The Design of an Evaporating Unit for Continuous Boiler Blow-down Water Purification

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Approved:

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### List of Symbols

- $q$: Heat flow rate in (J/s)
- $h$: Convective heat transfer coefficient (Watts/meter$^2$s)
- $h_i$: Internal heat transfer coefficient (Watts/meter$^2$s)
- $h_{mic}$: Microscopic heat transfer coefficient (Watts/meter$^2$s)
- $h_{mac}$: Macroscopic heat transfer coefficient (Watts/meter$^2$s)
- $A$: Area (m$^2$)
- $\Delta T$: Temperature difference (K)
- $k$: Thermal conductivity (Watts/m K)
- $c$: Specific heat (J/kg K)
- $\sigma$: Surface tension (N/m)
- $\mu$: Dynamic viscosity (N s/m$^2$)
- $S$: Suppression factor used in Chen correlation (dimensionless)
- $D$: Tube Diameter (m)
- $x$: Quality (dimensionless)
- $Nu$: Nusselt number (dimensionless)
- $Re$: Reynolds number (dimensionless)
- $Pr$: Prandtl number (dimensionless)
- $\dot{m}$: Mass flow rate (kg/s)
- $\rho$: Density (kg/m$^3$)
- $\lambda_{fg}$: Heat of vaporization (J/kg)
- $p$: Pressure (N/m$^2$)
- $L$: Length (m)
- $X_{tt}$: Martinelli parameter (dimensionless)
- $F$: Reynolds number factor (dimensionless)
- $G$: Mass flow velocity (kg/m$^2$s)
- $\dot{w}$: Mass flow rate of steam (kg/s)
ABSTRACT

This report describes calculations for the design of an evaporator portion of an evaporator/condenser system capable of recycling water from boiler blow-down water. These calculations are based upon heat balance equations with empirical correlations for convective boiling flows inside of pipes.

The system may be incorporated into a steam plant facility for the purpose of saving water. This system could be beneficial if applied in a desert climate or in a marine application. In these environments freshwater must be generated or stored. If water is purchased for these facilities the cost could become significant depending on the amount of water used.

Large industrial facilities require large commercial boilers. These boilers need to discharge a certain amount of water during operation to maintain clean heat transfer surfaces. This water is typically discarded as waste. If some of the water being discharged can be recycled then the operating company can save water as well as money.

The results of the analysis performed show that the proposed unit is feasible. The heat source of the evaporator unit is explored and the most economical heat source is selected and used in the analysis. The data obtained in the heat balance is consistent with Chen’s findings [6].
1.0 Introduction

As the price of energy continues to rise, more technology is being developed to increase energy savings. Examples of this include the use of solar panels as well as wind turbine farms. Alternative and green methods are not the only area of interest for cost savings. The aspect of monetary savings may also be represented in existing energy sources. These energy sources can include boiler technology by more efficient steam plant operation. Boilers operate in a variety of applications and locations. These can include marine boilers in commercial ships such as liquid natural gas (LNG) tankers. Boilers also operate in commercial nuclear and conventional power stations. No matter what application and location boilers have one common need. What is it? The need to maintain clean heat transfer surfaces so that the most efficient heat transfer can occur. With proper heat transfer occurring within the boiler the cost incurred by the operating company can be reduced.

1.1 Components of a Boiler

There are a few basic parts of a conventional boiler. These parts are the tubes, steam drum, and water drum. These items can be seen in Figure 1.
Heat is transferred from the exhaust gas generated by the combustion of fossil fuel in the furnace to the water within the boiler tubes. The water boils within the tubes generating steam and the steam separates at the steam water interface located in the steam drum. Within the steam drum there is a perforated pipe known as a dry pipe. This pipe carries the steam out of the steam drum to perform whatever application the boiler is operating for. The steam leaving the boiler is not able to carry any kind of solid material that may have been originally dissolved in the boiler water. As a direct result of this, an increase in the concentration of solid impurities can collect inside the boiler steam drum and tubes. Depending on how long the boiler operates the concentration of these impurities can become quite large. This could cause boiler heat transfer surfaces to become coated with hard scale that can reduce heat transfer capability. To maintain the required amount of heat transfer with a scale layer formed on the heat transfer surfaces increased energy consumption would be required. Boiler blow-down is a simple way to prevent the fouling of heat transfer surfaces within a boiler.

