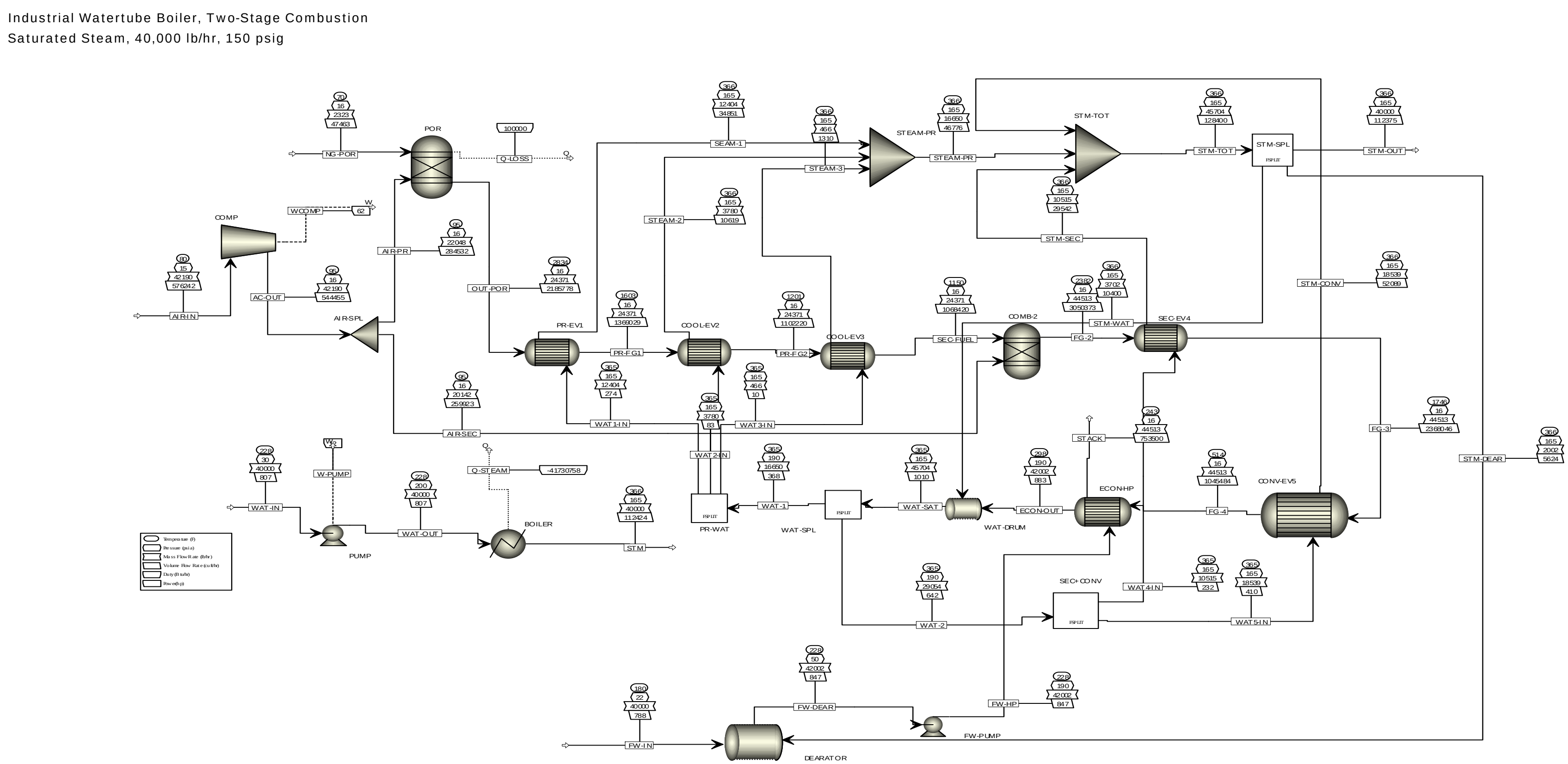

Figure 8 (A1). Schematic for Aspen Model of 2nd Generation IWT Boiler, 150 psig, 40,000 lb/hr Saturated Steam

