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A MEANS OF AVOIDING SULFUR RECOVERY REACTION FURNACE FIRED TUBE BOILER FAILURES

Mike Porter
Dynamic Analysis, Lawrence, KS, USA
Phone 785-843-3558
Email: mike@dynamicanalysis.com

Dennis Martens
Porter McGuffie, Inc, Lawrence, KS, USA
Phone 913-484-3868
Email: martensdh@pm-engr.com

Sean McGuffie
Porter McGuffie, Inc, Lawrence, KS, USA
Phone 785-856-7575 x102
Email: Sean@pm-engr.com

John Wheeler
Motiva Enterprises LLC Convent LA USA
Phone 225-562-6967
Email: John.Wheeler@motivaent.com

ABSTRACT

One of the common causes of premature tube failure in fired tube boilers - technically described as film boiling - is overheating of the tubes caused by steam blanketing. Current literature contains a significant amount of information on this problem, but not much in the way of definitive guidance for avoiding the problem. General "rules of thumb" are available for identifying the heat flux limit required to avoid the problem as in Martens et al [1]. Unfortunately, the values presented by different sources are often in disagreement.

This paper will look at a sulfur recovery unit (SRU) Claus waste heat boiler application and, through the use of Computational Fluid Dynamics (CFD), develop a means of predicting the conditions that lead to steam blanketing and resultant tube failure. Local heat flux conditions at gas side discontinuities (such as the tube inlet ceramic ferrule terminations) combined with associated local water side steam entrainment, and steam generation with coupled velocity effects are discussed.

Introduction

Sulfur recovery unit Claus reaction furnace process outlet hot gas (2100-2800 °F) (1149-1538 °C) passes through the tubes of the two pass kettle type boiler. The waste heat from this gas is used to generate 600 psi (41 bar) saturated steam on the shell side of the boiler. Normally, the boiler tube metal is expected to remain below a temperature of approximately 600 °F (316 °C). When the tube metal temperature exceeds 600 °F (316 °C), adverse high temperature sulfidation corrosion usually occurs in the tubes. If the tubes reach temperatures on the order of 1200-1400 °F (649-760 °C), short-term high temperature creep can cause the tubes to dimple and in a short time, a total collapse type failure of the tube may occur.

The subject boiler had suffered tube damage that was considered to be due to a high temperature condition. The purpose of the analysis reported in this paper was to determine the circumstances leading to the damage and how to avoid similar damage in the future.

This paper examines the circumstances under which the subject boiler tubes can reach elevated temperatures, and develops limiting operational parameters for avoiding the situation. The authors caution that although both kettle type boilers and induced flow separate steam drum type boilers are used in this service, the paper addresses only the specific set of geometries for a kettle boiler. Additionally, the authors caution that the information presented herein is for a SPECIFIC boiler geometry and process flow conditions. The presented information should only be construed as a general guideline for approaching the analysis described in this paper. The conclusions presented should not be used for other applications without performing an application-specific analysis.

A great deal of research has been undertaken by the nuclear industry in order to understand factors affecting the transition to film boiling and the quantitative shape of the curve once the maximum flux has been reached

and departure from nucleate boiling occurs. Exhaustive experimentation has been conducted, and at least one commercial CFD code [2] has implemented routines that allow for the phenomena to be accurately modeled.

Unfortunately, not much of this work has application to the problem under examination. The nuclear industry has primarily been interested in steam blanketing occurring on fuel rod bundles. In nuclear applications, the heat transfer surfaces (rods) are vertical with parallel, forced flow. Typical flow velocities are on the order of 6.6-9.8 ft/sec (2-3 m/s). The subject SRU waste heat boiler uses horizontal tube heat transfer surfaces and convectively/buoyancy-driven water flow. The expected velocities for the convectively/buoyancy driven flow is significantly less than 6.6-9.8 ft/s (2-3 m/s). The CFD code modifications developed for nuclear applications are simply not applicable for analysis of this boiler configuration.

A relatively extensive literature review was undertaken for this project. Unfortunately, few of the more than 40 papers reviewed contained information that was applicable to the problem at hand. Those papers with applicable information are referenced.

Problem description

Figure 1 illustrates the subject waste heat kettle boiler. In the unit under discussion, the hot gas is produced by an SRU Clause reaction furnace. After passing through the boiler, the gas goes on for further processing in the SRU.

In this kettle type of a boiler, sufficient makeup feedwater is introduced at the bottom of the unit. This water matches the steam extracted at the top and maintains a nearly constant water level within the boiler. The water flow on the shell side of the unit is purely buoyancy-driven. There is no induced flow.

The analysis input operational parameters for hot gas coming to the boiler were developed from three years of operational data. Four typical operating parameter sets were selected to represent the various averages and maximum operation of the SRU. One additional parameter set was included to represent a 10% mass flow increase above the maximum parameter set to investigate a postulated abnormal condition.

The operational parameter sets' inlet gas temperature ranged from approximated 2400 °F (1316 °C) to 2600 °F (1427 °C) depending on the process variations (such as SRU feed acid gas composition and use of oxygen enriched burner capability). Parenthetically, the typical measurement error for the temperature measurement of the gas is on the order of 100-150 °F (56-83 °C) during furnace operation. Thus, the actual gas temperature could conceivably be as high as 2750 °F (1510 °C), although this deviation was not included in the analysis data sets. The operational parameter sets' inlet mass flow varied by approximately ± 20%.

The tubes in this boiler were SA -179, which is a typical type of carbon steel for this service. At temperatures exceeding approximately 1100 °F (593 °C), high temperature creep can be expected in this material due to the compressive stress from the external (steam) pressure. This is exactly the type of failure that was observed in the subject boiler. Figure 2 illustrates a portion of a tube from just downstream of the ferrule termination. The tube has been sliced in half in the longitudinal direction along the vertical axis. The "dimple" on the top of the tube was caused by a high temperature creep failure due to excessive tube temperature. Although not clearly visible in this picture, there has also been considerable corrosion of the inside of the tube downstream of the dimple. As previously mentioned, this corrosion is indicative of high operating temperatures at this location.

The purpose of the analysis described in this paper was to determine the cause for the excessive temperatures and to develop a means of avoiding excessive tube wall temperatures.