1.2 Boiler Blow-down

As mentioned, the excessive concentrations of dissolved material within the boiler water should be minimized to prevent hard scale build up. This is performed by periodically draining water from the boiler and replacing the volume drained with fresh water. This operation reduces the concentration of the impurities in the boiler. An unfortunate result of this evolution is that it reduces the amount of heat and pressure inside the boiler as the fresh water mixes with the hot water in the steam drum. The freshwater must be heated to saturation temperature and this causes additional fuel consumption.

Another method of boiler blow-down is a continuous boiler blow-down. A small amount of water is drained from the boiler steam drum continuously as the boiler operates. To maintain a constant level in the boiler steam drum the loss of boiler water is offset by the continuous addition of feed water by the boiler feed water system. This method of impurity level reduction has the benefit of not causing a significant decrease
in boiler temperature and pressure. The water that is continuously removed is typically discharged as waste. In this project, the possibility of removing the impurities from the high temperature boiler water by evaporation is explored. The water evaporated could then be condensed and stored for reuse as boiler makeup water. This paper presents a method to design and size the evaporator of a distillation blow-down system to be used in a marine application.
2.0 Theory and Methodology

2.1 Boiler water purification/distillation

If boiler blow-down water is going to be recycled then the dissolved impurities within the water must be removed. However there is one problem. Boilers operate at high temperatures and pressures. The use of filters or resin beds with fine elements or resins could prove impractical and expensive.

A widely used method to remove salt and other impurities from sea water is to vaporize the sea water in a reduced pressure. The vapor generated cannot maintain any of the dissolved salt causing the solid materials to separate from the vapor. This method of making freshwater is known as distillation. The heat exchanger involved with this process is commonly known as an evaporator. The heating medium for an evaporator used in the distillation process could be low pressure steam or an electric heating element. Once the sea water is boiled off, the vapor can be condensed in another portion of the distiller chamber where cooling coils are located. The condensed distillate then is collected on pans and pumped to tanks for consumption or boiler use. A basic representation of this system is shown in Figure 2.

![Figure 2 – A generic representation of an evaporator/condenser system](image-url)
2.2 Evaporator unit

A similar approach may be used for the removal of impurities in the boiler blow-down water. The boiler water removed from the steam drum is already a heat saturated liquid. The significance of the water’s temperature applies to the boiling process. If the blow-down water removed is a heat saturated liquid then only the required amount of enthalpy to boil the water needs to be added. This required amount of heat is known as the “latent heat of vaporization”. The system needed to add the required amount of heat to the boiler water is a heat exchanger tube. This is schematically shown in Figure 3.

Figure 3 - Cross sectional view of the evaporator

If steam is used as a heat source the correct type of steam must be determined. Steam is used in a variety of applications in a steam powered vessel making multiple temperatures and pressures of steam readily available. The heating medium for this
evaporator could come from a stage of one of the propulsion turbines or even be a small portion of the superheated steam being generated by the boiler. The steam from these locations is superheated and has a significant amount of enthalpy. Another possible source of steam is the steam condensing in the main condensers of the ship. The main condenser of the ship is where the propulsion turbines exhaust into. A general representation of the conventional steam system is illustrated in Figure 4.

![Diagram of the steam cycle of the ship](image)

**Figure 4 – Basic representation of the steam cycle of the ship with sources of heating steam for the evaporator**

Even though multiple sources of steam are available not all sources may have the ability to vaporize water. Assuming that the heating steam is capable of evaporating the blow-down water into a steam/water mixture then the impurities will separate from the
vapor. The removed solids could collect in the bottom of a tank and be discharged. This process would help save at least some of the water that would normally be thrown away as waste.