Table 5 (A1). Detailed Modeling Results for 2nd Generation IWT Boiler, 150 psig, 40,000 lb/hr Saturated Steam

	AC-OUT	AIR-IN	AIR-PR	AIR-SEC	ECON-OU	FG-2	FG-3	FG-4	FW-DEAR	FW-HP	FW-IN	NG-POR	OUT-POR	PR-FG1	PR-FG2	SEAM-1	SEC-FUEL	STACK	STEAM-2	STEAM-3	STEAM-P	STM	STM-CON	STM-DEA	STM-OUT	STM-SEC	STM-TOT	STM-WAT	WAT-1	WAT-2	WAT-IN	WAT-OUT	WAT-SAT	WAT1-IN	WAT2-IN	WAT3-IN	WAT4-IN	WAT5-IN
Temperature F	95.4	80	95.4	95.4	298	2381.9	1746.2	514.4	227.7	228	180	70	2834.1	1603.2	1201.1	366.2	1150.2	242.9	366.2	366.2	366.1	366.4	366.2	366.1	366.1	366.2	366.1	366.1	365.4	365.4	228	228.4	365.4	365.4	365.4	365.4	365.4	365.4
Pressure psia	16	14.7	16	16	190	16	16	16	50	190	22	16	16	16	16	165	16	16	165	165	165	165	165	165	165	165	165	165	190	190	30	200	165	165	165	165	165	165
Vapor Frac	1	1	1	1	0	1	1	1	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0
Mass Flow lb/hr	42190	42190	22048.49	20141.51	42001.83	4.45E+04	4.45E+04	4.45E+04	42001.83	42001.83	40000	2323	2.44E+04	2.44E+04	2.44E+04	12404.18	2.44E+04	44513	3779.529	466.197	16649.91	40000	18539.31	2001.828	40000	10514.62	45703.84	3702.011	16649.91	29053.93	40000	40000	45703.84	12404.18	3779.529	466.197	10514.62	18539.31
Volume Flow cuft/hr	544455.1	576242	284532.2	259922.9	882.878	3.05E+06	2.37E+06	1.05E+06	847.235	847.143	787.836	47462.74	2.19E+06	1.37E+06	1.10E+06	34851.42	1.07E+06	753500	10619.16	1309.852	46776.05	112423.9	52089	5623.911	112375.5	29542.41	128399.8	10400.38	367.722	641.671	806.995	806.887	1009.527	273.989	83.484	10.298	232.252	409.505
Enthalpy MMBtu/hr	-1.491	-1.648	-0.779	-0.712	-278.063	-20.486	-29.815	-46.264	-281.369	-281.342	-270.049	-4.113	-4.992	-15.998	-19.351	-70.143	-19.765	-49.544	-21.372	-2.636	-94.152	-226.187	-104.836	-11.32	-226.192	-59.458	-258.446	-20.934	-108.925	-190.072	-267.95	-267.918	-298.997	-81.149	-24.726	-3.05	-68.787	-121.285
Density lb/cuft	0.077	0.073	0.077	0.077	47.574	0.015	0.019	0.043	49.575	49.581	50.772	0.049	0.011	0.018	0.022	0.356	0.023	0.059	0.356	0.356	0.356	0.356	0.356	0.356	0.356	0.356	0.356	0.356	45.279	45.279	49.567	49.573	45.273	45.273	45.273	45.273	45.273	45.273
Mass Flow lb/hr																																						
H2O	289.901	289.901	151.502	138.399	42001.83	4995.288	4995.288	4995.288	42001.83	42001.83	40000	0	2919.33	2919.33	2919.33	12404.18	2919.33	4995.288	3779.529	466.197	16649.91	40000	18539.31	2001.828	40000	10514.62	45703.84	3702.011	16649.91	29053.93	40000	40000	45703.84	12404.18	3779.529	466.197	10514.62	18539.31
N2	31637.4	31637.4	16533.71	15103.7	0	31825.02	31825.02	31825.02	0	0	0	187.617	16721.32	16721.32	16721.32	0	16721.32	31825.02	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
O2	9736.737	9736.737	5088.419	4648.318	0	1271.915	1271.915	1271.915	0	0	0	0	0	0	0	0	0	1271.915	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NO	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
H2	0	0	0	0	0	0.017	0.017	0.017	0	0	0	0	180.374	180.374	180.374	0	180.374	0.017	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CL2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCL	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CO	0	0	0	0	0	0.338	0.338	0.338	0	0	0	0	2392.409	2392.409	2392.409	0	2392.409	0.338	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CO2	0	0	0	0	0	5894.464	5894.464	5894.464	0	0	0	0	1738.141	1738.141	1738.141	0	1738.141	5894.464	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CH4	0	0	0	0	0	0	0	0	0	0	0	1933.996	145.05	145.05	145.05	0	145.05	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
H2S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
COS	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCN	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NH3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H6	0	0	0	0	0	0	0	0	0	0	0	201.387	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C3H8	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C4H10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
AR	525.962	525.962	274.867	251.094	0	525.962	525.962	525.962	0	0	0	0	274.867	274.867	274.867	0	274.867	525.962	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
N-PEN-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Table 6 (A2). Mole Fractions for Each Stream; Modeling Results for 2nd Generation IWT Boiler, 150 psig, 40,000 lb/hr Saturated Steam