Problem Physics

Provided that the outside of the tube is covered with the 600 psi (41 bar), 489 °F (254 °C) saturation temperature boiler water, the tube temperature can be expected to remain below ~600 °F (316 °C). In order for the tube temperature to rise above 600 °F (316 °C), the water needs to be replaced by a steam blanket. Once the steam blanket forms, the heat transfer coefficient on the shell side is significantly reduced, allowing for the tube temperature to rise significantly. Figure 3 (used by permission) is an illustration taken from a white paper by Walker [3]. It illustrates the relationship between the heat flux (on the vertical axis) and the temperature difference between the heating surface and the bulk fluid temperature, also known as the wall superheat. This differential temperature parameter is not to be confused with vapor superheat.

The shape of the curve demonstrates typical heat transfer performance through full film blanketing, a phenomena known as the Leidenfrost effect. Starting at the lower left of the graph, the heat flux increases with the differential temperature between the surface and the water. The heat flux reaches a maximum at some differential temperature and as the temperature differential increases, the heat flux actually drops until reaching a minimum flux. In some cases the heat flux can approach zero. This phenomenon explains the ability of Dr. Jearl Walker, a well know educator at Cleveland State University, to reach in and touch the bottom of a crucible of molten lead without getting burned! Figure 4 (used by permission), from Lienhard et al [4] illustrates the same phenomena with the various boiling regimes identified.

As shown in the illustrations in Figures 3 and 4, the boiling process transitions from that of nucleate boiling to complete steam blanketing as the differential temperature increases. Note that the temperature scale on the graph is logarithmic. A tube temperature in excess of 1100 °F (593 °C) would appear possible if we reach film boiling range on the graph in Figure 4. The question that needs to be answered is what are the conditions that lead to this departure from nucleate boiling or, as it is commonly referred to, steam blanketing for this particular boiler? As indicated by the dashed lines between the maximum heat flux point and the minimum heat flux point in Figure 4,the exact relationship between wall superheat temperature and heat flux is not quantitatively defined. .

Solving the Problem

In order to develop a model for the heat transfer process, it was necessary to examine the factors affecting the process. From Figures 3 and 4 and the work of Reisch [5], it can be determined that there are three factors that are somehow related: heat flux, wall superheat and the local quality of the steam (mass fraction of the vapor in the liquid as measured on a cell by cell basis). The initial portion of the curve is covered by the normal relationships between surface temperature, water temperature and the heat transfer coefficient at the water-to-metal interface. This process is accurately modeled by most commercial CFD codes.

Once steam blankets form, however, the heat transfer coefficient becomes greatly reduced in the region labled "transition boiling." While the ability to track the phase change from water to steam is adequately predicted by some commercial CFD codes, the change in heat transfer rate is, in general, not well modeled. As previously mentioned, custom routines that allow for the prediction of the boiling behavior during the departure from nucleate boiling on nuclear fuel rod bundles have been developed by at least one software company. As these routines rely on relations developed through significant experimentation on fuel rod bundle geometry, they are not directly applicable to the problem at hand – horizontally oriented tube heat transfer surface with low speed, convectively/buoyancy-driven water flow.

Specifically, most depictions of the heat transfer characteristics curve show the heat flux approaching zero as the wall superheat increases above the temperature of maximum heat flux as is illustrated in Figure 4. Note that the curve shape between the point of maximum and minimum heat flux, labeled as the "transition boiling" region is represented by a set of dashed lines. This reflects the fact that what occurs in the transition region is not well documented and tends to show extreme variation based on factors such as operating pressure, Merte and Suryanarayana [6]. Neither, for that matter, are the maximum and minimum heat flux values from various experimental sources in good (or even fair for that matter) agreement. The temperature differential (Wall Superheat on Figure 4) at which the maximum heat flux occurs seems to be in the 36-72 °F (20-40 °C) range from most of the sources. Likewise, most sources seem to indicate that the heat flux climbs back to the previous maximum value when the temperature differential reaches approximately 1832 °F (1000 °C). Additionally, work done for the nuclear industry by Reisch [5] suggests that once significant steam is present in the water (described as the void fraction of the water), at the heat transfer interface exceeds approximately 90%, the boiling regime can be taken as having fully transitioned from nucleate to film boiling. Using these points, the relationship depicted by the green and red lines in Figure 5 was developed related to tube wall temperature.

The green line in Figure 5 represents the relationship between the temperature of the tube wall in the 489 °F (254 °C) water and the heat flux during the nucleate boiling regime. In the analysis, this was explicitly calculated by the CFD software. The red line represents the heat transfer versus wall temperature relationship after a departure from nucleate boiling has occurred at the maximum heat flux. Once the tube temperature reached the departure temperature and the local void fraction exceeded 90%, departure was assumed to occur and the heat transfer rate was reduced accordingly. This was accomplished using a user-defined function that modified the external flux on the tubes accordingly when the aforementioned conditions were reached. The black line represents the heat flux on the inside of the tube. This is calculated by assuming a constant bulk gas temperature and a constant heat transfer coefficient; this heat flux is then modified to correct for radiative effects as the internal tube temperature increases. The maximum tube temperature, and a stabilization of the heat transfer regime only occurs when the inside and outside heat fluxes equalize. The dotted red lines indicate +/- 20% bounds on the assumed curve shape in the region above the minimum heat flux point. These bounds were used to ensure that the function used to represent the modified heat flux would return reasonable values within the limits of the available data. The intersection of the black line with the red lines in the 1300 to 1400 °F (704 to 760 °C) range is a possible stable operational point. However at this temperature, the carbon steel tube material is subject to a short term high temperature creep failure condition.

It is important to note that no heat flux values are shown on Figure 5. These were not used as an input for the calculation process. The heat flux was computed as a function of the other variables. The authors also noted that the condition of scaling on the water side will affect the tube wall temperature. A minimal scale was included in the analysis. The literature indicates that the

scale has an impact on the critical heat flux similar to the tube roughness effect; however there is no available information to quantify a scale impact. Consequently, a possible scale effect on steam blanketing could not be addressed.

