To the Graduate Council:

I am submitting herewith a thesis written by Tiffany Grant entitled "An Examination of Boiler Failures and Associated Design Issues." I have examined the final paper copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.

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An Examination of Boiler Failures and Associated Design Issues

A Thesis
Presented for the
Master of Science Degree
The University of Tennessee at Chattanooga

Tiffany Sherrell Grant
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DEDICATION

This thesis is dedicated to my parents, Jim and Phyllis Grant, for always giving me the strength to succeed and my fiancé, Bradley Winn, who always told me that I could do anything if I was willing to work hard enough for it.

Acknowledgments

I would like to thank all those who helped me attain my Master of Science degree in Mechanical Engineering. I wish to thank Mr. Wayne Michaelian from Performance & Design Engineering at ALSTOM Power Systems in Chattanooga, Tennessee who helped me throughout this journey. I would also like to thank Dr. James Hiestand for being such a great advisor who was always willing to help me when I needed it. Also, thank you to Dr. Mike Jones, Dr. Gary McDonald and Dr. Ethan Carver for serving on my thesis committee. Lastly, I would like to thank Mr. Paul VanKooten from the materials lab at ALSTOM Power for providing me with great assistance.

Abstract

The purpose of this study was to determine if common boiler failures were mainly due to uncontrollable events or from the effects from selecting lower grade materials and processes for the components. Eighteen case studies of tubing failures were selected and examined from boilers all around the world. Twelve of the tubes failed due to excess hoop stress and six failed from some kind of cracking. Calculations were made and it was determined that seventy-five percent of the failures would not have happened if a higher grade but more expensive material had been used. It was also found that 83.3 percent of the utilities did not perform any metal treatments for the tubes that experienced cracking. Although there was no way to prove that these cracks would not have happened with some sort of stress relieving treatment, the numbers suggest this. Utilities not properly treating their metal components should carefully consider such treatment.

The economizer, superheater and reheater were the boiler components selected for this study. The study was restricted to these three boiler components, although excess hoop stress and cracking failures do occur in other regions of the boiler as well. All three contained tube bend regions, but not all experienced failures in the actual bend of the tube.

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Chapter 1 – Introduction

A. Steam Generator Background Information

The purpose of a steam generator, or boiler using more common terminology, is to produce steam to drive a turbine and ultimately generate electricity. Although there are multiple ways of generating electricity, the cases considered burn coal to heat tubing filled with water. The water is then converted to superheated steam to drive the steam turbine. The turbine then drives the generator to produce electricity. The level of megawatts that the utility wishes to produce determines the amount of steam needed. Although this appears to be a simple procedure, there are many processes that must occur to produce steam of desired quality, temperature and pressure. While there are many components involved in boiler operation, the eighteen cases examined for this study considered only the superheater, reheater and economizer. This was done because all three of these boiler components contain bends and their tubing materials exhibit similar metallurgical properties. Data, such as design pressures, temperatures and materials, were examined to see why each of these cases resulted in failures.

It is important to note that the tubing located inside of a boiler is placed in a very hostile environment during its lifespan. An AutoCAD image was generated to show the environment that the tube is exposed to and is in Figure 1:

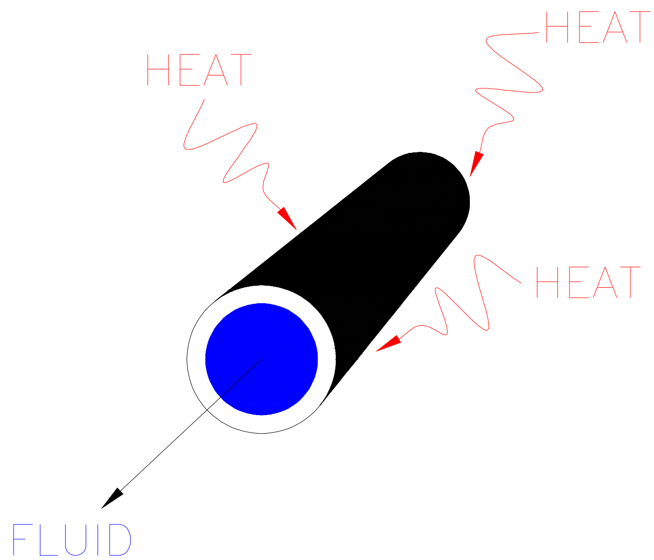

Figure 1: Tubing Placed in a Hostile Environment

This hostile environment includes both extremely high temperatures and internal pressures, so the selection of materials is vital to the success of boiler operation. The temperatures and pressures vary whether the tube is a part of the economizer, reheater or superheater and those values are given in sections B, C and D under Chapter 1. The material selection must not only be correct, but the fabrication and installation of the material itself must also be accurate.

B. The Economizer

The economizer consists of a set of bent tubes, formed into groups called assemblies that are located in the backpass area of the boiler. The assemblies of the economizer are responsible for heating the feedwater that is delivered by the pumps at the beginning of the cycle. These tubes are located where the gas from the furnace passes over and heats them. An illustration can be found in Figure 2:

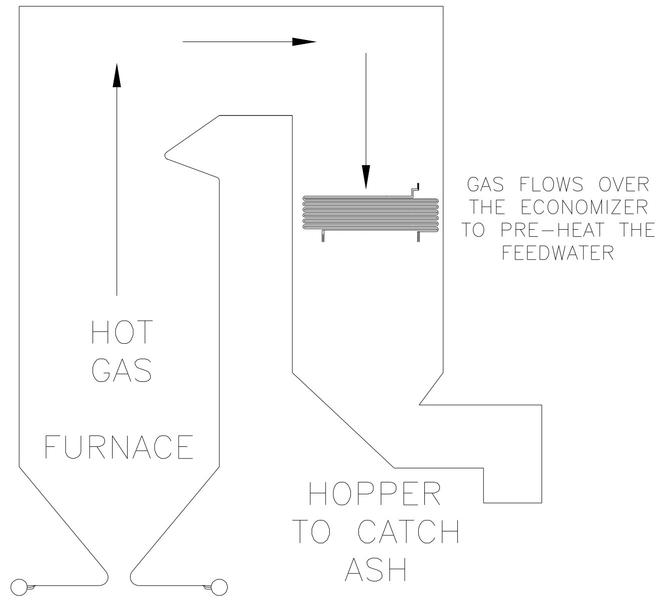

Figure 2: Economizer Location

As the gas travels over the economizer assemblies, the feedwater inside the tubes increases in temperature. This preheats the water before it reaches the waterwalls of the furnace and begins to be converted to steam. By putting already warm water into the furnace waterwalls, it is converted into steam more economically. Typical materials used in economizer assemblies are various carbon steels, such as SA-192 and SA-210. Although the majority of materials used are carbon steels, different classes and grades of carbon steels that range in chemical composition and stress allowables could be selected. It is important to note that even if a stainless steel was requested due to its high allowables and quality, it is a violation of ASME code to use a stainless material in an economizer assembly (ASME Boiler Code, Section 1 PG-5.5). This is because the chlorine in the water can directly attack the nickel that is in the stainless material, thus corroding the tube walls.

