Measurements and Design Enhancements in Firetube Boilers Using Improved Technology

David Thornock
Johnston Boiler Company

Hartman Mitchell
Fintube Technologies

ABSTRACT

Firetube, Scotch Marine boilers have been used throughout the world for over a century and have been one of the mainstays behind process steam applications and industrial heating plants. The design of the current boiler products has been in place for well over 30 years. However, both market conditions and a demand for furnace chambers that will be compatible to low NO_x combustion systems have caused Johnston Boiler Company to review their firetube boiler design.

This paper will report in-flame temperature measurements, utilizing precise pyrometry techniques, which were taken over a wide range of boiler sizes. These temperature measurements document the effect of furnace size, furnace volume and flame area on heat transfer and NO_x formation. In addition to the in-flame temperature measurements, inlet and outlet temperatures were taken over tubes with augmented heat transfer surfaces to document the effect of this tube construction on overall boiler convection performance. Lastly, this paper will address the advantages of using augmented surface tubes for boiler design to increase steam volume and quality.

INTRODUCTION

Johnston Boiler Company has been making boilers on the same site for 136 years and the purpose of these boilers has not changed from the very early days. Steam has been used for many different things, but the actual steam generation has remained the same. Over the course of history, boilers have improved with changing metal working technology, fuel availability, and increases in combustion and heat transfer technology. We have changed from a surface area to horsepower ratio of 10 ft^2/boiler horsepower to 5 ft^2/boiler horsepower for most applications. With the improvements in metal quality and welding procedures we no longer have to include long stayrods and thousands of rivets. The challenge that all boiler manufacturers face is in utilizing their past experience and current technology to produce boilers that have an advantage for the different processes to which they will be applied. This advantage can be gained from application information derived through practical experience and can be as extensive as using computational fluid dynamic models to simulate the combustion and heat transfer processes. This paper will describe some of the techniques that Johnston Boiler Company, with the support of Fintube Technologies, is currently using to improve performance and functionality of its boiler series.

Figure 1 shows a cross section of a typical water-backed firetube boiler. The burner is mounted at the face of the boiler and fires directly down the furnace. All of the hot flue gases exit the furnace into the combustion chamber (or turn around). From there, the flue gases enter the convective sections of the boiler. Depending on the pass arrangement of the boiler, the flue gases make 1, 2 or 3 passes through the water stored in the pressure vessel. For a 2-pass boiler, there is only one bank of tubes, for a 3-pass boiler there are 2 tube passes and for a 4-pass boiler there are 3 tube passes. After leaving the convective section of the boiler, the flue gases exit the pressure vessel at the stack. This is the type of boiler used for all of the data collection and analyses for this paper.