Figure 6 illustrates the model used to compute the peak tube temperatures within the two pass kettle boiler. The grid densities were established based on a number of parametric studies of the various regions in the overall model. These studies were conducted to achieve an acceptable level of the discretization error. The highest grid density was required near the tubes. This model consisted of approximately 4.3 million cells.

Since the boiling process is inherently variable, it was necessary to conduct a transient analysis. A 50 second (model time) analysis required approximately 5,000-6,000 processor hours to complete. The red section is the fluid surrounding the first pass tube field, which required a higher resolution mesh to accurately capture the near wall flow effects. Not visible in this image due to the zoom level are the tubes that were modeled explicitly. The orange section on the right is the second pass tube field with the tubes represented by a porous media approximation to model the water flow resistance that the tubes provide. The green region at the bottom is where the makeup water was introduced and the light blue section on top is the steam extraction region. The darker blue region is the bulk water/steam region which did not require the same level of mesh discretization as the mesh surrounding the tubes. The model extended from the tube sheet approximately 10 tube diameters past the end of the tube ceramic ferrules. Prior to running this model, a model of a single tube representing both passes of the bundle was used to verify the computed heat transfer rates. This model was also used to calculate the internal heat flux on the first pass of the tubes. This flux was applied to the entire tube field through the use of tabular field data. No heat transfer was defined on the second tube pass, as very little heat transfer occurs at the end of the second pass, which is located at the inlet end of the first pass on the subject boiler.

Figures 7 and 8 illustrate the void fraction computed by the model at two different times (t=25 and t= 28 seconds). As can be seen, the high steam void fraction regions change from one time to the next, as would be expected at the top of a boiling kettle.

The second pass tube field was hidden in these views to speed up the rendering process.

Figure 9 illustrates the water domain void fraction in the lower portion of the tube field just downstream of the end of the ferrules, which is the maximum heat transfer location at a specific time step of the analysis. The red "dots" seen near the top of some of the tubes represent regions where the local steam void fraction exceeds 90%. As can be seen, the pattern is somewhat random, with only a few of the tubes affected.

Figure 10 illustrates the pattern near the top of the bundle at the same time-step as Figure 9 and the same vertical tube row location. In the top portion of the bundle there are more tubes where the void fraction is indicated to be over 90%. As was the case near the bottom (Figure 9), however, the pattern is still random. Also note that high void regions appear on at least a few of the tubes on the bottom or side of the tube. This is due to bubbles separating from the top of one tube and rising to the bottom of the tube above. These regions of high void were not typically stable; as such, they would not be expected to cause operational difficulties. The pattern changes over a short period of time, indicating that in most cases, unstable nucleate boiling was occurring.

Where the high void fractions do remain stable for some period of time, the heat transfer coefficient is dramatically reduced due to film boiling. This temporarily reduces the heat flux at these locations on the outside of the tube. Since the heat flux on the inside of the tube remains nearly constant, the result is an increase in the tube wall temperature. As was indicated on Figure 5, this corresponds to a decreasing heat flux on the water side until the minimum flux condition is met. The reduction of heat transfer with increasing wall superheat causes an unstable feedback loop resulting in an increasing tube wall temperature. Only when the interior and exterior heat fluxes are equal as the external flux rises from the minimum flux condition, does the increase in wall temperature stop and a stable tube wall metal temperature is achieved.

Figure 11 illustrates the temperature profile computed for the tube bundle at the end of 50 model seconds where a quasi steady state solution was reached. Significant portions of the tubes have temperatures in the 1100-1200 °F (593-649 °C) range. In a very few regions scattered throughout the bundle, the computed temperatures reach 1,400 °F (760 °C). Thus, short-term creep failures could be expected in these regions for this operating condition.

Figure 12 illustrates the results of this run and several other runs conducted at other operating conditions and boiler geometries. The isolated data point above the line (point 1) is the result of data set #1for this analysis. The other analyses, points for data sets 2-5, at lower mass flow rates and/or temperatures, did not result in computed tube temperatures that exceeded approximately 800 °F (427 °C). The actual operating data indicated tube failures only at operating conditions similar to point 1. The dark line represents the heat flux on the outside of the tube as indicated on Figure 4. The points were computed using CFD, based on the differing operating conditions.

Based on additional analysis, operating limits for the unit were developed and a derived curve fit was

developed, as illustrated in Figure 13. The black line represents the operating limits derived from the analysis. The blue and red lines represent the limit with a 2.5% and 5% safety factor respectively. As this boiler had experienced tube failures at operations at what were considered to be similar to data sets 2-5 and perhaps less than the postulated data set 1 it was recommended that limiting operating parameters be utilized that address a reasonable safety factor based on operation control variability.

Conclusions

Using CFD, it is possible to develop operating parameter limits that will avoid steam blanketing and subsequent tube failure due to short-term high temperature creep. The operating limits are based on both the mass flux and the temperature on the inside of the tube. It is important to note that these results are highly geometry-dependent for both the process gas and boiler water sides. Changes in the tube sizes, ferrule geometry and/or spacing will result in different operating limits.

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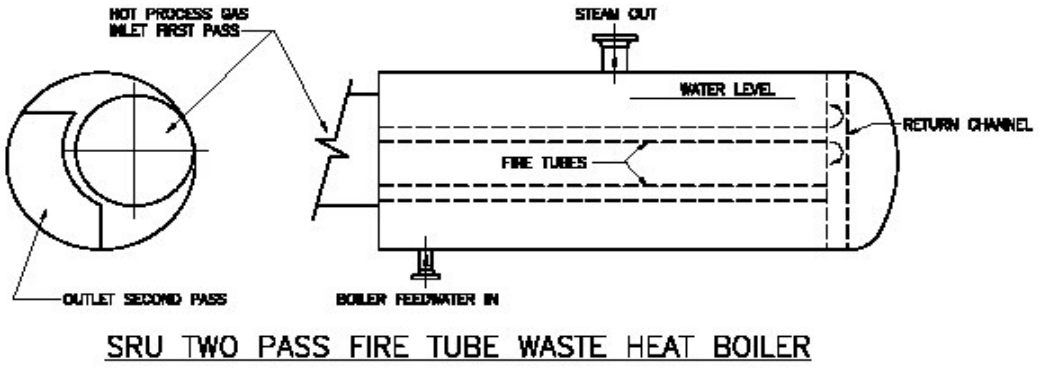