Water does go through a treatment process prior to entering the economizer to rid it of impurities. However, this process is imperfect. Rather than risking the tube wall thickness being deteriorated from the chemical reaction of the chlorine and the nickel, the code simply prohibits the usage of stainless steel in the economizer. The economizer assemblies are oriented horizontally and each is located directly behind one another. They are designed to withstand design pressures ranging from 1650 psi to 3040 psi, with actual operating pressures ranging from 1450 psi to 2840 psi. The lower number provided represents the pressure at the inlet portion of the economizer, while the higher number represents the pressure leaving the economizer. Fluid temperatures in the economizer range from 400°F to 650°F, while gas temperatures flowing over the tubing range from 800°F to 1200°F. These temperatures are much lower than that of the superheater and reheater because the economizer is located in the backpass, rather than in the furnace area.

C. The Superheater

The primary purpose of the superheater is to elevate the temperature of the steam to a temperature above saturation level, thus making it superheated. Prior to reaching the superheater, the steam still contains moisture, classifying it as “saturated” steam. After the steam travels through the superheater assemblies, it is converted into dry, superheated steam after being put through intense heat. This is vital because any moisture contained in the steam would damage the blades of the turbine. The superheater’s location can be seen in Figure 3:

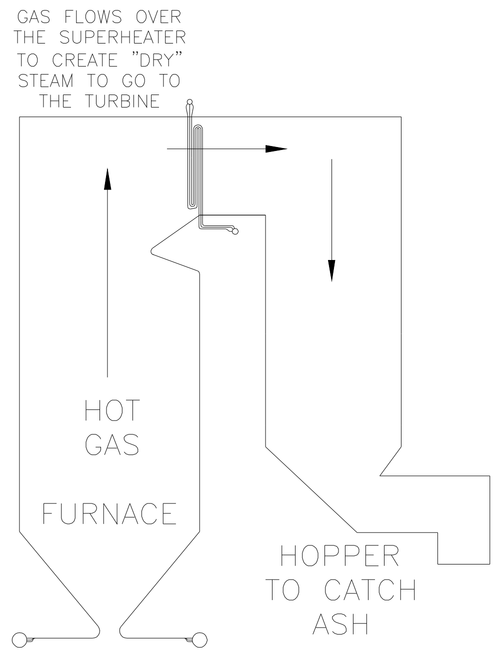

Figure 3: Superheater Location

The superheater assemblies are particularly important in boiler design because they are the final stage the steam goes through before exiting the boiler and going to the turbine. They typically are made for design pressures ranging from 1600 psi to 2990 psi, while operating pressures range from 1400 psi to 2790 psi. Many boiler design companies will choose stainless steel for their superheaters because of its excellent resistance to oxidation of the metal and ability to withstand very high temperatures. However, carbon steels and ferritic alloys in the initial or low temperature superheater assemblies sometimes are selected because of the low cost associated with those metals. Inlet fluid temperatures for the superheater start around 600° F, while outlet temperatures reach up to approximately 1050° F. The gas temperature flowing over the superheater ranges from 1000°F up to 3000°F. Typically, the final stages of superheating occur in assemblies that are located in the furnace over the flame where the highest gas temperatures occur.

D. The Reheater

Once the steam leaves the boiler and goes through the turbine, it loses energy. Temperature and pressure are reduced. The function of the reheater is to elevate these values back to the point that the steam can enter the turbine again for another cycle. By doing this, the efficiency of the boiler is increased. The reheater location is in Figure 4:

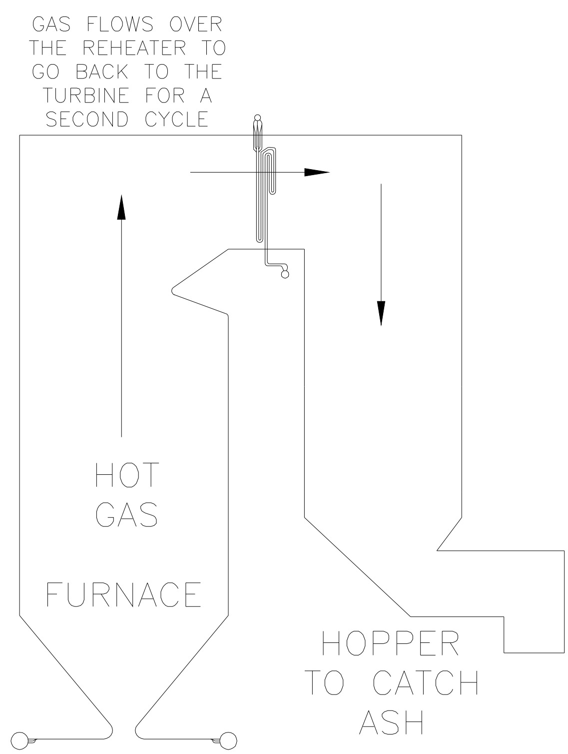

Figure 4: Reheater Location

Reheaters typically do not contain stainless steel due to the high cost of the material. Common materials for reheater units are various types of ferritic alloys. The design pressure for the reheater ranges from 500 psi to 900 psi, depending on the output power of the utility. The operating pressure of the reheater is from 300 psi up to about 700 psi. The inlet fluid temperature is approximately 450°F, while the outlet temperature can reach up to 1050°F. The temperature of the gas going over the reheater assemblies ranges from 1000°F to 3000°F. Whether the gas temperature flowing over the tubes is hotter for the reheater or superheater depends on the orientation of the boiler. Some boilers have the reheaters closer to the direct furnace gases, while others have the superheater closer. All of the orientations of the boiler components vary depending on how much power the utility wishes to produce.