	AC-OUT	AIR-IN	AIR-PR	AIR-SEC	ECON-OU	FG-2	FG-3	FG-4	FW-DEAR	FW-HP	FW-IN	NG-POR	OUT-POR	PR-FG1	PR-FG2	SEAM-1	SEC-FUEL	STACK
Mole Frac																		
H2O	0.011	0.011	0.011	0.011	1	0.173	0.173	0.173	1	1	1	0	0.164	0.164	0.164	1	0.164	0.173
N2	0.772	0.772	0.772	0.772	0	0.71	0.71	0.71	0	0	0	0.05	0.603	0.603	0.603	0	0.603	0.71
O2	0.208	0.208	0.208	0.208	0	0.025	0.025	0.025	0	0	0	0	0	0	0	0	0	0.025
NO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NO	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
H2	0	0	0	0	0	0	0	0	0	0	0	0	0.09	0.09	0.09	0	0.09	0
CL2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCL	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CO	0	0	0	0	0	0	0	0	0	0	0	0	0.086	0.086	0.086	0	0.086	0
CO2	0	0	0	0	0	0.084	0.084	0.084	0	0	0	0	0.04	0.04	0.04	0	0.04	0.084
CH4	0	0	0	0	0	0	0	0	0	0	0	0.9	0.009	0.009	0.009	0	0.009	0
H2S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
COS	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCN	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NH3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H6	0	0	0	0	0	0	0	0	0	0	0	0.05	0	0	0	0	0	0
C3H8	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C4H10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
AR	0.009	0.009	0.009	0.009	0	0.008	0.008	0.008	0	0	0	0	0.007	0.007	0.007	0	0.007	0.008
N-PEN-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
N-HEX-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Industrial Watertube Boiler, Two-Stage Combustion
Superheated Steam, 40,000 lb/hr, 1300 psig, 950 F