Figure 1 – Fired tube kettle boiler schematic

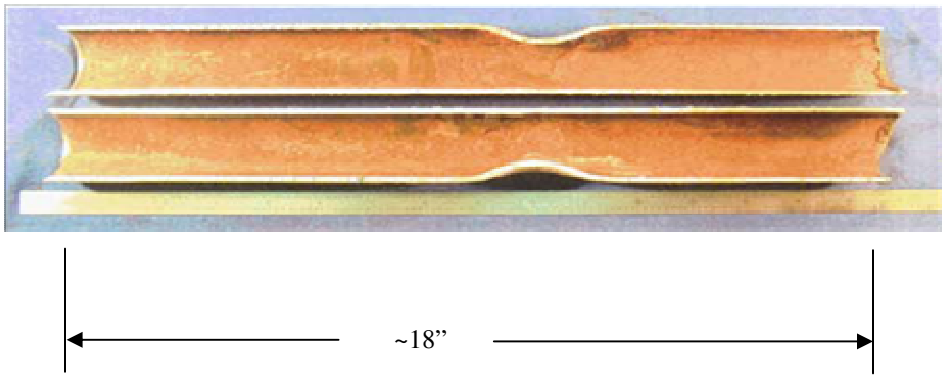

Figure 2 – Image of dimples in tubes caused by creep failure

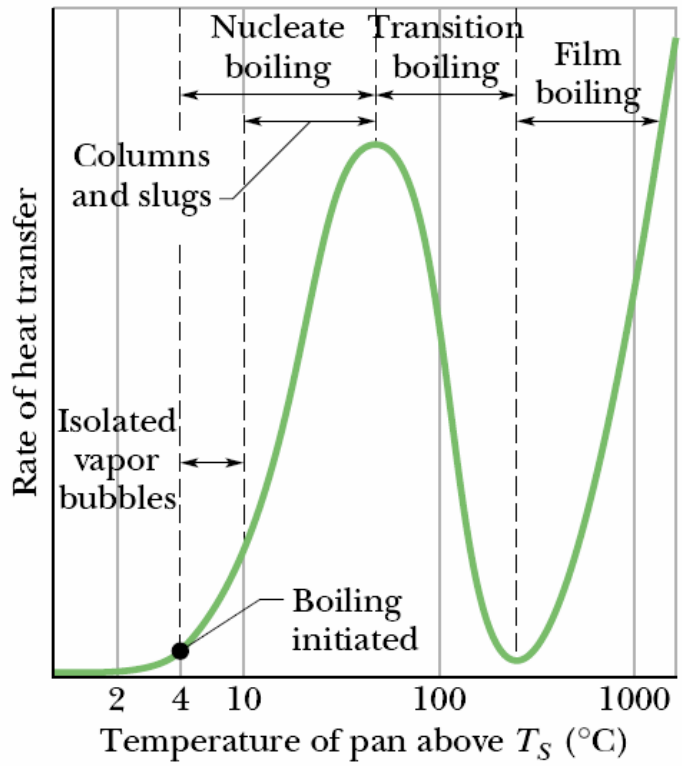

Figure 3 – Schematic of boiling regimes

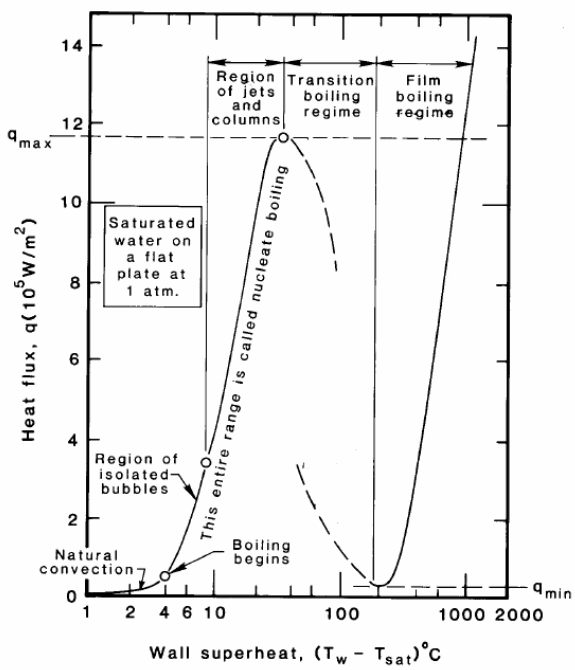

Figure 4 – Heat transfer characteristics for a specific flat plate with water at 1 atm.