E. Design Pressure

The design pressure is a key component in boiler design because the closer the operating pressure is to the pressure the boiler was designed to withstand, the greater risk of a tubing failure. However, the rupturing of the tubing is not the only potential damage that the boiler could experience due to inappropriate operating pressure levels. The design pressure can be thought of as a "worst-case" pressure for the boiler to be operating at where no harm can be done. It is specified by the boiler designer and is used to determine the minimum wall thickness of the tubing used for varying components of the steam generating system.

The design pressure should always be greater than the actual operating pressure of the boiler because of the need for a safety factor. If the operating pressure exceeds the pressure the boiler was designed to withstand, yield values could be exceeded, resulting in tube rupture. It is important to note that the design pressure is not at a constant value throughout the boiler. The pressure varies for different portions of the boiler. Safety factors also vary for different locations by ASME requirement.

Monitoring pressure is vital because the ultimate function of a boiler is to produce power. The turbines, which are responsible for generating the power, are attached to the boiler and are very sensitive pieces of equipment that operate at a specific temperature and pressure. If the operating pressure of the boiler becomes greater than the design pressure, it can damage the blades of the turbine. Also, the boiler is designed to generate a specified amount of power. The design pressure is determined for a specified power level. If the pressure is below this value, less power is produced, causing a loss of money for the utility.

One way that designers have learned to help control the design pressure is by installing safety valves prior to the turbine. Their primary function is to constantly monitor the output pressure that will be going to the turbine. The safety valve is set to a level that is no greater than the design pressure, but is typically greater than operating pressure. Once the pressure approaches the design level, the safety valve opens and releases steam into the atmosphere to reduce the system pressure. It is important to note that this pressure drop affects all components of the boiler since it is operating in the same circuit.

Design pressure also has a direct effect on the level of hoop stress that the tube experiences. Hoop stress is defined as the stress from inside of the tube that occurs in the circumferential direction. These stresses stretch the walls of the tubing, due to the internal pressurization.

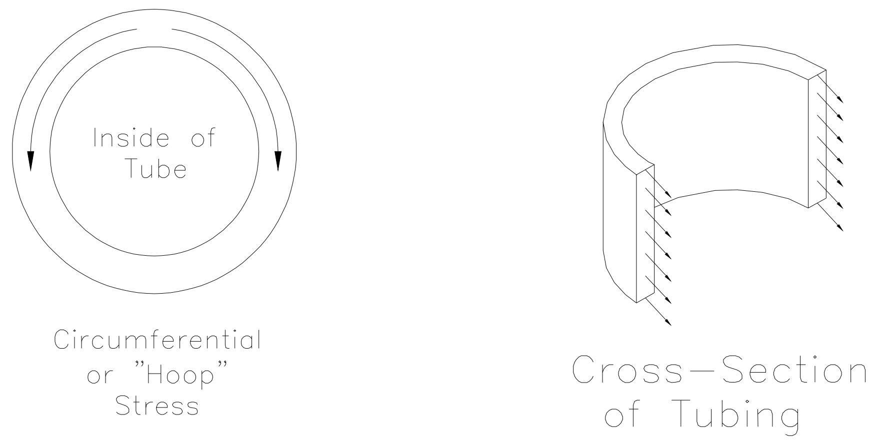

Figure 5: Hoop Stress Illustration

This enlarges the tube and if the value exceeds the allowable, then the tube will burst. Once the tube bursts, it is called a “fish-mouth” rupture. An illustration of this kind of damage can be seen in Figure 6, where F represents force:

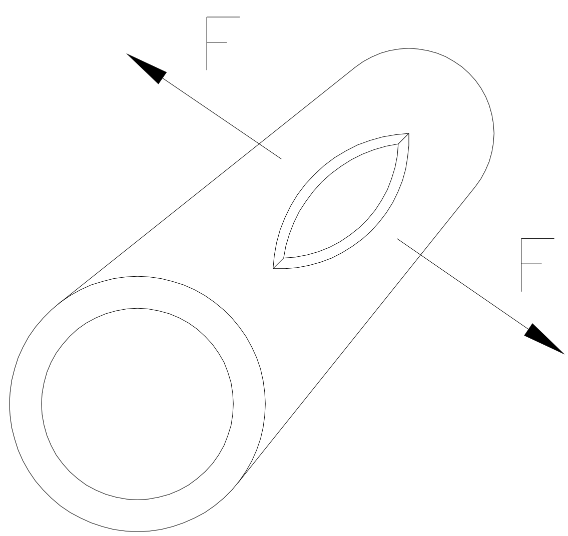

Figure 6: Fish-Mouth Failure Illustration

F. Design Temperature

The design temperature is the temperature that the materials in a specific section of the boiler can withstand. As with the design pressures, there is a safety factor put into the design temperature, so it is higher than the actual operating temperature. The design temperatures were found using analysis letters from each utility site and have a direct effect on the stress allowable per material. For example, the allowable stress value for a design temperature of 1000° F for SA-213 TP304H material is 14,000 psi. However, if the temperature is elevated by just 50°F the material can now withstand only 12,400 psi. If the temperature is again raised by 50°F to 1100° F, the allowable stress is lowered to 9,800 psi. This corresponds to a thirty percent loss in sustainable stress with only a 100° F increase in temperature. Table 1 illustrates these trends:

Table 1: Allowable Stress Values for TP304H

Temperature (Degrees F)	Allowable Stress (psi)	% Loss From Original Stress Value
1000	14,000	-
1025	13,200	5.71%
1050	12,400	11.43%
1075	11,100	20.71%
1100	9,800	30.00%
1125	8,750	37.50%
1150	7,700	45.00%

As can be seen in Table 1 (*ASME Boiler Code for Carbon, Alloy and Stainless Steels*. Section Two Part D), the SA-210 TP304H material has lost almost half its allowable stress value with an increase of only 150° F in temperature. A similar comparison was also done with the TP304H material's upgraded option, SA-213 TP347H. This TP347H is also a stainless steel material, but is a more expensive option. Using the same temperatures as the TP304H material, a similar chart for TP347 is in Table 2:

Table 2: Allowable Stress Values for TP347H

Temperature (degrees F)	Allowable Stress (psi)	% Loss From Original Stress Value
1000	16,400	-
1025	16,300	0.61%
1050	16,200	1.22%
1075	15,150	7.62%
1100	14,100	14.02%
1125	12,300	25.00%
1150	10,500	35.98%

Table 2 (*ASME Boiler Code for Carbon, Alloy and Stainless Steels*. Section Two Part D) perfectly illustrated the entire point of this study. The upgraded material only sustained a thirty-five percent loss from its original value with a 150° F increase in temperature, rather than the forty-five percent loss of the TP304H material. For a 50° F increase in temperature, there was an 11.43 percent loss with the TP304H material, while the TP347H suffered only a 1.22 percent loss. It is also important to note how much higher the stress values are for the upgraded material for the same initial temperature. The upgraded material is able to take much greater levels of heat, and can sustain its original strength for a longer period of time. This is one example of why the proper selection of tubing is so important for boiler design. More expensive materials have higher allowable values and can survive a more hostile environment. It is important to

note that only the material and the corresponding design temperatures determine the stress allowables.