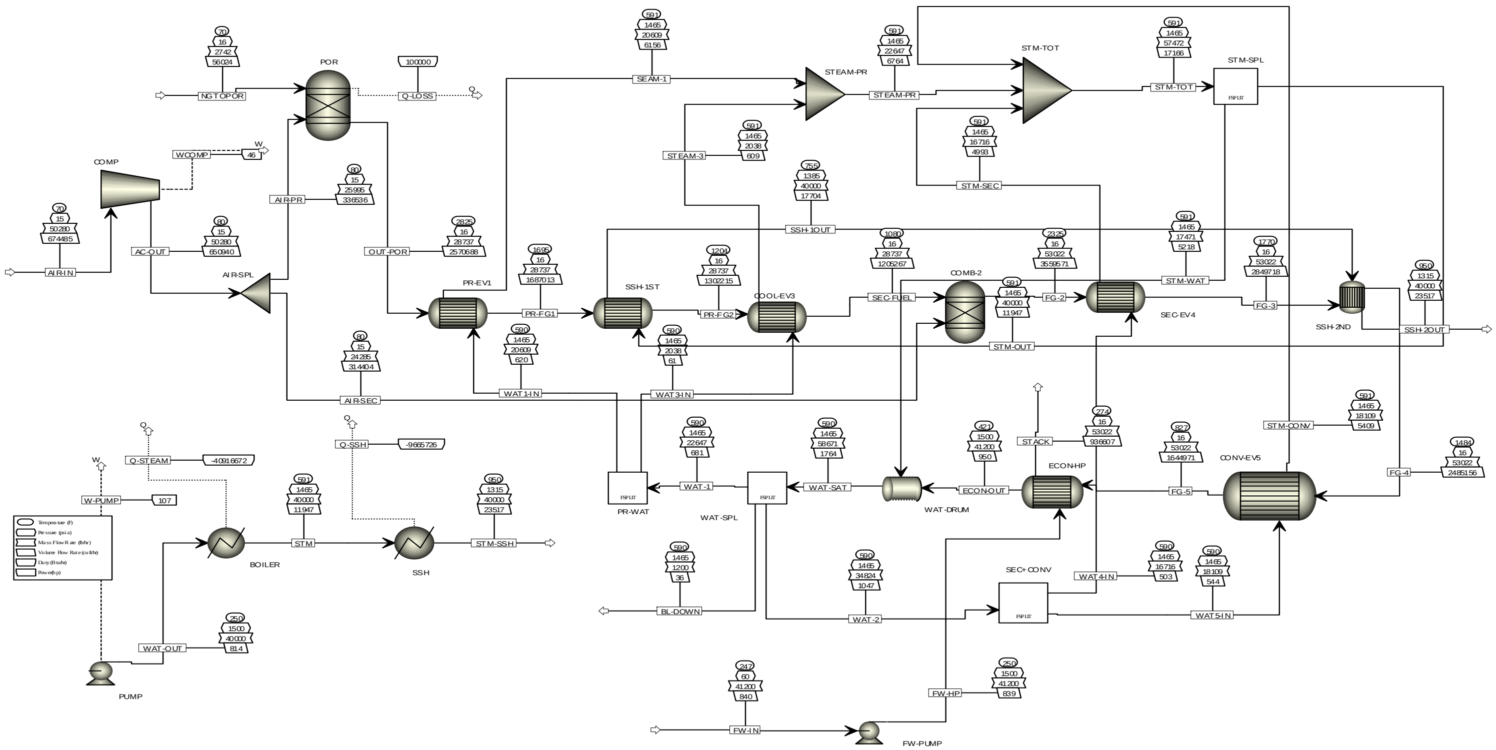

Figure 9 (A2). Schematic for Aspen Model of 2nd Generation IWT Boiler, 1300 psig, 950 F, 40,000 lb/hr Saperheated Steam

Table 7 (A3). Detailed Modeling Results for 2nd Generation IWT Boiler, 1300 psig, 950 F, 40,000 lb/hr Superheated Steam