2.3 Generation of a steam/water mixture

A complete vaporization of the entire amount of boiler blow-down water into dry saturated steam is not the intent. If this occurred, then the collection tank would just be completely covered by hard scale from the impurities in the boiler water as these solids precipitate out of the steam. For this project a quality of approximately 50% vapor was chosen for the steam/water mixture. This would allow the possible saving of 50% of the boiler blow-down water while concentrating the other 50% of the water containing the impurities as waste. This is so because the lower density water vapor will rise in the chamber while the more dense water will collect at the bottom of the chamber.

A steam ship was chosen as the platform for this project. The operating boiler pressure is 600 psig. The steam turbines of the ship use superheated steam for propulsion. Due to piping flow reductions from the steam drum to the proposed system a pressure of 529 psig was chosen as the heat saturated boiler water at 473 F (245 C).

All aspects of the heat exchanger are unknown so dimensions as well as the steam heat source need to be determined. One design characteristic is specified. The heat transfer geometry would be a tube. A tube with an inner diameter of 2 cm was chosen. A flow rate of .06295 kg/s was specified as the blow down capacity. This flow rate was chosen because it is small enough to not affect boiler water levels and confuse operators when the ship is maneuvering. A second reason this flow rate was chosen is that this level reduction should be easily maintained continuously with a conventional boiler feed system.

2.4 Heat balance

An analysis needed to be performed to track the heat flow in this evaporator. The starting point was chosen to be the inner tube wall. This location is where convection heat transfer is occurring between the water being heated and the surface of the pipe.
This heat flow rate can be calculated by:

\[ q = h_i A (\Delta T) \] \[1\]

The difference in temperature indicated above is between the internal surface of the pipe of the heat exchanger and the bulk temperature of the boiler water flowing within the pipe. The \( h_i \) term is the convective heat transfer coefficient. The \( A \) in equation (1) is the surface area through which the heat is being transferred.

### 2.5 Heat Transfer Coefficient

To calculate the heat transfer coefficient \( h_i \), the Chen correlation was used [1, page 613], [6]. As stated above the boiler water that has been removed from the steam drum is heat saturated water close to boiling. Since the boiler water is a heat saturated liquid carrying a significant amount of heat it is assumed that the transition from bubble and slug flow in flow regimes of boiling quickly turns into the annular flow regime. For this project it is assumed that the boiling within the evaporator tubes occurs in the annular flow regime.

Chen has suggested that there is a two part correlation to the boiling heat transfer coefficient in the annular flow region. The two parts consist of a convection heat transfer coefficient as well as a nucleate boiling heat transfer coefficient [6, page 324]. The convective heat transfer coefficient is known as the macroconvective heat transfer coefficient. The macroconvective heat transfer coefficient describes the normal heat transfer associated with fluid flow. The microconvective heat transfer coefficient is associated with nucleate boiling which correlates to bubble nucleation and growth [6, page 325]. The two coefficients are calculated separately and then their affects are additive to create a singular heat transfer coefficient.

\[ h = h_{mac} + h_{mic} \] \[2\]
The Chen correlation of the annular flow region was chosen for this project because this correlation has been tested and the results proven. This specific correlation has provided satisfactory data for water, methanol, cyclohexane, pentane, heptane, and benzene for a variety of pressures. The pressure ranges evaluated were between 0.5 and 35 atm and the quality of vapors produced is reported to be between 1 and 71% [1, page 675], [6].

Numerous experiments have been conducted on the area of forced convection boiling. However at this time it is not yet possible to understand all of the characteristics of forced convection boiling due to the significant number of variables the process depends upon. Another point that is not completely understood in the area of forced convection boiling is the two phase flow patterns that occur. These two phase flow patterns occur as the quality of the vapor-liquid mixture increase during vaporization [1, page 667-668]. Most of the data used for the correlations of forced convection represented in this project are based upon photographs taken by high speed cameras. These photographs are then used to obtain a qualitative description of the process [1, page 668].