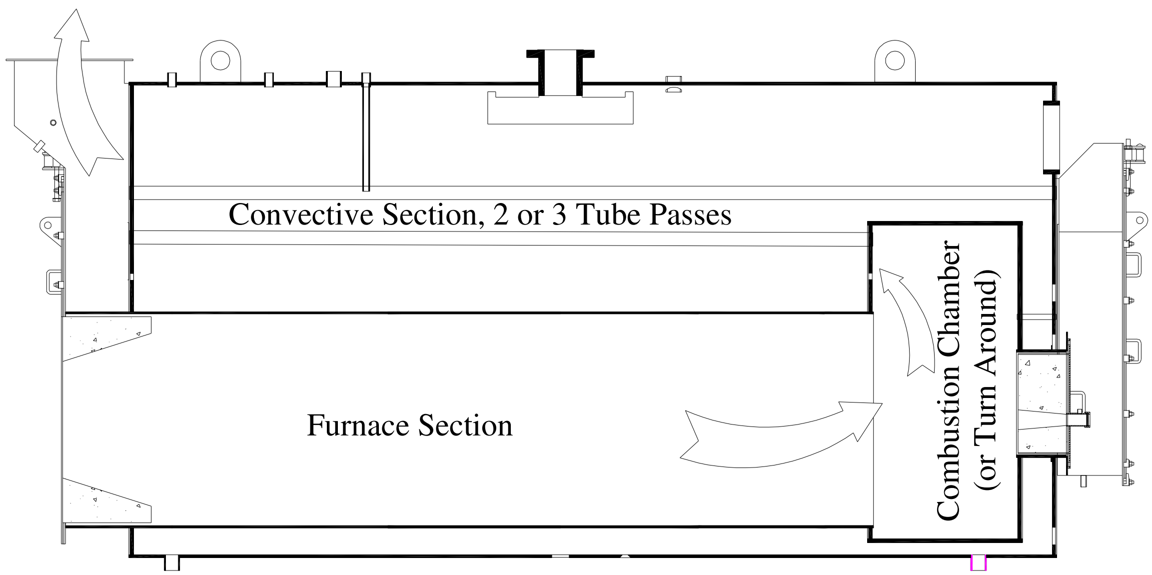


Figure 1 – Cross Section of a Typical Firetube Boiler

At the heart of any boiler design is overall boiler efficiency. A good boiler design must take into account several parameters and, at the same time, maintain a good overall efficiency. These other parameters include pressure drop, furnace size and heat release rate, steam height and storage, and shell diameter and length. Each of these different design parameters must be considered in order to produce a boiler package that will meet the end user's need in the most cost effective manner. The technology used to improve the design of the boilers with these parameters in mind will be discussed individually.

Boiler Surface Area Optimization

A longstanding rule of thumb for firetube boilers is 5 ft^2 of heating surface area per boiler horsepower. This number has been used in the ASME code to set firing rates and steam generation capacities and is typical for the majority of boiler manufacturer's designs. However, through the use of new tube technology, the 5 ft^2/bhp can be optimized to lower values without any adverse affects to the boiler performance. With this in mind, the boiler designer must first decide on the split between how much heat is absorbed in the furnace versus what is absorbed in the convective passes. Once the heat absorption split has been decided, the next step is to generate tube patterns in an optimum arrangement for convective heat transfer and flue side pressure drop. All of this must be done while conserving size and keeping the overall boiler reasonably priced.

To aid in the process of tube surface arrangement, Johnston Boiler Company has developed a computational program that allows us to optimize the flue side pressure drop and heat transfer through any arrangement of 3 or 4 pass boilers. Finetube Technologies has also developed a similar program for the X-ID tubes. The critical inputs to both programs are temperature and flow rate of combustion products entering the tube sheet. In an effort to define these temperatures, Johnston Boiler Company has used a High Velocity Thermocouple (HVT). Figure 2 shows a cross section of the HVT. The hot flue gases are pulled at a high velocity through the outer ceramic tip through a single hole on the side. Combustion gases are then drawn across the type B thermocouple using convective heat transfer[1] to bring the thermocouple bead to the temperature of the hot gases. The ceramic tips on the end of the probe serve as a radiation shield and effectively eliminate radiation from the flame to the thermocouple and from the thermocouple to the colder walls.

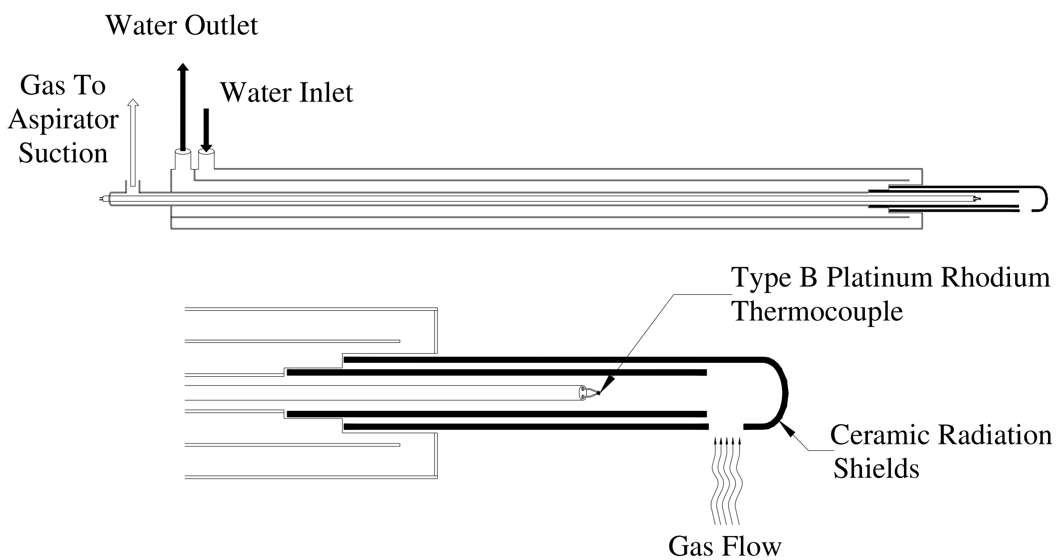


Figure 2 Cross Section of a High Velocity Thermocouple (HVT)