	AC-OUT	AIR-IN	AIR-PR	AIR-SEC	BL-DOWN	ECON-OU	FG-2	FG-3	FG-4	FG-5	FW-HP	FW-IN	NGTOPOR	OUT-POR	PR-FG1	PR-FG2	SEAM-1	SEC-FUEL	SSH-1OUT
Temperature F	79.9	70.4	79.9	79.9	590.1	421	2325	1769.6	1484.3	827.1	250.2	247	70	2824.5	1695.4	1203.8	591.5	1079.9	755
Pressure psia	15.5	14.7	15.5	15.5	1465	1500	16	16	16	16	1500	60	16	16	16	16	1465	16	1385
Vapor Frac	1	1	1	1	0	0	1	1	1	1	0	0	1	1	1	1	1	1	1
Mass Flow lb/hr	50280.5	50280.5	25995.02	24285.48	1199.829	41200	5.30E+04	5.30E+04	5.30E+04	5.30E+04	41200	41200	2742	2.87E+04	2.87E+04	2.87E+04	20608.9	2.87E+04	40000.17
Volume Flow cuft/hr	650940.2	674485.3	336536.1	314404.1	36.075	949.786	3.56E+06	2.85E+06	2.49E+06	1.64E+06	838.823	839.868	56023.61	2.57E+06	1.69E+06	1.30E+06	6155.522	1.21E+06	17703.82
Enthalpy MMBtu/hr	-1.966	-2.08	-1.016	-0.949	-7.472	-266.7	-24.917	-34.605	-39.41	-49.905	-274.841	-275.121	-4.855	-5.971	-17.915	-22.776	-116.399	-23.957	-221.06
Density lb/cuft	0.077	0.075	0.077	0.077	33.259	43.378	0.015	0.019	0.021	0.032	49.116	49.055	0.049	0.011	0.017	0.022	3.348	0.024	2.259
Mass Flow lb/hr																			
H2O	345.494	345.494	178.62	166.873	1199.829	41200	5899.655	5899.655	5899.655	5899.655	41200	41200	0	3438.266	3438.266	3438.266	20608.9	3438.266	40000.17
N2	37704.3	37704.3	19493.12	18211.18	0	0	37925.75	37925.75	37925.75	37925.75	0	0	221.457	19714.58	19714.58	19714.58	0	19714.58	0
O2	11603.89	11603.89	5999.21	5604.678	0	0	1612.123	1612.123	1612.123	1612.123	0	0	0	0	0	0	0	0	0
NO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NO	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
H2	0	0	0	0	0	0	0.013	0.013	0.013	0.013	0	0	0	213.738	213.738	213.738	0	213.738	0
CL2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCL	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CO	0	0	0	0	0	0	0.25	0.25	0.25	0.25	0	0	0	2824.667	2824.667	2824.667	0	2824.667	0
CO2	0	0	0	0	0	0	6957.883	6957.883	6957.883	6957.883	0	0	0	2050.49	2050.49	2050.49	0	2050.49	0
CH4	0	0	0	0	0	0	0	0	0	0	0	0	2282.831	171.212	171.212	171.212	0	171.212	0
H2S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
COS	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCN	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NH3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H6	0	0	0	0	0	0	0	0	0	0	0	0	237.712	0	0	0	0	0	0
C3H8	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C4H10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
AR	626.822	626.822	324.067	302.755	0	0	626.822	626.822	626.822	626.822	0	0	0	324.067	324.067	324.067	0	324.067	0
N-PEN-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
N-HEX-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