Chen is one of the first researchers to suggest a forced convection boiling correlation. His proposal has proven very useful and is widely used [2]. Chen proposed a major difference between conventional nucleate pool boiling and nucleate boiling in forced convention. The difference between the two boiling mechanisms corrects the phenomena occurring between the hot inside surface of a pipe and the fluid bulk temperature in forced convection boiling; compared to the temperature difference of the boiling fluid and surface occurring in nucleate pool boiling. Chen proposed that the Reynolds number for microconvective boiling heat transfer is governed by the growth rate of bubbles. The growth rate of the bubbles within the boiling liquid whether in convective boiling or nucleate boiling is constant for a designated wall superheat. Observing this Chen theorized that the Reynolds number used in the original Forster-Zuber equation is an effective Reynolds number that is directly correlated to an effective wall superheat. The original Forster-Zuber correlation for nucleate boiling ignored this
value of the effective superheat since it was insignificant in nucleate pool boiling [6, page 325]. However, this value cannot be ignored in convective boiling.

Chen also noticed that as evaporation took place the vapor voids formed within the flow channel had the effect of changing the velocity of the fluid flowing through the pipe. The difference in fluid velocity created a change in the heat transfer coefficient of convection as would be compared to a regular single phase flow of the fluid [2]. Based upon these findings Chen theorized that the flow conditions internal to a pipe undergoing convective boiling vary differently based upon the quality increase of the vapor water mixture flowing through the pipe [3, page 123]. This phenomenon is explained by Chen using an effective Reynolds number in the Dittus-Boelter correlation [6, page 324].

Chen found two dimensionless functions that can be used to describe his findings. These values are \( S \) and \( F \) [3, page 123]. The value of \( S \) is known as the suppression factor and relates the suppression of nucleation sites internal to the pipe where two phase flow is occurring. The value of the suppression factor corrects the fully developed nucleate boiling prediction determined by Forster-Zuber for specific wall superheat as described above [5, page 613]. This corrected value is known as the microscopic contribution to the two phase heat transfer coefficient and is listed below [5, page 613]:

\[
h_{\text{mic}} = 0.00122 \left( \frac{k_t^{0.79} \sigma^{0.45} \mu_t^{0.49} \Delta T_x^{0.24} \Delta p_{\text{sat}}^{0.75}}{\rho_t \rho_v^{0.24} \lambda_{fg}^{0.25}} \right) S \quad [3]
\]

The value of \( S \) in the equation above was calculated by Chen using an empirical function of the effective or Reynolds two phase flow number [6, page 322]. Values of the suppression factor calculated by Chen are shown in Appendix A.

The suppression factor can be calculated using the following equation [5, page 614]:

\[
S = (1 + 2.56 \times 10^{-6} \ Re_{tp}^{1.17})^{-1} \quad [4]
\]

The value of the Reynolds number listed above with the subscript “tp” is known as the Reynolds two phase flow number described above [5, page 614]. The Reynolds two phase flow number is calculated using the following equation [5, pages 613-614]:

\[
Re_{tp} = \frac{\rho_{avg} \rho_{avg}^{0.24} \mu_{avg}^{0.25} \Delta T_x^{0.24}}{k_{avg}^{0.45}} \quad [5]
\]
The value of G in the equation above is known as the mass flux vector which can be calculated below [4, page 6].

\[ G = \frac{m}{A} \]  

[7]

The value of F in the equation (5) is the Reynolds number factor. This value is calculated using the Martinelli parameter [5, page 610] incorporated into the equations listed below [5, page 614]:

\[ F = \begin{cases} 1.0 & \text{when } \frac{1}{X_{tt}} < 0.1 \\ 2.35 \left( \frac{1}{X_{tt}} + 0.213 \right)^{0.736} & \text{when } \frac{1}{X_{tt}} > 0.1 \end{cases} \]  