G. Common Boiler Problems and Issues

Ash can damage a tube from the outside through collisions with the metal. The gas velocity inside of the furnace for a coal-fired unit is approximately 60 ft/sec. The ash moves with the gas. There is also alpha quartz in the gas/ash mixture, which is basically a very hard ash particle. These particles can hit the tubes very violently, causing the tube to deteriorate. As the tube deteriorates, the thickness of the tube wall is decreased, which increases the level of hoop stress in the tube. The circumferential stress will eventually exceed the allowable, which will result in a failure. Many tubes in this study experienced this problem.

There can be a reduction in fluid flow from internal deposits, which results in a higher tube metal temperature. This puts the tube in danger of exceeding the now lower allowable and eventually failing. The internal deposits decrease the inner area that the fluid can flow through, which in turn decreases the flow rate through the affected tube. The internal deposits also reduce film conductances and affects thermal-conductivity, resulting again in an increase in metal temperature and a decrease in allowable stress. Although this is a common problem in boilers, none of the eighteen case studies experienced this issue.

Corrosive elements in the fly ash can build up on the surface of the tube and begin a chemical reaction that will reduce the tube wall thickness, which increases the level of hoop stress. Again, although this is a common problem in boilers, none of the eighteen cases selected experienced this problem.

H. Problem Definition

A reoccurring theme was found when looking at the case studies selected. The incidences where tubes failed due to high levels of circumferential stress contained metals that were of a lower grade than is often recommended by boiler designers. Also, there was a high incidence of cracking in the bends of the tubes. The majority of these cases were found to have not had extra care put into the tubing material after the bending process. If companies make these choices to reduce initial costs, these types of failures will continue to occur. These repetitive failures affect not only the utility with greater costs, but they also affect the rest of the country's population who have to pay for electricity at a higher rate due to unplanned boiler outages.

I. Problem Objective

The ultimate goal of this paper is to determine whether the failures were due to choosing lower grade tubing materials and avoiding expensive metal treatments or if a rupture would have occurred even if the more expensive option had been chosen. This theory was tested using basic engineering concepts, theories and calculations so a recommendation could be made to help prevent the failures from happening again in the future. The design temperatures and pressures were found through the engineering data sheets and from contacts at each utility site. The goal of this paper was completed by:

1. Examining all of the data found to calculate the levels of hoop stress for all fish-mouth failures both before and after the rupture occurred.

2. Examining all of the data for the failures due to cracking to determine if any heating or stress relief process was applied to the tubing prior to it being installed into the boiler unit.
3. Combining all of the data to determine if the failures could have been prevented if the higher-grade materials and costly metal treatments had been chosen by the utility.

Chapter 2 – Methodology

A. Data Collection

The case studies were found with the help of Paul VanKooten of the metallurgical lab at ALSTOM Power in Chattanooga, Tennessee. Originally, twenty-six cases were chosen and each was numbered. However, they were eventually narrowed down to eighteen. The eighteen studies were chosen from hundreds of options and only the component damaged was looked at prior to reading each case. Neither the location nor utility was known until after the cases had been selected. Also, the modes of failures were not known until after the eighteen had been chosen. The boilers were built by ALSTOM, Babcock & Wilcox and Foster/Wheeler. Because the data are proprietary, the names of the actual utilities are not listed in this study. It is important to note that aspects of the boilers, such as selected materials and metal processes, are ultimately up to the utilities. The final decisions made for materials and treatments were not made by the boiler designers.

B. Project Population

All of the eighteen boilers are coal-fired units. Their geographic locations vary, but all are located in the United States, with the exception of one case that is located in India.

C. Limitations of The Study – Assumptions Made

1. It was assumed that there was a uniform wastage rate around the tube. Although this is unlikely, there is no way to know the exact thickness at every point of the inside wall of the tube without access to many more data and advanced modeling.
2. It has been noted that there is a difference between design and operating pressures. The design pressures must be used to conduct the calculations for this project since there is no way to determine the exact operating pressure at a particular point in a boiler the moment that rupture occurred. It is important to remember that design pressure is used as a worst-case pressure; therefore, it will be higher than the operating pressure.
3. Likewise, design temperatures were used to conduct this study since there is also no way to determine the exact temperature at one specific point in the boiler at the time of failure. Since allowable stress is a direct function of the temperature, it was a very important factor in this study.

D. Summary

No utility company wants to see a failure because of the obvious cost of repairing the damage. However, it is not only the cost of repair that must be considered when analyzing monetary data. The utility is also faced with huge loses in revenue due to unplanned outages because they are not making power when the plant is down. It can cost the utility up to one million dollars per day to suddenly shut down. Also, many of the components in the boiler are not easily accessible. In some cases, the utility must first put up scaffolding so people can get around the inside of the boiler, which takes time. An engineering firm is also typically hired if the damage is severe enough, which can be a huge expense to the utility. Although there are some "quick fixes" to various problems, they typically lead to larger problems if they are not properly addressed. For example, if a tube leaks, many utilities will simply plug the tube so no fluid passes through it. Even though this will temporarily fix the problem, a basic engineering situation arises. During normal applications, while hot gases are flowing over the tubes from the outside, the tube metal is being cooled from the fluid on the inside. Since "cold" fluid is no longer in the tube to help cool the material, the temperature of the tube metal itself is rapidly elevated. As temperature rises, the allowable stress for that material decreases. Once the allowable stress level is exceeded, the tube will rupture. In some cases, one tube failure can cause nearby tubes to fail as well. Now a huge problem exists for the utility where an extended outage will be necessary that resulted from one tube leak. The primary intent of this study is to demonstrate that material selection may prevent common types of tube failure.

