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# Study on The Thermal Performance of High Pressure Three-Zones Feed Water Heaters

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## Abstract

The thermal efficiency of a steam power plant cycle can be thermodynamically improved using different ways. A classical method involves introducing feed water heating system, which increases the mean temperature of heat addition. The feed water heater is a heat exchanger used in steam power plants to preheat water delivered to a steam generating boiler, in such a way that the necessary thermal energy transferred to the steam to increase the enthalpy of the system will be lower. Numerous researchers have reported enhancement in thermal efficiency by dividing overall enthalpy equally via feed water heaters. In this paper, a computer program is developed for studying and predicting the performance of high pressure three-zones feed water heaters. The very well known MATLAB tool is used as a frame-work for thermal calculations. Several parameters are studied such as terminal temperature difference (TTD), drain cooler approach temperature (DCA) and the temperature rise (TR). The computations have been performed successfully using practical data.

**Keywords:** Feed water heaters, steam power plant, terminal temperature difference, MATLAB.

## الملخص

يمكن تحسين الكفاءة الحرارية للمحطات البخارية لتوليد الطاقة باستخدام طرق مختلفة. احدى الطرق الكلاسيكية التي تستخدم لرفع الكفاءة الحرارية للمحطات البخارية هي عن طريق زيادة متوسط درجة الحرارة التي يتم إضافة كمية الحرارة عندها. هذه العملية تتم عن طريق إدخال أنظمة تسخين مياه التغذية. سخان الماء المغذي عبارة عن احد مكونات محطات توليد الطاقة يستخدم لتسخين الماء المُسَلَّم إلى مرجل (غلاية) توليد البخار. وفي هذا البحث تم كتابة و تطوير برنامج حاسوب باستخدام برنامج (ماتلاب) للتنبؤ بالأداء الحراري لمسخنات مياه التغذية ذات الضغط العالي المكونة من ثلاث مناطق. حيث أمكن توقع الأداء الحراري لمسخنات مياه التغذية ذات الضغط العالي بسرعة فائقة جدا دون الحاجة للحسابات اليدوية الطويلة وبهذا تم توفير الوقت والجهد بواسطة هذا البرنامج. تم إجراء الحسابات بنجاح باستخدام البيانات العملية. تمت دراسة عدة متغيرات مثل فرق درجة الحرارة الطرفية (TTD) وقيم درجات حرارة التصريف (DCA) وارتفاع درجة حرارة مياه التغذية (TR).

**كلمات مفتاحية:** الاداء الحراري، المحطات البخارية ، مسخنات مياه التغذية ، درجة حرارة مياه التغذية.

## 1. Introduction

The limitations of power resources and continuing increase in the demand of power generation on a large scale is considered a major concern for the world. The growth of any nation relay on capital energy production and consumption. Thermal power plants play essential role in development of countries due to their vital contribution towards power production. Any thermal power plant should be operated at highest point of competency and effectiveness. To increase the thermal performance and consequently the overall efficiency of the steam power plants, the feed water heaters are incorporated to form a major part in the regenerative system of a plant (Kumar & Kumar, 2014). Closed type feed water heaters, which are simply shell and tube heat exchangers, influences the performance of steam power plants either directly or indirectly. Operating a feed water heater at lower or higher capacity than the design condition has an effect on performance and ultimately the net unit heat rate. For that reason, the investigation of the thermal performance of these heat exchangers is the main aim of this study.

It is well known that the ideal cycle for a simple steam power plant is the Rankine cycle. This simple ideal cycle does not involve any internal irreversibilities and consists of: isentropic compression in a pump, constant pressure heat addition in a boiler, isentropic expansion in a turbine and constant pressure heat rejection in a condenser as shown in Fig. (1). However, heat is transferred to the working fluid in the boiler at a relatively low temperature. This makes the average heat addition temperature relatively low and thus the cycle efficiency (Cengel, 2002). In actual steam power plants, a practical regeneration process is accomplished by bleeding, or extracting, steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feed water instead through a heat exchanger known as feed water heater as shown in Fig.(2). This process has many operational advantages such as improving overall cycle efficiency, increasing the boiler tube life and reducing the outage due to tube leakages because of the reduction in the metal temperature of boiler tube, reducing the turbine blade damage due to extractions from last stages of turbine act as moisture extractor. Additionally, using feed water heating reduces the cost of generated electrical power and finally, reducing the rate of smoke and energy rejection to the environment (Cengel, 2002; Kumar & Kumar, 2014; Bode & Gore, 2016).