[8]

Where \( \frac{1}{X_{tt}} \) is:

\[ \frac{1}{X_{tt}} = \left( \frac{x}{1-x} \right)^{0.9} \left( \frac{\rho_l}{\rho_v} \right)^{0.5} \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \]  

[10]

The value of F was obtained by Chen as a function of the Martinelli parameter by use of an empirical correlation of heat transfer data as well as a momentum-analogy analysis [6, page 322]. Values of the Reynolds Number factor obtained by Chen are shown in Appendix B.

After performing the calculations listed above, the macroscopic portion of the two phase convective heat transfer coefficient can be determined. The macroscopic portion of the convective boiling heat transfer coefficient was described by Chen using the Dittus-Boelter macroconvective heat transfer relation for forced convection [3, page 123]. The standard Dittus-Boelter correlation is:

\[ Re_{tp} = Re_l F^{1.25} \]  

[5]
\[ Nu = 0.023(P_{r}^{0.4})(Re^{0.8}) \]  

Since the Nusselt number is defined as:

\[ Nu = \frac{hD}{k} \]  

The heat transfer coefficient is then given as:

\[ h = 0.023(P_{r}^{0.4})(Re^{0.8}) \left( \frac{k_{l}}{D} \right) \]  

This correlation is corrected for the two phase flow occurring within the pipe using the two phase Reynolds number calculated above. The two phase Prandtl and thermal conductivity values are used in the equation as well. However, these values are the same values as the liquid in two phase flow. Since the evaporator tube is assumed to occur in the annular flow regime of boiling heat is transferred through the liquid film adhering to the wall. Therefore the liquid Prandtl and thermal conductivity have the most significant affect [6, page 324]. The following equation is obtained:

\[ h_{mac} = 0.023(P_{r_{tp}}^{0.4})(Re_{l}^{0.8}) \left( \frac{k_{tp}}{D} \right) F \]  

Once the microconvective and macroconvective heat transfer coefficients have been calculated they can be added to obtain the total convective heat transfer coefficient for the two phase flow as described above [5, page 613]:

\[ h = h_{mic} + h_{mac} \]  

\[ 2.6 \quad \text{Tracking the quality increase} \]

As previously stated the design requirements of the evaporator require providing the correct amount of latent heat of vaporization to the heat saturated water to obtain the
50% quality vapor. The Chen correlation described in the previous section can be used in a downstream marching scheme to calculate the required length of the heat exchanger to obtain 50% quality. A schematic representation of the computation method is shown in Table 1.

Table 1- A Schematic representation of the calculation process
The values used in the Chen correlation are listed in Table 2 indicated below.

<table>
<thead>
<tr>
<th>properties</th>
<th>water</th>
<th>Heating Steam</th>
</tr>
</thead>
<tbody>
<tr>
<td>pressure</td>
<td>3Q47986</td>
<td>423910.345</td>
</tr>
<tr>
<td>pressure</td>
<td>3Q47986</td>
<td>423910345</td>
</tr>
<tr>
<td>pressure</td>
<td>529</td>
<td>KPA</td>
</tr>
<tr>
<td>pressure</td>
<td>518</td>
<td>K</td>
</tr>
<tr>
<td>pressure</td>
<td>473</td>
<td>F</td>
</tr>
<tr>
<td>pressure</td>
<td>245</td>
<td>C</td>
</tr>
<tr>
<td>0.06295 kg/s mass flow rate</td>
<td>0.031475 kg/s mass flow rate</td>
<td>0.02 m diameter</td>
</tr>
<tr>
<td>3.14E-04 m^2 area of flow element</td>
<td>100.18606 kg/m^2 s Mass flux of water</td>
<td></td>
</tr>
<tr>
<td>0.58 W/mK Thermal cond. Of water</td>
<td>48054.49 J/kg K Specific heat</td>
<td></td>
</tr>
<tr>
<td>806.425 kg/m^3 density</td>
<td>0.00931 N/s/m^2 viscosity of water</td>
<td></td>
</tr>
<tr>
<td>0.05885 N/m Surface tension</td>
<td>0.311 Pr Prandtl number</td>
<td></td>
</tr>
<tr>
<td>2 N Tubes</td>
<td>0.016 W/mK Thermal cond. Of vapor (at 125 C)</td>
<td></td>
</tr>
<tr>
<td>3640.22 J/kg K Specific heat</td>
<td>182788 kg/m^3 density</td>
<td></td>
</tr>
<tr>
<td>1.73062E-05 N/s/m^2 viscosity of vapor</td>
<td>1741080 J/kg heat of vaporization</td>
<td></td>
</tr>
</tbody>
</table>