Temperature measurements have been taken along the centerline of the furnaces and combustion chambers (or rear, water cooled turnarounds) for several different size boilers. Figure 3 shows a portion of this data. Because of the fixed length of our HVT we were only able to measure the peak flame temperature on the 125 hp boiler. Our measured peak flame temperature was approximately 2,880°F along the center of the furnace. For the 400 hp boiler, the probe was near the peak and measured 2,831°F. The temperature profile for the 2,000 hp boiler shows that our probe did not extend into the furnace far enough to reach the flame front. It should also be noted here that the temperatures recorded are average temperatures and that the flame in the furnace is far from steady state and does not produce a steady state temperature profile. The instantaneous temperature readings would bounce ± 50°F on average. In the flame front, the swings were amplified even more and would swing ± 100°F.

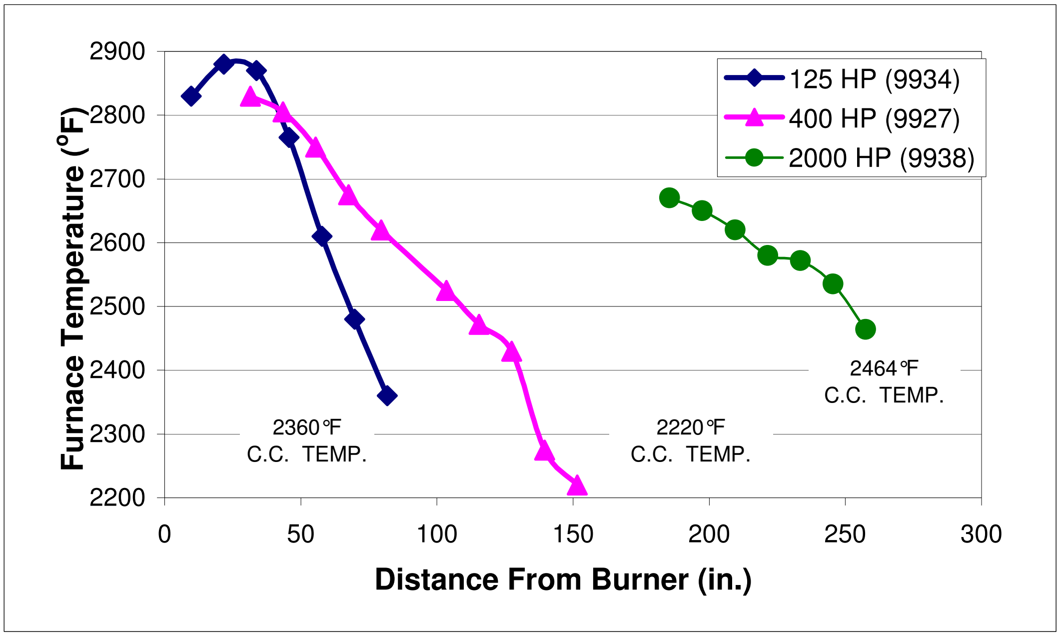


Figure 3 – In-flame Temperature Measurements

This temperature data has given us some critical information to help size and optimize the furnace chamber. Johnston Boiler Company has set a standard of 150,000 Btu/Ft^3 for the Furnace Volumetric Heat Release Rate (the fuel thermal input to the boiler divided by the total furnace volume, VHRR). This standard was derived from many years of both burner and boiler experience. The scientific reasons include the following 1) the higher the heat release rate the harder it is to fire fuel oil, 2) smaller diameter furnaces do not allow good burner aerodynamics to be established in the furnace volume, and 3) higher heat release rates generate higher temperature profiles corresponding to higher NO_x formation.