Chapter 3 –Calculations

A. Determining Design Pressures

Table 3 shows the design pressures and damaged components in the eighteen cases considered.

Table 3: Damaged Components and Corresponding Design Pressures

Utility Number	Damaged Components	Design Pressure (psi)
1	Superheater	2,650
2	Superheater	2,925
3	Economizer	2,700
4	Superheater	2,525
6	Superheater	2,990
8	Reheater	700
9	Superheater	2,525
10	Superheater	2,950
11	Superheater	2,950
12	Reheater	750
13	Reheater	1,056
16	Economizer	2,800
17	Economizer	2,565
19	Superheater	3,050
21	Superheater	2,525
22	Reheater	700
24	Economizer	3,050
25	Superheater	2,925

B. Determining Thin/Thick-Walled Pressure Vessels

The calculation of hoop stress in a tube depends on whether the tube can be considered to be a “thin-walled pressure vessel” or a “thick-walled pressure vessel.” The difference of the two terminologies is due to the ratio of the inner radius of the tube to the actual thickness of the wall of the tube. After examining ASME code, an equation was found to determine if a piece of tubing could be considered as "thin-walled (2008a Section 1 ASME code)." The equation is listed as follows:

$$\textit{thickness} < (\textit{Outer Diameter}/4)$$

Calculations were performed to determine if the tubing could be classified as "thin-walled" under ASME code. The results are in Table 8 in the Appendix. After examining the classifications of all eighteen cases, it was determined that each tube could be categorized as "thin-walled," thereby validating the ASME equations for hoop stress.

C. Determining Factors Contributing To Hoop Stress

After design pressures were found for all eighteen utilities, the design temperature had to be found. To determine the design temperature for each case, a proprietary Alstom computer program was used where various inputs were required. The tube's outer diameter, wall thickness, and design pressure were input. It was difficult to determine if a constant design temperature should be used for the material selected versus the higher-grade material since they technically had varying wall thicknesses. Because the temperature is a function of thickness, the question became whether or not using a constant design temperature would be a valid assumption. As the thickness of the wall is reduced, usually the tube wall temperature goes up. However, there was no way to determine the temperature of the metal at the time of failure. After speaking with multiple design professionals, it was determined that a constant design temperature should be used, as long as this assumption was specifically stated to the reader. The design professionals also suggested using the final ruptured thickness to determine the new stress allowables since that was the condition of the tubing at the time of failure.

D. Determining the Hoop Stress

After all the values that affected hoop stress were found, the original hoop stress or circumferential pressure, of each component before the tube went into operation was calculated. This was the stress that was present while the tube was in its original condition. The hoop stress prior to rupture in each component was then compared to each respective allowable stress for the material that was used. This would determine if the correct material was installed in the boiler. There have been instances where an incorrect material was installed into the boiler accidentally, which resulted in major failures. After inspection, it was determined that the correct materials had originally been installed in all eighteen of the utilities. This can be seen in Table 9 in the Appendix.

The hoop stress then had to be calculated when rupture occurred. One of the primary causes for excessive hoop stress was due to decreases in tube wall thickness. The eighteen case studies that were examined had many tubes that suffered a wall thickness reduction (from the originally designed wall thickness) due to various operating conditions. The reduced tube wall thickness measured after tube failure was used to find the ruptured hoop stress level. These were then compared to allowable stress values for higher-grade materials that could have been selected. This comparison determined if selecting higher-grade materials could have prevented failure.

The equation that boiler designers use is different from the standard (Pr/t) equation that is taught in basic engineering courses. The equation that was used to determine the level of hoop stress when the bent tube was in original condition was found in the “Pressure Vessel Design” pamphlet for nuclear and chemical applications (John F. Harvey 1963). It is important to understand that the hoop stress is different in straight and bent tubes. This meant that there would be two different equations to calculate hoop stress for “thin-walled pressure vessels.” The stress was different because of the change in stress distribution for the material that occurred during the bending process. The equation for a bent tube is:

$$\sigma_{hoop} = [(Pr/2h)*((2R_o+r)/(R_o+r))]$$

where P was the internal pressure, h was the thickness of the tube, r was the inner radius, and Ro was the bend radius of the tube. The tube dimensions of cases with fish-mouth failures in the actual bend of the tube are listed in Table 10 in the Appendix.

The hoop, or circumferential stress, is not dependant on what kind of metal the tube is made of, and what the temperature of the environment surrounding the tube is. The material and temperature determine the allowable stress level to which the circumferential stress is compared. The hoop stress depended on whether the tube is straight or bent, while the allowable was a function of the material and operating temperature. The equation above was for a tube with a failure in the bend of the tube. However, there were also tubes in which the failure occurred in the straight portion of the tube, rather than the bend. This meant that another equation for hoop stress had to be located to analyze those data for the cases where the failures occurred in the straight portions of the tubing. The following equation was used and found from the ITT Grinnell “Piping Design and Engineering” handbook:

$$\sigma_{hoop} =[P(D-t)/(2t)]$$

where P was the design pressure, D was the outer diameter of the tube and t was the wall thickness. Dimensions for tubes containing ruptures in the straight sections of their tubing are listed in Table 11 in the Appendix. A sample calculation for both bent and straight tube failures can also be found in the Appendix.

All of the bent and straight cases of fish-mouth failure have been placed in Table 12, which is located in the Appendix. They are listed for comparison purposes, along with original and ruptured tube thickness values. It was important to remember that the outer diameter of the tube remained constant throughout the tube's lifetime in the boiler. Since this value was kept constant, this meant that as the thickness decreased, the inner radius increased. As can be seen from Table 12, the hoop stress that occurred at rupture was higher than the hoop stress that existed in the original tubing. This was because of the reduction in wall thickness. The hoop stress was only calculated for these twelve cases because they were the only ones that failed due to a fish-mouth rupture. It was unnecessary for the other cases that did not fail due to excess circumferential stress.

It was important to determine if all of the original hoop stresses were below their allowable stress values. This was done to make sure that the selected material was originally designed according to ASME code. After comparing the values, all of them were acceptable to use. After this was completed, the ALSTOM Performance Design Department provided suggestions for alternate higher-grade materials that the utility could have selected. Several options were provided. The material that had the lowest acceptable value was listed under the results section in Table 4. This was done because although there would be several acceptable replacements, the material mentioned would be the most economical to the utility.