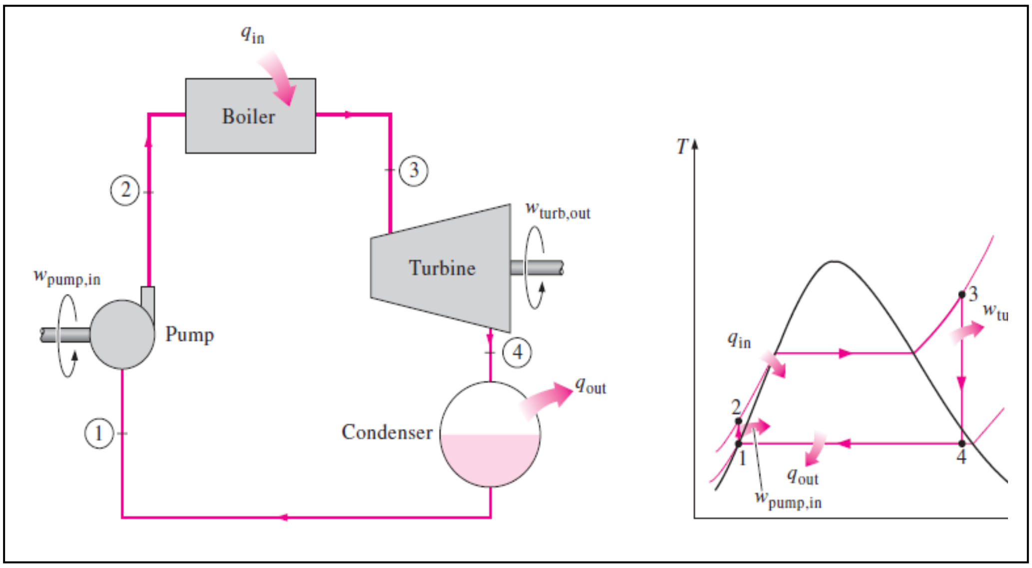

Figure 1: The simple ideal Rankine cycle (Cengel, 2002).

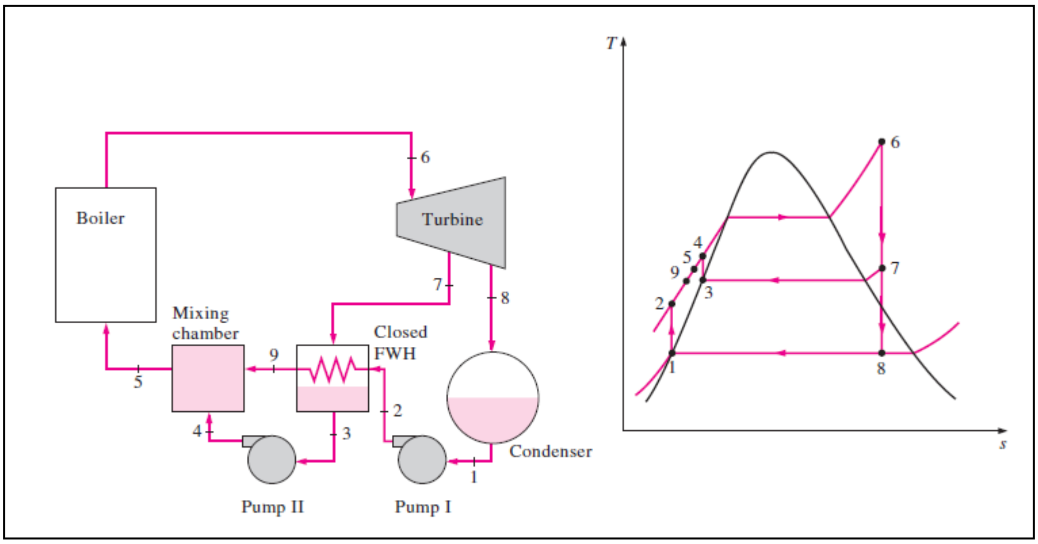

Figure 2: The ideal regenerative Rankine cycle with a single closed feed water heater (Cengel, 2002).