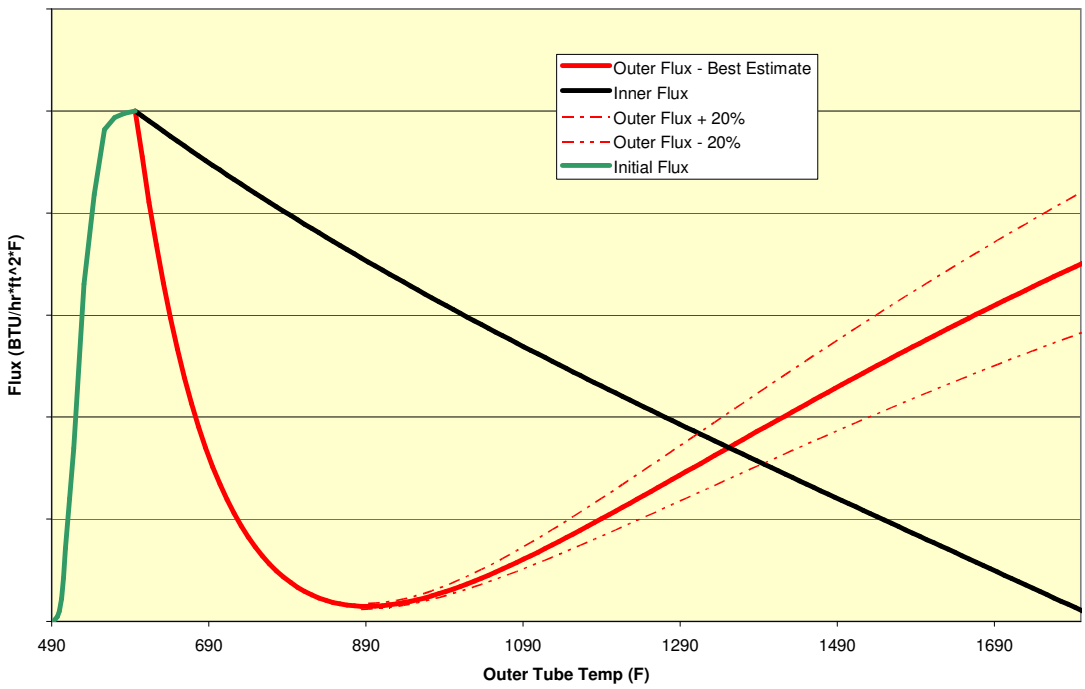

Figure 5 – Representation of the external heat flux versus the tube outer wall temperature