Table 2 – Thermodynamic properties used in the heat balance analysis

This calculation method is applicable based upon the heat flux over a differential length “ΔL” of the tube being directly linked to the increase in quality of the steam and water mixture at its saturation temperature illustrated by the following [1, page 677]:

\[ \dot{m}\lambda_f \Delta x = q\pi D\Delta L \quad [16] \]

This equation can be rearranged with the values of Δx and ΔL on either side.

\[ \Delta x = \frac{q\pi D\Delta L}{\dot{m}\lambda_f} \quad [17] \]

Initially the quality of the boiler water is assumed to be .001 at the entrance of the heat exchanger. The values of Table 1 were entered in each row as the quality of the
steam and water mixture ($\Delta x$) values increased. Since the values of the microconvective and macroconvective heat transfer coefficients change as the quality of the water and vapor mixture increase the combined heat transfer coefficient for forced convection boiling changes as well. The assumption of constant wall superheat was made based upon saturation conditions occurring within the heat exchanger.
3.0 Results and Discussion

While analyzing the methods for designing the system described above many interesting aspects of the convective boiling as well as heat transfer were observed. In addition to these findings some of the initial approaches attempted in solving the heat balance analysis were found to be incorrect or impractical. The result of these dead ends opened the door to different approaches until the correct path was obtained. An example of this is the method of determining the source of the heating steam.

3.1 Initial attempts

Initially the heating medium chosen to be used in the evaporator was superheated steam. Superheated steam has much more enthalpy per kilogram than normal dry saturated steam. With the more enthalpy available per kilogram more heat may be available to use in evaporating the vapor/water mixture. This fact resulted in a hypothesis that it may be possible to design a heat exchanger using superheated steam as the heating medium without having the superheated steam condense. The possible heat exchanger for this system could be an annulus type heat exchanger illustrated in Figure 5.

![Figure 5 – Annulus type heat exchanger](image-url)
Without the presence of a phase change the simple correlation of the two fluids within a heat exchanger can be described by the equation listed below:

\[ \dot{m} \dot{\lambda}_f \dot{\Delta} x = \dot{w} c \Delta T \] \hspace{1cm} [18]

The temperature difference “\( \Delta T \)” in the equation above refers to the difference in temperature between the steam’s inlet and outlet of the heat exchanger. The \( \Delta x \) term in the equation above refers to the quality increase of the vapor water mixture. The equation above can be rearranged to obtain:

\[ \frac{\dot{m} \dot{\lambda}_f \dot{\Delta} x}{\dot{w} c} = \Delta T \] \hspace{1cm} [19]

The equation above describes the change in temperature as the quality of the steam and water mixture increases. Using the equation above, an excel spreadsheet could be set up with an iterative process written to obtain the outlet temperature of the steam over certain length increments of the heat exchanger. However further investigation into the Chen correlation resulted in a key finding. The Chen correlation requires a constant amount of superheat at the pipe’s inner wall [5, page 614]. This means that the pipe’s inner wall is maintained at a constant temperature over the length of the pipe from entrance to exit. The data used in generating the empirical correlations was originally obtained with a constant wall superheat maintained. With superheated steam being used as the heating medium the assumption of constant wall temperature may not be valid. Further data collection would need to be obtained to verify if the different wall superheat temperatures provided the same results. Data was not available however. This resulted in the utilization of a new heat source for the evaporator. The heating steam was changed to the condensing steam in the ship’s main condenser.
3.2 Final design