In Literature, many studies were carried out to investigate the performance of low and high pressure regenerative feed water heaters in different thermal power plants. For instance, the effect of fouling on performance evaluation of multi-zone feed water heaters was investigated by (Antar & Zubair, 2007). Thermodynamic analysis was carried out to predict mass flow rate of steam, coal consumption rate, and thermal efficiency of an actual 210 MW coal-fired power plant located in India (Kumar et al, 2014). The study also included an economic analysis. Modeling and simulation of low pressure closed feed water heater using Flownex SE software was performed and studied by (Byregowda et al, 2014). The study emphasized on one dimension flow inside the feed water drum and also provided detailed information, where measurements are difficult to make by experiments alone. The performance and the behavior of feed water heater under various operation conditions including Off design conditions was studied by (Kushwaha & Koshti, 2015). The effects of the feed water heaters on the performance of coal fired power plants were studied by (Devandiran et al, 2016). The study revealed that by adding the feed water heaters in power plant cycle, the overall efficiency of the power plant was increased by 2.4 %. An evaluation of the performance of the different components of the regenerative system including feed water heaters in 210 MW power plant was performed by (Kumar & Buckshumiyanm, 2017). The off design analysis and the evaluation of the performance of the high and low pressure feed water heaters on a 3×135 MW Coal Fired Power Plant were conducted by (Almedilla et al, 2018). Recently, technical and economical evaluation of full repowering of a steam power plant with a capacity of 320 MW in Iran was performed using different scenarios by (Naserabad et al, 2019). The results of the study revealed that full repowering plan with parallel feed water heating was the best alternative in terms of energy and exergy efficiencies and the price of generating electricity.

The rest of the paper is organized as follows: the objective, methodology and the summary of the main equations that utilized in thermal calculations are presented in sections two and three respectively. Section four presents the operational data of the considered feed water heater. Section five shows the results of the thermal analysis and in Section six, the conclusion is derived.

## 2. Objective and Methodology

In this paper a MATLAB code is developed to predict the performance of high pressure closed type feed water heaters under different flow conditions. The code is based on theoretical models and empirical correlations, which are available in the literature. First, the program will be verified then utilized to predict the performance of an actual high pressure three zones closed type feed water heater. Figure (3) illustrates a three zones closed type feed water heater. The three zones are separate areas within the shell in a feed water heater. These areas are known as de-superheating zone, condensing zone and sub-cooling zone.

The most widely accepted procedure for calculating the performance of closed feed water heaters in the power industry is called ASME PTC 12.1, which is developed by the American

Society of Mechanical Engineers (ASME PTC 12.1, 2000). This procedure has been adopted for the thermal calculations in the current work.

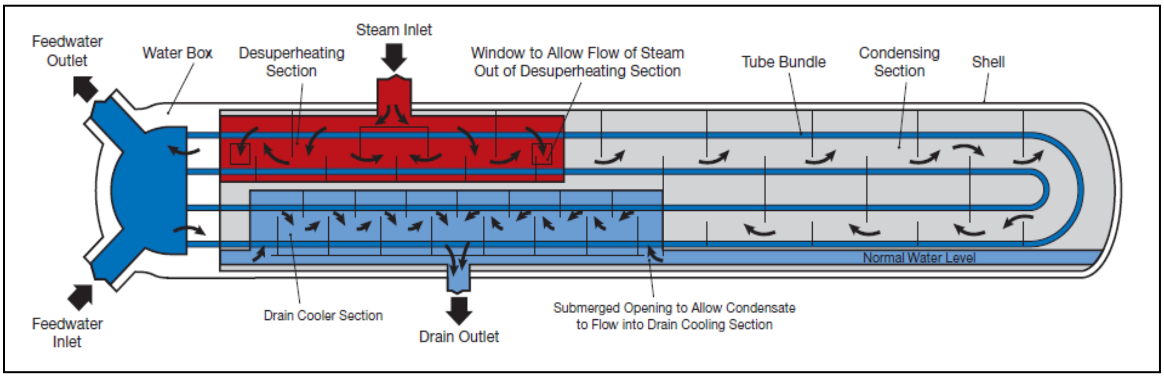

Figure 3: Three zones - closed type feed water heater (Hite, 2010).