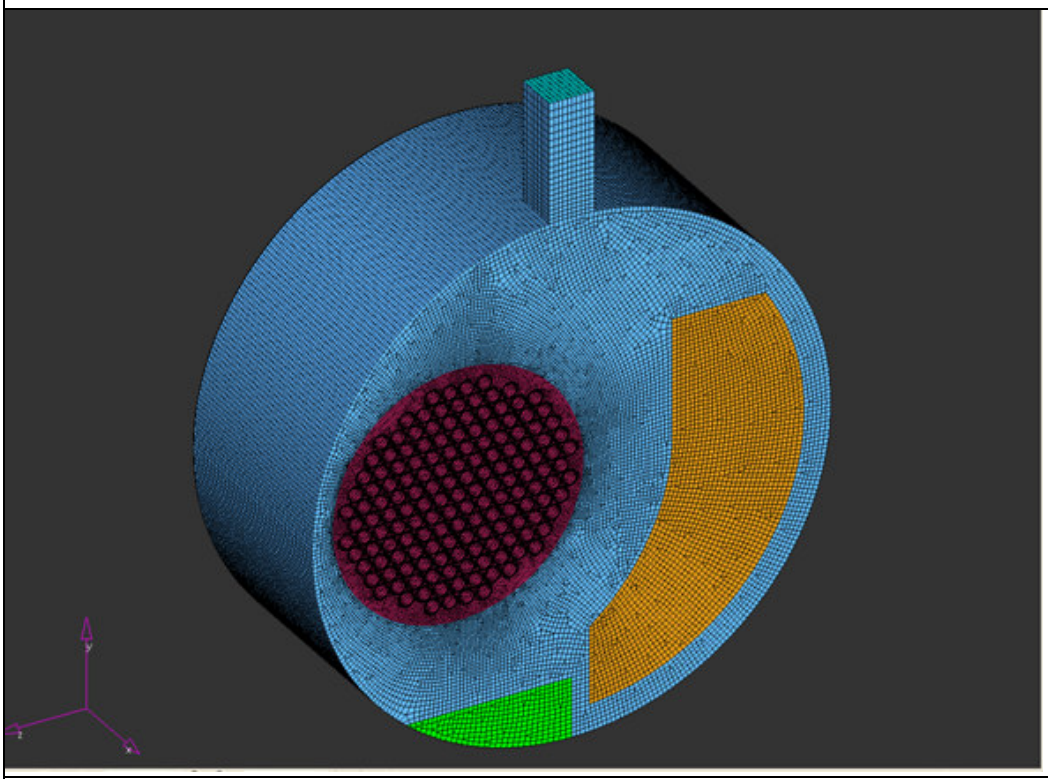

Figure 6 – 3D CFD model used to predict kettle boiler behavior

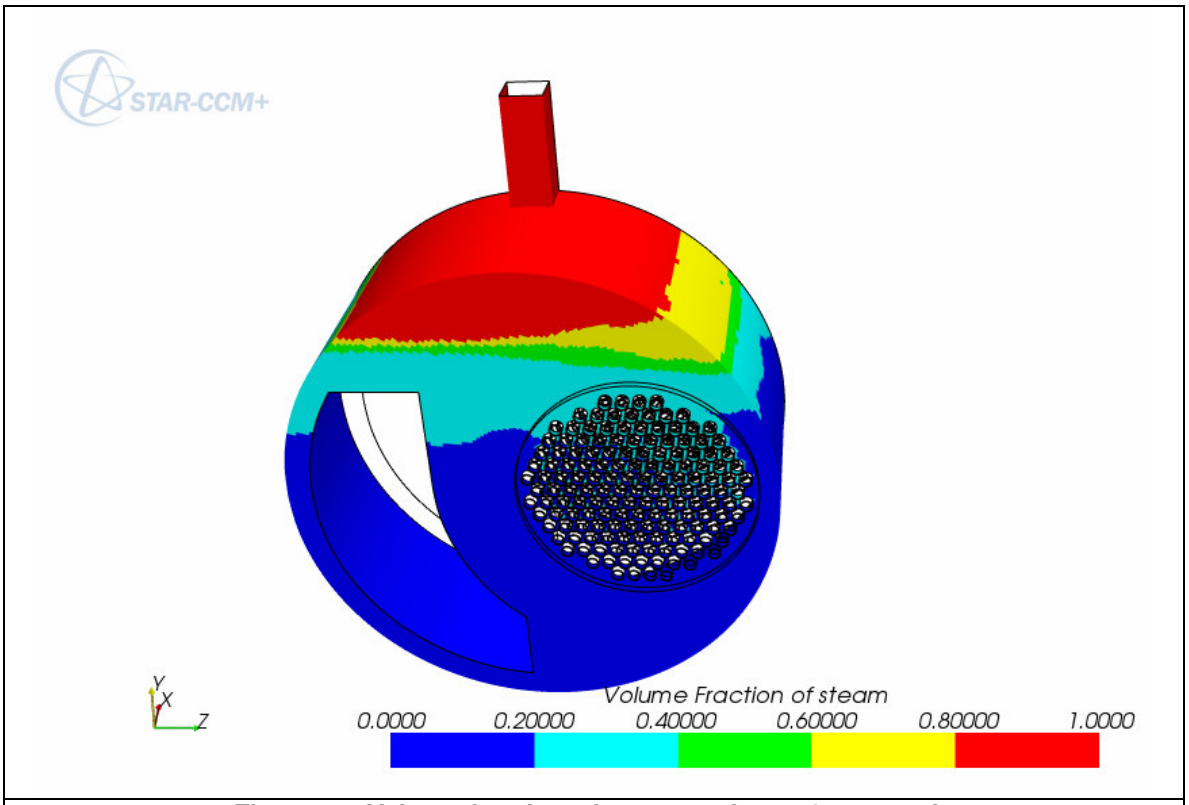

Figure 7 – Volume fraction of steam at time t=25 seconds

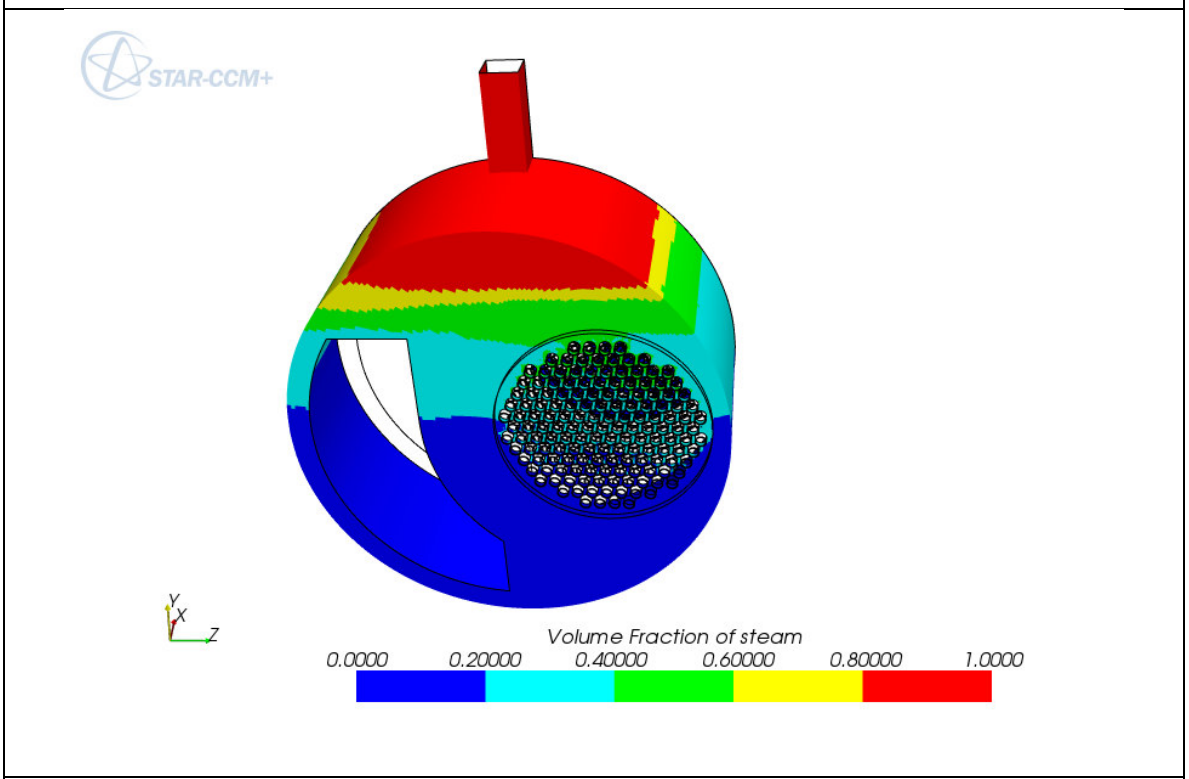

Figure 8 – Volume fraction of steam at time t=28 seconds

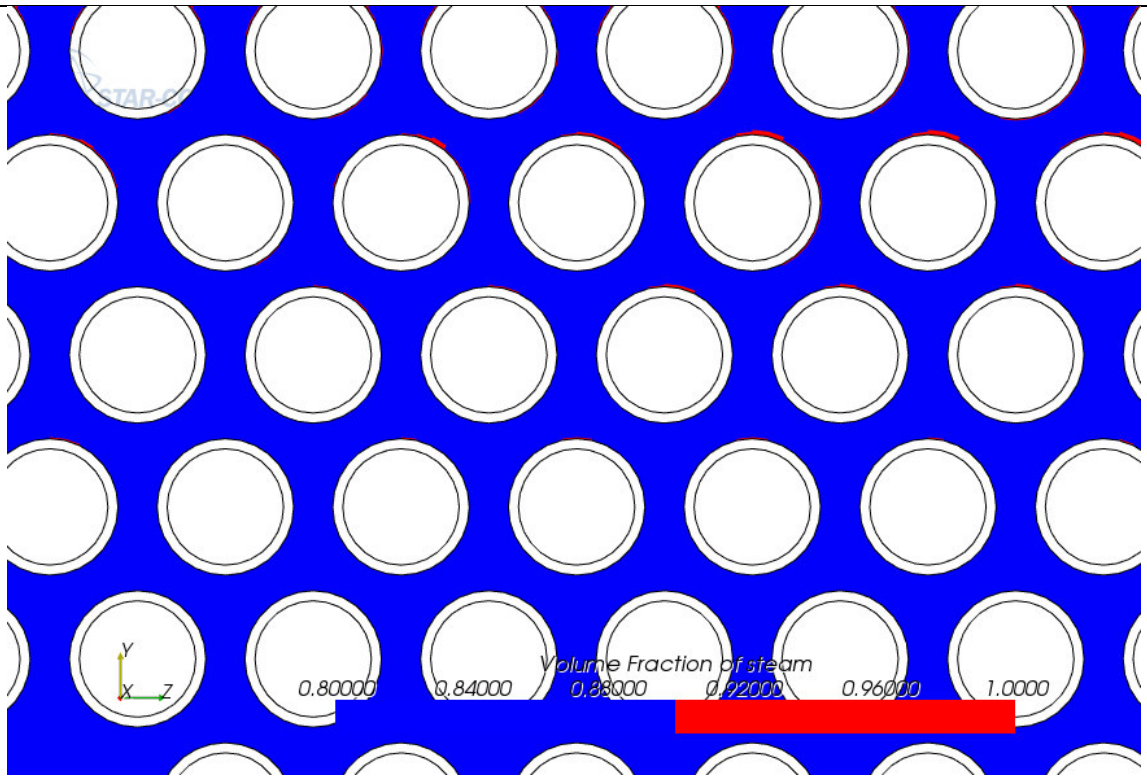

Figure 9 – Lower tube field highlighting areas with high local steam volume fractions