Chapter 4: Results

A. Results For Fish-Mouth Ruptures

Table 4 was constructed to show each original material, its original allowable stress, and the original and ruptured hoop stresses, along with the stress allowables for higher-grade materials. Table 4 can be found below:

Table 4: Results for all Fish-Mouth Ruptures

Utility Number	Straight/ Bent	Original Hoop Stress (psi)	Ruptured Hoop Stress (psi)	Original Material	Original Allowable (psi)	High-Grade Material	High-Grade Material Allowable (psi)
4	Bend	9,179	30,364	SA-192	12,400	SA-210C	18,300
6	Bend	4,802	7,299	SA-213 T22	7,080	TP304H	13,360
8	Bend	4,345	8,099	SA-213 T22	5,976	TP304H	12,592
13	Bend	5,847	16,231	SA-213 T22	8,112	TP347H	16,408
19	Bend	10,304	15,785	SA-213 TP347H	13,668	SUPER 304H	15,840
22	Bend	4,303	7,340	SA-213 T22	6,252	TP304H	12,784
9	Straight	5,125	10,932	SA-213 T22	5,792	TP304H	12,464
10	Straight	8,627	12,915	SA-213 TP304	10,060	TP347H	14,310
11	Straight	11,934	16,189	SA-213 TP347H	14,100	SUPER 304H	16,000
12	Straight	4,243	9,598	SA-210 A1	9,330	SA-210C	10,110
17	Straight	8,763	11,984	SA-192	11,040	SA-210C	15,500
21	Straight	9,784	21,282	SA-192	11,550	SA-210C	16,550

The allowable stresses listed above are from the "ASME Boiler Code For Carbon, Alloy, and Stainless Steels, Section II Part D." Of the twelve cases that failed due to excessive hoop stress, nine of them would have not failed if a higher-grade material had been selected. In fact, the higher-grade material for cases 6 and 22 could have withstood almost twice as much circumferential stress and still not have ruptured. For the straight tube failures, four out of six cases would have been okay, though some were close, correlating to a 66.7% success rate with a higher-grade material. For the bent tubes, five out of six would have not experienced failure, meaning an 83.3% percent success rate with the more expensive option. Cases 4, 11 and 21 would have still failed with the upgraded materials and have been highlighted to better show their identities. Table 5 shows the costs of the lesser and higher grade materials per pound:

Table 5: Cost per Material

Utility Number:	Original Material:	Cost per Pound:	Upgraded Material:	Cost per Pound:
4	SA-192	$0.76	SA-210C	$0.76
6	SA-213 T22	$1.45	SA-213 TP304H	$4.00
8	SA-213 T22	$1.45	SA-213 TP304H	$4.00
13	SA-213 T22	$1.45	SA-213 TP347H	$4.30
19	SA-213 TP347H	$4.30	SUPER 304H	$9.50
22	SA-213 T22	$1.45	SA-213 TP304H	$4.00
9	SA-213 T22	$1.45	SA-213 TP304H	$4.00
10	SA-213 TP304H	$4.00	SA-213 TP347H	$4.30
11	SA-213 TP347H	$4.30	SUPER 304H	$9.50
12	SA-210 A1	$0.76	SA-210C	$0.76
17	SA-192	$0.76	SA-210C	$0.76
21	SA-192	$0.76	SA-210C	$0.76

As one can see from the highlighted cells, four out of twelve upgraded materials had the same cost as the lesser grade. The costs listed above came from the estimating department at ALSTOM and are all listed as dollars per pound of material. The average weights for the components varied for the superheater, reheater and econominzer assemblies. The average weight of a superheater assembly was approximated at 3,000 to 3,500 pounds, while the reheater was estimated to be 2,000 to 2,500 pounds. The average weight for an economizer assembly was approximately 1,500 to 2,000 pounds. All of these ranges were given by the Performance Design Department and are for only one assembly. It is important to point out that each utility could have over one hundred assemblies for each component, so the overall cost would vary depending on the number of assemblies in each boiler. For example, if case 6 contained one hundred superheater assemblies, the lower grade material would have cost $1,450 compared to the $ 4,000 for the upgraded material. This is a $2,550 difference in cost between the two types of materials.

Although selecting a better grade material often proved to be a worthwhile investment, this was not the case with three of the utilities. Utility numbers 4, 11 and 21 would have experienced failures, even with the upgraded materials. The level of calculated ruptured stress for utility 4 was almost twice what even the highest-grade material could withstand. Although the suggested replacement for SA-192 material is SA-210C, any of the higher grade options, including the alloys and stainless materials, would not have been able to sustain that much stress. Rupture was inevitable even with the best grade of metal tubing.

B. Results For Tubes with Failures in the Bends

Where failures occurred in the actual bend of the tube, almost all of the damage was found on the “extrados” or outermost side of the bend. Table 6 below illustrates this finding:

Table 6: Locations of Failures on the Bends of Tubes

Case Number	Location of Failure	Side That Failed?
1	Bend	Extrados
2	Bend	Intrados
3	Bend	Extrados
4	Bend	Extrados
6	Bend	Extrados
8	Bend	Extrados
13	Bend	Extrados
16	Bend	Extrados
19	Bend	Extrados
22	Bend	Extrados
24	Bend	Extrados
25	Bend	Intrados

Bending produces changes in tube wall thicknesses. However, ASME code does not specify a minimum wall thickness for bent tube materials. It only specifies minimums for straight portions of tubing. It is up to the boiler designer to determine what is acceptable. ALSTOM allows for a ten percent reduction on the extrados end of the bend. During the bending process, the outer wall was put into tension; hence wall thinning would occur. Since the inner wall was in compression, the original wall thickness increased. An image of this process was created using AutoCad and can be seen in Figure 7.