Figure 4 is a comparison of VHRR with the predicted furnace exit gas temperature. Predicted furnace exit gas temperatures were developed by Fintube using the computational fluid dynamic code FLUENT. The FLUENT submodels were set up to model the combustion processes in a Johnston Boiler furnace with the following parameters: 1) a typical natural gas (90% CH_4, 2.5% C_2H_6, and 7.5% N_2), 2) 19.02% Excess Air (3% O_2 after combustion), heat input equaling a 600 hp, and 3 different furnace sizes with corresponding heat release rates shown in the figure. This graph shows very well the trend associated with increasing VHRR. As the ratio of firing rate to gas volume increases, the furnace exit temperature increases dramatically.

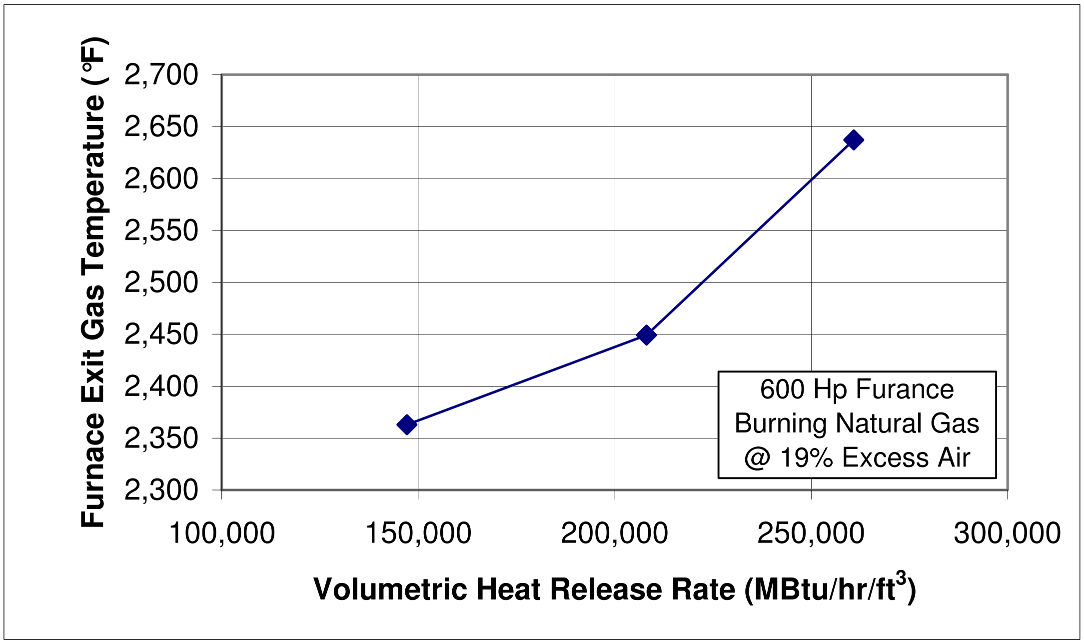


Figure 4 – Comparison of Predicted Furnace Exit Gas Temperatures with VHRR

This VHRR and furnace exit gas temperature trend is also detrimental to NO_x formation. Burning natural gas, approximately 95% of the NO_x generated comes from the thermal NO_x formation mechanism. The thermal NO_x formation mechanism was first described in 1946 by Zeldovich[2]. Thermal NO_x comes from the fixation of atmospheric nitrogen when it is heated to flame temperatures in the presence of oxygen and other NO_x precursors. The thermal dissociation of elemental nitrogen in a flame front is exponentially dependent on temperature and time. The reaction starts to take place at temperatures between 1,500°F and 1,800°F. Once these reactions have started, they grow exponentially with temperature. If there is no oxygen present once dissociation has occurred, the nitrogen radicals will recombine with the other nitrogen radicals to form elemental nitrogen. Therefore, the NO_x formed is very dependent on the oxygen concentration available locally where the highest temperatures are achieved. Typically at the flame front, there is a wide variety of oxygen concentrations ranging from 0% at the flame front, where all of the oxygen is consumed by the fuel, to 21% at the inlet of the combustion air. At various stages of the combustion process the combustion gases and the nitrogen compounds will be exposed to oxygen concentrations and temperatures that are conducive to the formation of NO_x.

Advanced staged low NO_x burners control the mixing of the combustion air with the fuel at the flame front. By controlling how the combustion air mixes with the fuel, the rate of combustion can also be controlled. When the peak flame temperature has passed, the rest of the combustion air can be mixed in, eliminating a substantial amount of NO formation. Other low NO_x combustion systems reduce the NO_x formation by diluting O_2 concentration at the flame front with FGR. Using FGR also lowers the peak flame temperature, which directly affects the formation of the thermal NO_x.