There are two main areas of the ship’s propulsion plant where phase change is occurring. One of the areas is the boiler where the steam is being produced. The other area is known as the ships main condenser. The main condenser is where the steam coming from the ships turbines condenses. The medium for removing the latent heat of vaporization is usually seawater. If the evaporator tubes being used for the boiler blow-down evaporator were placed at a location where steam was condensing on them then a constant wall temperature would be obtained. The assumption is made that the significant amount of steam being introduced from the rotation of the ships turbines maintains the evaporator tubes at a constant temperature. This constant wall temperature is not a significant value and provides the necessary amount of superheat for the convective boiling occurring within the evaporator tubes.

Once the steam exhausting from the propulsion turbines was chosen as the heating medium the Chen correlation spreadsheet was generated. The initial analysis performed resulted in a singular pipe that was too long. Due to space restrictions on any type of ship a pipe even several meters long would not fit in the main condenser. Therefore to account for the size restrictions of the evaporator system the concept of a multiple pipe heat exchanger was proposed. The more pipes added to the heat exchanger would increase the heat transfer surface area available to pass the same flow rate of the fluid through multiple pipes. Adding multiple pipes to the heat exchanger changed certain aspects of the design however.

The Chen correlation requires calculation of a Reynolds number. This dimensionless number is used to obtain the microconvective and macroconvective heat transfer coefficients within the heat exchanger pipe. The Reynolds number originally calculated was based upon the mass flow rate through a singular pipe. A multiple pipe heat exchanger would require calculating a different mass flow rate of each tube.

The final result of the heat balance analysis performed can be observed in Appendix C. Each row of Appendix C describes the quality increase occurring in small increments of the evaporator pipes. Column 1 of the table is the length of the heat exchanger. The
The values of column 1 are the sum of each incremental length of the heat exchanger indicated in column 2. The values of the liquid Reynolds number for each incremental length of the analysis are written in column 3. The Martinelli parameter is calculated using equation 10 and can be observed in column 5. The value of column 5 is the reciprocal of column 4. The values of the Reynolds number factor is listed in column 6 and corrects column 3 values of the liquid Reynolds number to obtain the two phase Reynolds number of column 7. Column 8 provides the suppression factor for each length increment and is used to calculate the microscopic heat transfer coefficient of column 10 using equation 3. The macroscopic heat transfer coefficient is calculated in column 11 and is determined using equation 14. The combined heat transfer coefficient of microscopic and macroscopic heat transfer is indicated in column 12. The wall superheat is indicated in column 13 and is constant for the length of the pipe. Column 14 is the heat flow rate and is obtained using equation 1. The incremental increase in quality of each length of pipe is observed in column 15. The increase in quality of the heat exchanger from entrance to exit can be observed in column 16.

The final values of the heat exchanger design are indicated in the last row of Appendix C. The heat exchanger consists of two tubes with a .02 m diameter. The length of the two tubes is 1.28 m long. An attempt was made to maintain the heat exchanger as small as possible since its operating platform is a ship.