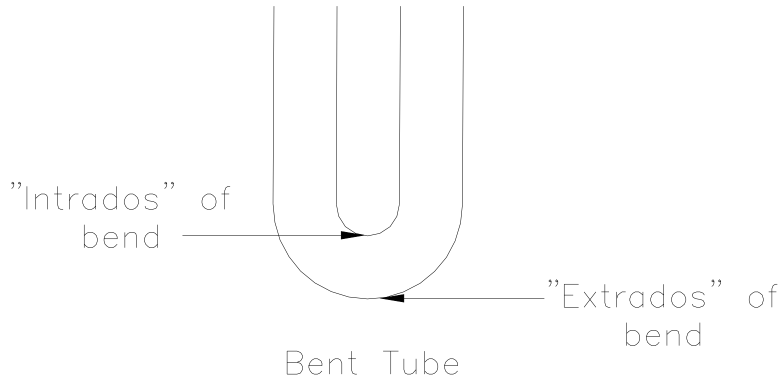

Figure 7: Bent Tube Illustration

Because the wall thickness on the extrados, or outer-most, side of the bend was at the smaller wall thickness, this correlated to a larger hoop stress because thickness was in the denominator of the equation:

$$\sigma_{hoop} = [(Pr/2h)*((2R_o+r)/(R_0+r))]$$

Cases 2 and 25 were the only cases that had a failure on the intrados, or innermost, side of the tubing. They were highlighted in Table 6 to make them easier to identify. It should be noted that cases 2 and 25 did not experience a fish-mouth rupture. They all failed due to cracking, rather than excess circumferential stress. This was probably because of the increase in wall thickness that was attained during the bending process. The increased thickness made the tube able to sustain a higher degree of hoop stress, but could not offer protection to cracking.

C. Results For Cracking Failures

Six of the cases examined contained tubing that experienced failures due to some form of cracking or leaking. Although the cause of these failures could not be quantitatively explained, the argument is qualitative. Failures occurred in many tubes that had not been heat treated after the bending process to relieve stresses in the material. Therefore, it is believed that no heat treatment process being applied to the tube made failure possible. These six cases can be found in Table 7 below:

Table 7: Tubing Materials and Classifications

Case #	Boiler Component	Material	Tubing Classification
1	SH	SA-210 A1	Carbon Steel
2	SH	SA-213 TP347H	Stainless Steel
3	Econ	SA-210 A1	Carbon Steel
16	Econ	SA-213 T91	Alloy
24	Econ	SA-210 A1	Carbon Steel
25	SH	SA-213 TP347H	Stainless Steel

Of the six cases, it was determined that five utilities decided to not perform any post-heating treatment that is typically done after the bending process. Although boiler designers often suggest these treatments, they are not required. These were cases 1, 2, 3, 24 and 25. For case 16, a hot-forming technique had been performed on the alloy material. All six instances of cracking or leaking occurred in either the economizer or superheater components.

As the tubes were bent in the manufacturing facility, they were overloaded with a large amount of stress. Typically, to help relieve the stresses in the grains of the material, a post-heating treatment is applied to the tubing. However, this treatment adds an extra cost to the assembly price. For example, stress relieving an assembly after bending can cost approximately $25 for each bend. Therefore, this can be an expensive process for a utility if they have multiple assemblies that contain a large number of bends.

Case 16 did have a post-heating treatment, yet failed anyway. It was highlighted in Table 7 to make it easy to identify. When the case report was examined, it was determined that the failure was due to severe fly ash erosion that caused both wall thickness reduction and cracking. The tube was made of SA-213 T91, which is a very expensive alloy material. It was the only case of the carbon and alloy cracking failures that did not contain SA-210 A1 material. The cost of SA-213 T91 is $6.50 per pound, versus the $0.76 per pound for the SA-210 A1 carbon steel. This shows that even though a more expensive material with an expensive treatment is usually a better investment, it will not absolutely guarantee that an outside factor such as fly ash will not harm it. The post-heating treatments only combat cracking, and do nothing to help against other harmful factors.

Two of the six cases contained superheater tubing made up of SA-213 TP347H stainless steel material. Most boilermakers recommend that any bent stainless steel tubing be solution-annealed to relieve stresses. It was found in the case reports that both utilities went against the advice of boiler designers and specifically requested that solution-annealing not be performed. This was undoubtedly due to the high cost of the stress relieving treatment. Solution annealing one assembly is approximately $300. Considering that there can be over one hundred assemblies in a superheater component, this would be a very costly process. Also, this is in addition to the $4.50 per pound that the tubing material alone costs. However, both tubing cases where no solution-annealing was performed resulted in failures, making it probable that the process would have been a good investment for the utility.

Although any sort of treatment to a tube is expensive, it was suggested through the examination of these six case studies that it is a worthwhile investment. The metallurgical laboratory that examined the sections of tubing all said that the cracking was a direct cause of not properly dealing with the stresses imposed during the bending process, with the exception of case 16. Therefore, 83.3 percent of the failures due to cracking probably could have been prevented if a greater initial monetary investment had been originally made by the utility.

Chapter 5: Conclusions and Recommendations

A. Conclusions

It has been determined that utilities are better off selecting the higher-grade materials. Nine of the twelve cases with fish-mouth failures would have had hoop stress less than the allowable with a higher-grade material. This is a 75 percent higher success rate. For the tubes with cracking, five out of six had no post-bending heating treatment applied to it. This is an 83.3 percent failure rate of the tubes not being treated properly after bending prior to being installed in the boilers. Although it cannot be certain that stress relieving would have absolutely prevented failure, a large majority of the tubes with no stress relief failed due to cracking. Although they are more expensive initially, it was shown in these eighteen cases that the higher-grade material and treatments generally resulted in greater life of the tubing. The number of failures that could have been reduced with better quality design procedures proved that there would be a greater chance of survival for the tube with these higher materials and processes.

B. Recommendations to Utilities

After examining all eighteen cases, it would be strongly recommended to the utilities to consider carefully when choosing materials for their boiler components. Rather than only concentrating on initial costs, the potential long-term outcomes with lower grade materials should be considered. Also, care should be taken with bent tubes to relieve any stresses that may have developed during the bending process. Although the lower grade materials may be more appealing at first, results show that this could be a costly mistake in time.