Using techniques to lower the overall flame temperature can be successful at reducing NO_x formation; however, lower flame temperatures also affect heat transfer and boiler performance. Because radiation heat[3] is transferred at $Q=\varepsilon\alpha(T^4-T^4_{surr})$, the decrease in flame temperature also inhibits radiative heat transfer in the furnace. In the furnace of a firetube boiler, radiative heat transfer is the main mode of heat transfer. Lowering the flame temperature reduces the amount of heat removed from the combustion gases and increases the temperature of the gases entering the convective sections of the boiler. By using FGR, the total amount of gases entering the boiler is also increased by the percentage of FGR used. This increase in gas flow also increases the heat losses at the exit of the boiler.

The next piece of data that is needed for the design model is the temperature entering the first pass of tubes. To get this temperature information, we used a 600 hp boiler and placed thermocouples at the entrance to the first tube pass. The thermocouples were not HVTs, but were designed with a shield to eliminate the effects of radiation and thus measure only the gas temperature. Figure 5 shows a set of temperature data that was taken firing the boiler over a 5 to 1 turndown ratio. The data labeled rear tube inlet top and bottom are at the entrance to the tube pass. It is interesting to note that the peak gas temperature entering the tubes was measured at 1,857°F. This is nowhere near the measured and predicted temperatures exiting the furnace. All of the Johnston boilers have a water back design which effectively removes enough energy from the gas stream to reduce the bulk gas temperature by 200 to 400°F. Fintube, again using their computational fluid dynamic model (FLUENT), has confirmed these values. It should be noted that large boilers have temperatures entering the convection pass often in excess of 2,300°F. This information has been critical to evaluating our computational models and algorithms.

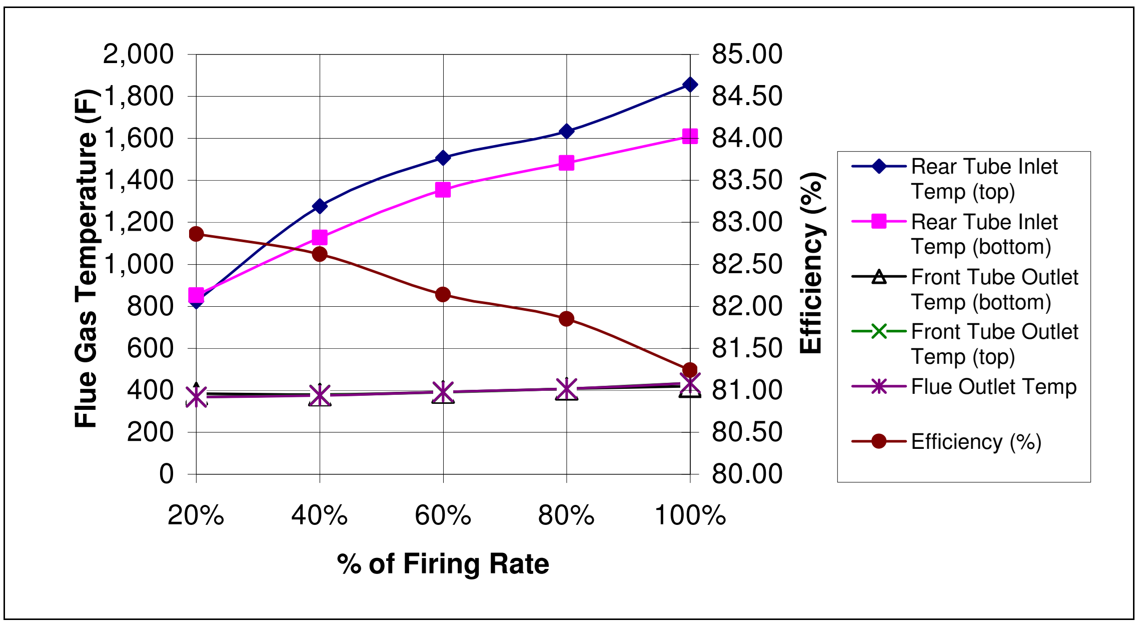


Figure 5 – Tube Entrance and Exit Temperatures Combined with Measured Efficiency (600 hp – 2 Pass XID Boiler operating at 100 psig steam pressure)