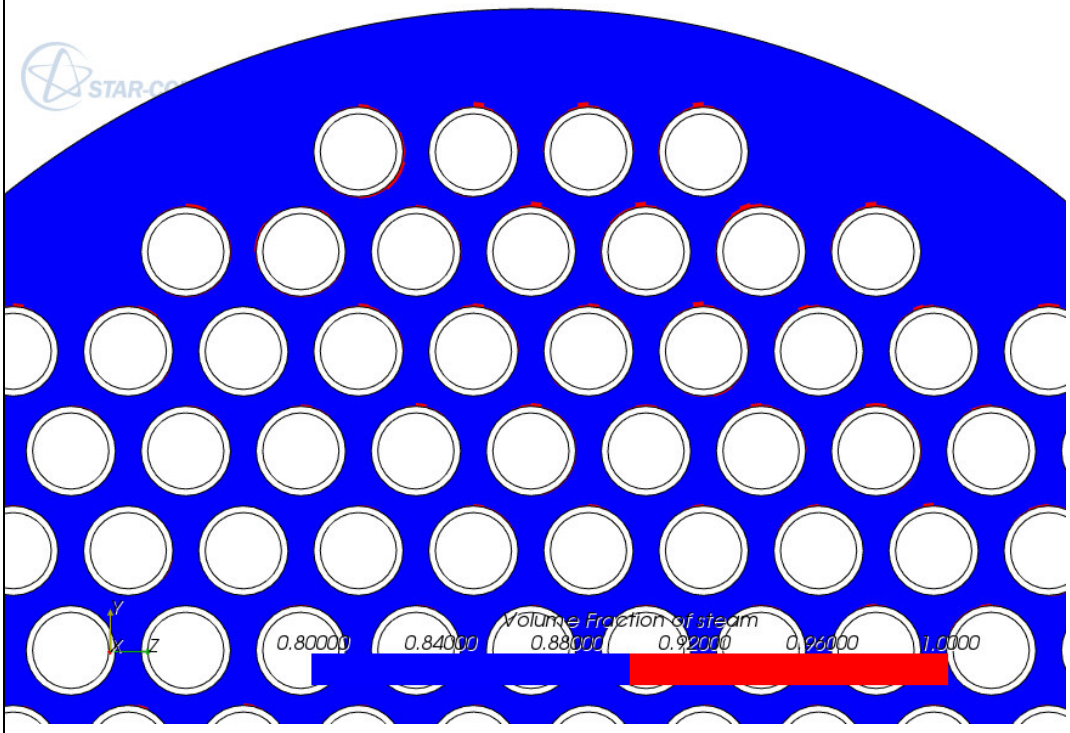

Figure 10 – Upper tube field highlighting areas with high local steam volume fractions