Appendix

Data Tables

Table 8: Classification of Materials

Case #	Inner Radius (in.)	Outer Diameter (in.)	Original Thickness (in.)	(D/4)	New Classification
1	0.838	2.125	0.225	0.531	Thin-Walled
2	0.625	2.000	0.375	0.500	Thin-Walled
3	0.800	2.000	0.200	0.500	Thin-Walled
4	0.727	1.750	0.148	0.438	Thin-Walled
6	0.625	2.000	0.375	0.500	Thin-Walled
8	1.085	2.500	0.165	0.625	Thin-Walled
9	0.643	2.125	0.420	0.531	Thin-Walled
10	0.708	2.000	0.292	0.500	Thin-Walled
11	0.780	2.000	0.220	0.500	Thin-Walled
12	1.047	2.500	0.203	0.625	Thin-Walled
13	1.070	2.500	0.180	0.625	Thin-Walled
16	1.200	2.750	0.190	0.688	Thin-Walled
17	1.117	3.000	0.383	0.750	Thin-Walled
19	0.780	2.000	0.220	0.500	Thin-Walled
21	0.727	1.750	0.200	0.438	Thin-Walled
22	1.085	2.500	0.165	0.625	Thin-Walled
24	0.675	1.750	0.200	0.438	Thin-Walled
25	0.607	2.000	0.393	0.500	Thin-Walled

Table 9: Original Hoop Stress and Stress Allowables

Case Number	Original Hoop Stress (psi)	Original Allowable (psi)
4	9,179	12,400
6	4,802	7,080
8	4,345	5,976
9	5,125	5,792
10	8,627	10,060
11	11,934	14,100
12	4,243	9,330
13	5,847	8,112
17	8,763	11,040
19	10,304	13,668
21	9,784	11,550
22	4,303	6,252

Table 10: Fish Mouth Failures in the Bends of the Tubing

Case Number	Boiler Component	Location of Failure	Design Pressure (psi)	Bend Radius (in)	Tube OD (in)	Original Tube Thickness (in)	Ruptured Tube Thickness (in)
4	SH	BEND	2,525	5.250	1.750	0.148	0.139
6	SH	BEND	2,990	8.000	2.000	0.375	0.282
8	RH	BEND	700	8.625	2.500	0.165	0.163
13	RH	BEND	1,056	6.750	2.500	0.180	0.071
19	SH	BEND	3,050	7.500	2.000	0.220	0.155
22	RH	BEND	700	7.250	2.500	0.165	0.102

Table 11: Fish Mouth Failures in Straight Tubing

Case Number	Boiler Component	Location of Failure	Design Pressure (psi)	Tube OD (in)	Original Tube Thickness (in)	Ruptured Tube Thickness (in)
9	SH	Straight	2,525	2.125	0.420	0.220
10	SH	Straight	2,950	2.000	0.292	0.205
11	SH	Straight	2,950	2.000	0.220	0.167
12	RH	Straight	750	2.500	0.203	0.094
17	ECON	Straight	2,565	3.000	0.383	0.290
21	SH	Straight	2,525	1.750	0.200	0.098

Table 12: All Fish-Mouth Failures

Case #	Boiler Piece	Design Press. (psi)	Bend Radius (in)	Tube OD (in)	Tube ID (in)	Inner Radius (in)	Original Tube Thk. (in)	*Original Hoop Stress (psi)*	Ruptured Tube Thk. (in)	Ruptured Inner Radius (in)	*Ruptured Hoop Stress (psi)*
4	SH	2,525	5.250	1.750	1.390	0.695	0.180	9,179.434	0.063	0.812	30,364.793
6	SH	2,990	8.000	2.000	1.250	0.625	0.375	4,802.778	0.282	0.718	7,299.347
8	RH	700	8.625	2.500	2.170	1.085	0.165	4,345.858	0.094	1.156	8,099.798
9	SH	2,525	n/a	2.125	1.285	0.643	0.420	5,125.149	0.220	0.843	10,932.102
10	SH	2,950	n/a	2.000	1.416	0.708	0.292	8,627.740	0.205	0.795	12,915.244
11	SH	2,950	n/a	2.000	1.560	0.780	0.220	11,934.091	0.167	0.833	16,189.671
12	RH	750	n/a	2.500	2.094	1.047	0.203	4,243.227	0.094	1.156	9,598.404
13	RH	1,056	6.750	2.500	2.140	1.070	0.180	5,847.874	0.071	1.179	16,231.828
17	ECON	2,565	n/a	3.000	2.234	1.117	0.383	8,763.192	0.290	1.210	11,984.741
19	SH	3,050	7.500	2.000	1.560	0.780	0.220	10,304.298	0.155	0.845	15,785.588
21	SH	2,525	n/a	1.750	1.350	0.675	0.200	9,784.375	0.098	0.777	21,282.143
22	RH	700	7.250	2.500	2.170	1.085	0.165	4,303.433	0.102	1.148	7,339.944

Sample Calculations

1. Hoop Stress in a Bent Tube:

<u>Case #4:</u>
P = 2525 psi
h = 0.18 in
r = 0.695 in
R_o = 5.25 in

$$\sigma_{hoop} = [(Pr/2h)*((2R_o+r)/(R_o+r))]$$
$$\sigma_{hoop} = [((2525 \text{ psi})(0.695\text{in})/2(0.18 \text{ in}))*((2(5.25 \text{ in})+0.695 \text{ in})/(5.25 \text{ in}+0.695 \text{ in}))]$$
$$\boldsymbol{\sigma_{hoop} = 9{,}179.43 \text{ psi}}$$

2. Ruptured Hoop Stress in a Bent Tube:

<u>Case #4:</u>
P = 2525 psi
h = 0.063 n
r = 0.812 in
R_o = 5.25 in

$$\sigma_{hoop} = [(Pr/2h)*((2R_o+r)/(R_o+r))]$$
$$\sigma_{hoop} = [((2525 \text{ psi})(0.812 \text{ in})/2(0.063 \text{ in}))*((2(5.25 \text{ in})+0.812 \text{ in})/(5.25 \text{ in}+0.812 \text{ in}))]$$
$$\boldsymbol{\sigma_{hoop} = 30{,}3064.79 \text{ psi}}$$

3. Hoop Stress in a Straight Tube:

<u>Case #9:</u>
P = 2525 psi
D = 2.125 in
t = 0.420 in

$$\sigma_{hoop} =[P(D-t)/(2t)]$$
$$\sigma_{hoop} =[2525 \text{ psi}(2.125 \text{ in}-0.420 \text{ in})/(2*(0.420 \text{ in}))]$$
$$\boldsymbol{\sigma_{hoop} =5125.15 \text{ psi}}$$

4. Ruptured Hoop Stress in a Straight Tube:

<u>Case #9:</u>
P = 2525 psi
D = 2.125 in
t = 0.220 in

$$\sigma_{hoop} = [P(D-t)/(2t)]$$
$$\sigma_{hoop} = [2525 \text{ psi}(2.125 \text{ in}-0.220 \text{ in})/(2*(0.220 \text{ in}))]$$
$$\boldsymbol{\sigma_{hoop} = 10{,}932.10 \text{ psi}}$$

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