Steam Storage, Steam Quality and Steam Height

Critical to some processes, is the boiler's ability to handle rapid and large load swings. These processes may occur when cookers, dryers or any other piece of equipment is brought on and off line frequently, or when the demand for steam is cycled as a function of varying production lines. Regardless of what creates the cycling load conditions, the volume of available steam greatly effects how the boiler will handle the change. Larger steam storage volumes allow the boiler to handle the load swings better because they have larger steam reservoirs at the operating temperature and pressure from which to draw. As the load demand increases, the larger the steam storage, the longer it takes for the overall boiler pressure to drop. The key is to have large enough steam storage to allow the burner to reach high fire and catch up with the new load demand, before the boiler pressure drops too far. The larger steam volume in the chest of the boiler acts like a separate steam drum/accumulator that was commonly used on the top of older boiler designs.

Equally critical to many processes is steam quality. While steam quality is influenced by water chemistry, the two design parameters that affect it the most are steam nozzle size and steam height. The steam nozzle size is easy to change and should limit the steam velocity to at or below 80 ft/sec. Steam height is important because as the water boils, water droplets get entrained with the steam as it leaves the surface. The quantity of these water droplets leaving the boiler with the steam is a function of how far they have to travel before exiting the boiler and the velocity of the steam traveling through the boiler steam cavity. The water droplets are heavier than steam and can easily be eliminated by reducing the steam velocity and allowing them to drop out of the vapor flow. Larger steam volumes and steam heights allow more time for gravity to pull these water droplets back to the water surface.

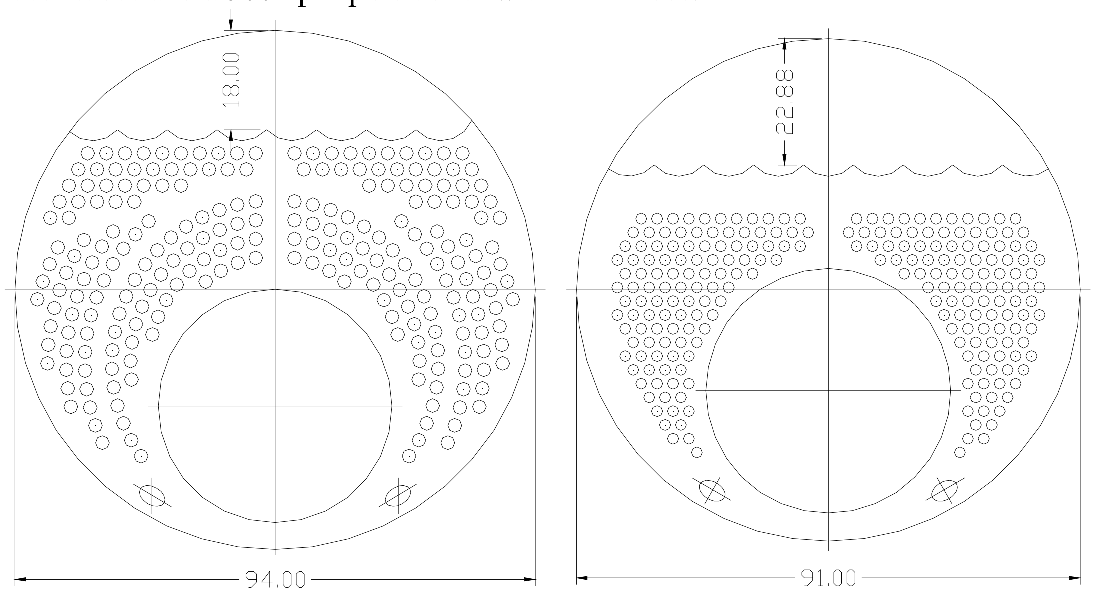
The steam height is not difficult to adjust, but normally adds significant cost. Once the heating surface is determined, the minimum size of the boiler is also developed. Increasing the steam volume and height then becomes a cost consideration. The shell material thickness increases with increasing shell diameter. The following equation is taken from Section I of the ASME Boiler and Pressure Vessel Code[4].