	SSH-2OUT	STACK	STEAM-3	STEAM-PI	STM	STM-CON	STM-OUT	STM-SEC	STM-SSH	STM-TOT	STM-WAT	WAT-1	WAT-2	WAT-IN	WAT-OUT	WAT-SAT	WAT1-IN	WAT3-IN	WAT4-IN	WAT5-IN
Temperature F	950	273.6	591.5	591.5	591.5	591.5	591.5	591.5	950	591.5	591.5	590.1	590.1	247	250.2	590.1	590.1	590.1	590.1	590.1
Pressure psia	1315	16	1465	1465	1465	1465	1465	1465	1315	1465	1465	1465	1465	60	1500	1465	1465	1465	1465	1465
Vapor Frac	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0
Mass Flow lb/hr	40000.17	53022.5	2038.242	22647.14	40000	18108.67	40000.17	16715.7	40000	57471.51	17471.34	22647.14	34824.37	40000	40000	58671.34	20608.9	2038.242	16715.7	18108.67
Volume Flow cuft/hr	23517.44	936607.4	608.788	6764.31	11947.29	5408.749	11947.36	4992.692	23517.34	17165.75	5218.388	680.928	1047.06	815.405	814.392	1764.063	619.645	61.284	502.589	544.471
Enthalpy MMBtu/hr	-216.255	-58.045	-11.512	-127.911	-225.919	-102.278	-225.92	-94.41	-216.254	-324.598	-98.678	-141.036	-216.87	-267.108	-266.836	-365.378	-128.343	-12.693	-104.098	-112.773
Density lb/cuft	1.701	0.057	3.348	3.348	3.348	3.348	3.348	3.348	1.701	3.348	3.348	33.259	33.259	49.055	49.116	33.259	33.259	33.259	33.259	33.259
Mass Flow lb/hr																				
H2O	40000.17	5899.655	2038.242	22647.14	40000	18108.67	40000.17	16715.7	40000	57471.51	17471.34	22647.14	34824.37	40000	40000	58671.34	20608.9	2038.242	16715.7	18108.67
N2	0	37925.75	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
O2	0	1612.123	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NO	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
H2	0	0.013	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CL2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCL	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CO	0	0.25	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CO2	0	6957.883	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CH4	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
H2S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
COS	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCN	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NH3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H6	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C3H8	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C4H10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
AR	0	626.822	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
N-PEN-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
N-HEX-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Table 8 (A4). Mole Fractions for Each Stream; Modeling Results for 2^{nd} Generation IWT Boiler, 1300 psig, 950 F, 40,000 lb/hr Superheated Steam

	AC-OUT	AIR-IN	AIR-PR	AIR-SEC	BL-DOWN	ECON-OU	FG-2	FG-3	FG-4	FG-5	FW-HP	FW-IN	NGTOPOR	OUT-POR	PR-FG1	PR-FG2	SEAM-1	SEC-FUEL	SSH-1OUT	SSH-2OUT	STACK
Mole Frac																					
H2O	0.011	0.011	0.011	0.011	1	1	0.172	0.172	0.172	0.172	1	1	0	0.164	0.164	0.164	1	0.164	1	1	0.172
N2	0.772	0.772	0.772	0.772	0	0	0.71	0.71	0.71	0.71	0	0	0.05	0.603	0.603	0.603	0	0.603	0	0	0.71
O2	0.208	0.208	0.208	0.208	0	0	0.026	0.026	0.026	0.026	0	0	0	0	0	0	0	0	0	0	0.026
NO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NO	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
SO3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
H2	0	0	0	0	0	0	0	0	0	0	0	0	0	0.091	0.091	0.091	0	0.091	0	0	0
CL2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCL	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
CO	0	0	0	0	0	0	0	0	0	0	0	0	0	0.086	0.086	0.086	0	0.086	0	0	0
CO2	0	0	0	0	0	0	0.083	0.083	0.083	0.083	0	0	0	0.04	0.04	0.04	0	0.04	0	0	0.083
CH4	0	0	0	0	0	0	0	0	0	0	0	0	0.9	0.009	0.009	0.009	0	0.009	0	0	0
H2S	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
COS	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
HCN	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
NH3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H6	0	0	0	0	0	0	0	0	0	0	0	0	0.05	0	0	0	0	0	0	0	0
C3H8	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C4H10	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
C2H2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
AR	0.009	0.009	0.009	0.009	0	0	0.008	0.008	0.008	0.008	0	0	0	0.007	0.007	0.007	0	0.007	0	0	0.008
N-PEN-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
N-HEX-01	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

END OF REPORT