The data provided in Appendix C can provide some conclusions to the Chen correlation. For example it is observed that the value of \( \frac{1}{X_{it}} \) increases as the quality of the steam/water mixture increases.

\[
\frac{1}{X_{it}} = \left( \frac{x}{1-x} \right)^{0.9} \left( \frac{\rho_l}{\rho_v} \right)^{0.5} \left( \frac{\mu_v}{\mu_l} \right)^{0.1} \tag{20}
\]

The \( \frac{1}{X_{it}} \) is then used in calculating the Reynolds number factor (F). The Reynolds number factor is used in correcting the single phase Reynolds number for a two phase flow. Thus the turbulence created by the increasing vapor quantity in the two phase flow becomes more substantial with an increasing vapor fraction [6, page 326]. The result of this is a larger two phase Reynolds number:
The larger two phase Reynolds number results in an increase of the macroscopic heat transfer coefficient.

\[ Re_{tp} = \frac{G(1-x)D}{\mu_l} F^{1.25} \quad [21] \]

The data in Appendix C indicates that the value of S decreases to near a value of one at low values of the two phase Reynolds number [6, page 326]. The trend of the value of S increases with smaller values of the two phase Reynolds number [6, page 326]. Chen’s data specified that at low flow rates with low vapor fraction that the microconvective heat transfer coefficient played a more important role in forced convection boiling and this can be observed in Appendix C [6, page 326]. The \( h_{mic} \) or the microscopic convective heat transfer coefficient is consistently larger than the macroconvective heat transfer coefficient. Therefore the dominant effect in the evaporator is the nucleate boiling term. The values of S decreases as the two phase flow Reynolds number increases as indicated below:

\[ S = (1 + 2.56 \times 10^{-6} \quad Re_{tp}^{1.17})^{-1} \quad [23] \]

The resulting microconvective heat transfer value decreases as the quality increases in the steam water mixture. However, the microconvective heat transfer coefficient is still the dominant term in the combined heat transfer coefficient.
4.0 Conclusions

This paper provides a method to design an evaporator unit for the purpose of recycling boiler blow-down water. The blow-down water removed from a boiler is typically discharged as waste. If some of this water is recovered, purified, and reused then the cost incurred by the operating company can be reduced.

The calculations used in the analysis of this evaporator are based on the principle of energy conservation using empirical heat transfer correlations. The heat balance performed calculates the required amount of enthalpy that must be added to heat saturated water within the evaporator to increase the quality of the vapor/water mixture. The heat exchanger causes the blow-down water to undergo convective boiling. A 50% quality vapor exiting the heat exchanger was selected as the design target in this analysis.

The final design of the evaporator unit consisted of a heat exchanger containing two tubes. These tubes are 1.28 meters long with an inner diameter of .02 meters. The Chen correlation was used to describe the convective boiling process. The Chen correlation is used to calculate the nucleate boiling heat transfer coefficient and the convective heat transfer coefficient correlations using the dimensionless functions S and F.

The calculated values of the S and F parameters shown in Appendix C agree reasonably well with data provided in Chen’s paper (Appendices A and B). These two values that correct the standard equations used in convection heat transfer vary significantly based upon the flow conditions within the heat exchanger and they must be incorporated into the calculation of the heat transfer coefficient. The local heat transfer coefficients calculated for each incremental length of heat exchanger pipe use the values of S and F and are represented in Appendix C.

The system described above may be a very suitable addition to a steam powered vessel that must create its own water for the crew’s consumption. Additional savings are possible if the evaporator is located in the ship’s main condenser. The steam being condensed has already served its purpose rotating the turbines and must lose its latent heat of vaporization anyway. The question of cost to the operating company is the key
factor. The addition of a system listed above provides an additional “one time” cost to the manufacturer of the steam vessel. The purchaser would have to determine if the increased cost of the system specified above would provide savings over the life of the ship.
5.0 References


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M. Mohammad Shah

Evaluation of General Correlations for heat transfer during Boiling of Saturated Liquids in Tubes and Annuli

HT2005-72025
Appendix A

Suppression Factor “S” Vs. Two phase Reynolds number

Obtained From:

Appendix B

Reynolds number factor “$F$” Vs. $1/X_{tt}$

\[
\frac{1}{X_{tt}} = \left( \frac{x}{1-x} \right)^{0.3} \left( \frac{\rho_1}{\rho} \right)^{0.5} \left( \frac{\mu_2}{\mu} \right)^{0.1}
\]

Obtained From:

Appendix C

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