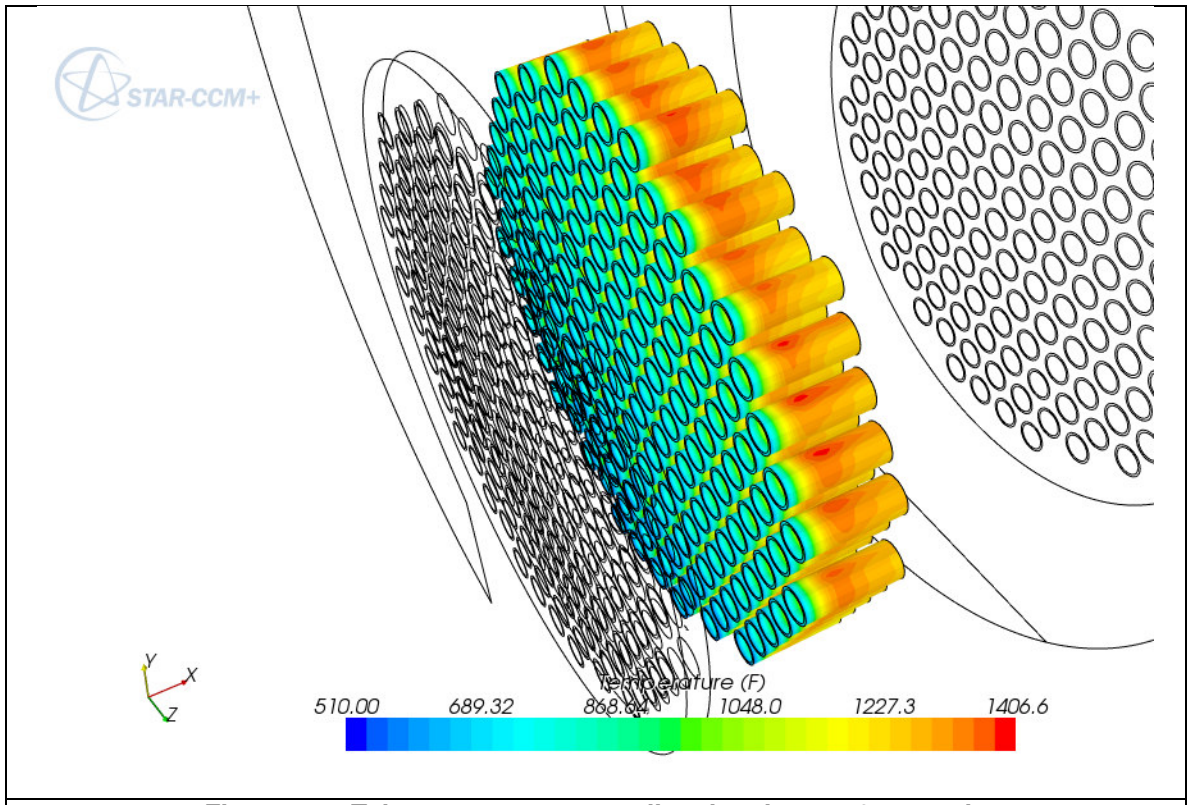

Figure 11 – Tube temperatures predicted at time t=50 seconds

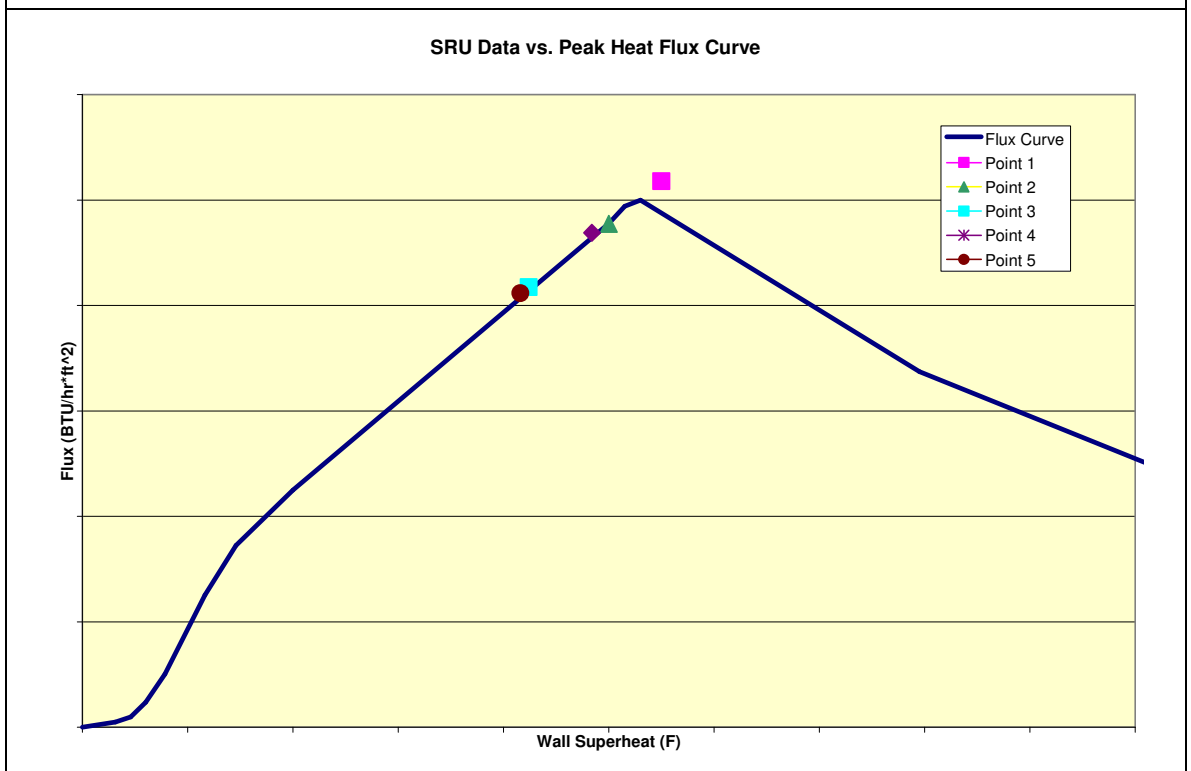

Figure 12 – Process conditions analyzed versus typical heat transfer curve

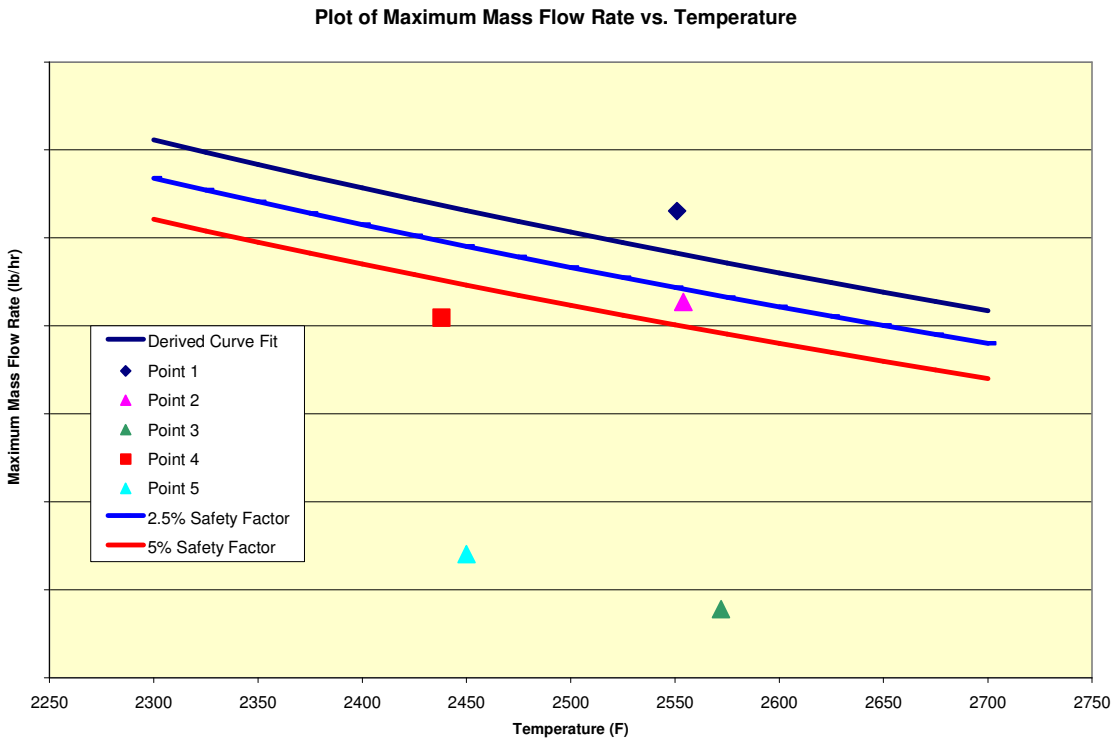

Figure 13 – Mass flow rate versus temperature for boiler under consideration