$$t_{shell} = \frac{P_{design} D_{shell}}{2S_{shellmaterial} E + 2yP_{design}} + C$$

where P_{design} is the maximum allowable working pressure, $S_{shellmaterial}$ is the maximum allowable metal stress value at the design temperature, E is the efficiency factor related to the type of weld used, y is a temperature coefficient, C is an allowance for threading or structural stability, D_{shell} is the diameter of the shell, and t_{shell} is the thickness of the boiler shell. Holding the design pressure and the other material related variables constant, shows that shell diameter directly affects the required shell thickness. The shell is also one of the largest components of the boiler and as such, is one of the factors that boiler manufactures often look to reduce costs. Therefore, the design of the steam storage volume is a significant parameter to be included in the overall boiler design for cost as well as performance considerations.

Four-pass boilers were designed to achieve higher fuel to steam efficiencies. Four-Pass boilers were initially built by adding a 3rd tube pass and redistributing the heating surface to optimize the pressure drop and take advantage of the additional tube length. While the 4-pass boiler has the highest efficiency of any of the firetube boilers, typically, its steam volume is smaller than that of other designs. This is a typical result from attempting to minimize the difference in cost for the shell between the 3- and 4-pass designs. Designing a boiler for larger steam volume also significantly increases the cost of the boiler and is not commonly applied. In answer to this problem, Johnston Boiler has joined forces with Fintube Technologies to design a 2-pass boiler utilizing their X-ID boiler tube. Their X-ID boiler tubes allow us to design a pressure vessel that has significant advantages in steam storage area.

The 2-pass boiler has been around for many years. The reason most designers left the 2-pass platform was the improved efficiency of additional tube surface in a 3- or 4-pass design. However, with the new technology provided by the X-ID tubes, the efficiency of the boiler tube surface is improved by 78%. This increase in convective heat transfer allows us to revisit the 2-pass boiler design and make it as efficient as a 3-pass boiler and much closer to the design efficiencies of a 4-pass. The advantage to Johnston of this design is that the 2-pass X-ID boiler allows the optimization of the boiler shell for larger steam volumes and heights. Figure 6 is a comparison of a 500 hp 4-pass boiler with smooth tubes and a 500 hp 2-pass boiler with XID tubes.



1st Tube Pass – 90 tubes
2nd Tube Pass – 74 tubes
3rd Tube Pass – 68 tubes
Total Tubes – 232
(2.5" Tubes)

Single Tube Pass
Total Tubes – 224
(2.0" Tubes)

Figure 6 – Tube Layout Comparison of Johnston Boiler Company's 500 hp

509 Series and XID Series Boilers

The head pattern and tube layout on the left is a 4-pass boiler with a total of 232 - 2.5"tubes. The head pattern and tube layout on the right is a 2-pass boiler with 224 – 2.0" tubes. The length of the two boilers is the same and the diameter of the shell is only 3 inches different (XID boiler shell is 3"smaller). The tube arrangement for the 2-pass boiler allows a significant increase in the steam height of 4.88" and an increase of over 60 ft^3 of steam storage volume. This boiler was first designed for use in the corrugation industry where it is imperative that they have dry steam, even when they are rapidly changing the load conditions by bringing multiple process heaters and ovens online and offline. Some of the design parameters for both boilers are listed in Table 1 for comparison purposes.

Table 1 – Comparison of Johnston Boiler Company's 500 hp 509 and X-ID Boilers

	509 Series 4 Pass Boiler	X-ID Series 2 Pass Boiler
Heating Surface (ft^2)	2,558	2,126
Heating Surface/Boiler HP	5.12	4.25
Steam Volume (ft^3)	95.36	155.6
Steam Height (in)	18.0	22.88
Boiler Efficiency (%)	82.4	81.2
Boiler Pressure Drop (in H_2O)	6.0	2.5

The disadvantage of the X-ID tube is that while it increases the convective heat transfer substantially, it also substantially increases the pressure drop over an equivalent length of tube surface. If the X-ID tubes were used as a direct replacement for the straight tubes in a 3- or 4-pass boiler, the pressure drop across the boiler would be nearly double that of the same smooth tube design. The 2-pass boiler does not have this problem. Because all of the flue gases exiting the furnace are spread over only one tube pass, the velocity through the single tube pass is significantly lower than what is required for the gases to travel across 2 or more tube passes of the 3- or 4-pass design boilers. Table 1 also shows the pressure drop across the two different boiler designs.

The augmented tubes have also been used for special applications where the size of the boiler footprint is critical. In these situations, the steam height and steam volume are kept the same when compared to typical firetube boilers and the shell size is reduced. This has been very effective for several applications. Johnston Boiler Company's standard 1,500 hp, 4-pass boiler has a shell diameter of 133 inches. When the boiler was put into the 2-pass boiler configuration, the shell diameter was reduced to 109 inches while maintaining the overall boiler length. This augmented tube 2-pass boiler can be an extremely useful tool for confined boiler rooms.

Another useful application of an augmented tube 2-pass boiler is the "Ohio Special." The state of Ohio has a law that requires a licensed boiler operator be present when firing any boiler with a heating surface over 360 ft^2. To eliminate the operating cost associated with several shifts of licensed boiler operators, boilers have been designed that are termed "Ohio Specials." These special boilers have less than 360 ft^2 of boiler heating surface and can be fired up to 200 hp or even 225 hp. Johnston Boiler Company, with the help of Fintube Technologies, designed a firetube boiler that uses the augmented tubes. The boiler was then fired at 225 hp with exceptional results. The stack temperature was lowered by over 120°F compared to standard "Ohio Special" designs; which relates to a 3% increase in overall boiler efficiency (depending on the firing rate of the boiler). The pressure drop was also lowered by 2 to 3 inches of water. With a burner normally used on the standard 3-pass Ohio Special boiler fired at 200 hp, we were able to fire the augmented tube 2-pass Ohio Special boiler at 233 hp. The furnace included in the 2-pass X-ID series Ohio Special design also had a larger diameter to better accommodate the burner flame and the radiative heat transfer.

Conclusions

The following conclusions can be drawn from the data and research that has been mentioned in this paper:

- Good boiler design practices must take into account the operation of the boiler and not simply the heat transfer. Parameters that a good boiler design addresses include:
    - Ample furnace volume must be included to absorb a significant portion of radiative heat transfer and allow the new Low NO_x burner designs to function.
    - Optimized pressure drop across the boiler convective passes. The pressure drop determines the fan size required for the boiler application.
    - Ample steam storage and steam height. The volume of steam and distance from the steam nozzle to the normal water level determine to a very large extent the steam quality and the amount of water that will be carried over into the system.

- Boiler design and optimization programs have been written to determine the performance of firetube boilers. These programs can be applied to analyze a wide variety of boiler scenarios for many different boiler applications extending from simple gas fired systems to complex waste heat applications.

- In-flame gas temperature data for firetube boilers has been obtained. The data follows expected trends and has been very useful in the validation of predictive optimization models. This data has been compared to predicted results from computational fluid dynamic combustion models and good agreement has been found. This data has been used to optimize furnace and heat transfer surfaces for typical firetube boilers.

- Gas temperatures measured at the entrance to the convective tube surfaces provided excellent data that validated the heat transfer submodels.

- Augmented surface tubes have proven to be a valuable resource in the design of firetube boilers for many special applications. The advantages of the augmented tube are that it allows the designer to include larger steam storage and steam height resulting in higher steam quality and rapid load swing handling ability. Using the augmented tube also allows the designer to have a lower overall pressure drop with a boiler efficiency that is still over 81%.

- The augmented tube boiler may be used to reduce the boiler shell diameter and still maintain standard steam volumes, steam heights, and boiler efficiency.

- Augmented surface tubes obviate the guideline of 5 ft^2/hp. For example, boilers rated at 1000hp, and greater, have been designed with efficiencies greater than 81% when operated at 100 psig steam pressure. These units require only 3.3 ft^2 /hp of heating surface.

---

[1] Mechanical Measurements, Beckwith, T.G., Buck, N.L., and Marangoni R.D., Addison Wesley, Reading, MA 1982.
[2] Zeldovich, Ya. B., Acta Physecochem USSR, 21, 557, 1946.
[3] Fundamentals of Heat and Mass Transfer, Incropera, F.P. and DeWitt, D.P., John Wiely & Sons, New York, 1985.
[4] 1998 ASME Boiler & Pressure Vessel Code, The American Society of Mechanical Engineers, New York, New York.