## 3. Thermal Analysis

Common assumptions and general considerations are accounted for the thermal analysis of the feed water heaters. These can be summarized as follows: feed water heaters are modeled as steady-flow heat exchangers as they usually operate for long periods of time with no change in their operating conditions; changes in fluid velocity and elevation within the feed water heaters are insignificant. Therefore, the kinetic and potential energy changes are negligible; within a specified range of operating temperature, the specific heat of fluids can be treated as a constant at some average value; the axial heat conduction along the pipe is usually extremely small and can be considered negligible; finally, the outer surface of the feed water heater is assumed to be perfectly insulated (Cengel, 2002).

### 3.1 Heat Balance Equation:

$$q = \dot{m}_c c_{pc} \left(T_{c,out} - T_{c,in}\right) = \dot{m}_h c_{ph} \left(T_{h,in} - T_{h,out}\right) \tag{1}$$

where $q$ is the total heat transfer in (Watt), the subscripts $c$ and $h$ stand for cold and hot fluids, respectively, $\dot{m}_c$ and $\dot{m}_h$ are mass flow rates, $c_{pc}$ and $c_{ph}$ are specific heats, $T_{c,in}$ and $T_{h,in}$ are inlet temperatures, $T_{c,out}$ and $T_{h,out}$ are outlet temperatures.

In the case of one of the working fluids undergoes a phase-change process such as the fluid in a condenser, the rate of heat transfer is expressed as:

$$q = \dot{m} h_{fg} \tag{2}$$

where $\dot{m}$ is the rate of condensation or evaporation of the fluid and $h_{fg}$ is the enthalpy of vaporization of the fluid at the specified temperature or pressure (Cengel, 2002).

## 3.2 The Overall Heat Transfer coefficient and Fouling Factors:

The heat transfer rate between the two working fluids can be expressed as:

$$q = \frac{\Delta T}{R_T} = UA\Delta T \tag{3}$$

Where $R_T$ is the total thermal resistance in the path of heat flow from the hot fluid to the cold one, $U$ is the overall heat transfer coefficient, whose unit is ( W/m$^2$.C ), $A$ is the surface area of the heat transfer in (m$^2$), and $\Delta T$ is a suitable mean temperature difference across the feed water heater. By canceling $\Delta T$, equation (2) reduces to:

$$UA = \frac{1}{R_T} \tag{4}$$

In the most practical applications the overall heat transfer coefficient is computed based on the outside area of the tube. Beside that the heat-transfer surfaces become coated with various deposits after a period of operation, or the surfaces may become corroded as a result of the interaction between the fluids and the material used for construction of the feed water heater. This effect is usually represented by a fouling factor $R_f$ (Cengel, 2002).

Therefore, for an unfinned surface shell and tube feed water heater the overall heat transfer coefficient can be expressed as:

$$\frac{1}{UA} = R_T = \frac{1}{h_i A_i} + \frac{R_{f_i}}{A_i} + \frac{\ln(r_o/r_i)}{2\pi kl} + \frac{R_{f_o}}{A_o} + \frac{1}{h_o A_o} \tag{5}$$

where $h_i$ and $h_o$ are the convection heat transfer coefficients on tube side and shell side respectively, $R_{fi}$ and $R_{fo}$ are the fouling resistance on the tube side and shell side respectively, $r_i$

and $r_o$ are the the inner and outer radius of the tube, $k$ is the thermal conductivity of the tube, and $l$ is the length of the tube.

## 3.3 The Logarithmic Mean Temperature Difference (LMTD):

The logarithmic mean temperature difference $\Delta T_{LM}$ is obtained by tracing the actual temperature profile of the fluids along the heat exchanger. It is an exact representation of the average temperature difference between the hot and cold fluids. It can be written in general form for the simple double pipe heat exchanger as follows:

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{ln(\Delta T_1/\Delta T_2)} \qquad (6)$$

Where $\Delta T_1$ and $\Delta T_2$ represent the temperature difference between the two fluids at the two ends of the heat exchanger.

In case of using different types of heat exchangers as in the present work, where the shell and tube feed water heater is considered in the calculations, a correction factor should be applied to the LMTD for the heat transfer calculation. The correction factor $F$ for common shell and tube heat exchanger configurations is given in Fig. (4).

The equation of heat transfer then takes the following form:

$$q = UAF\Delta T_{LM} \qquad (7)$$

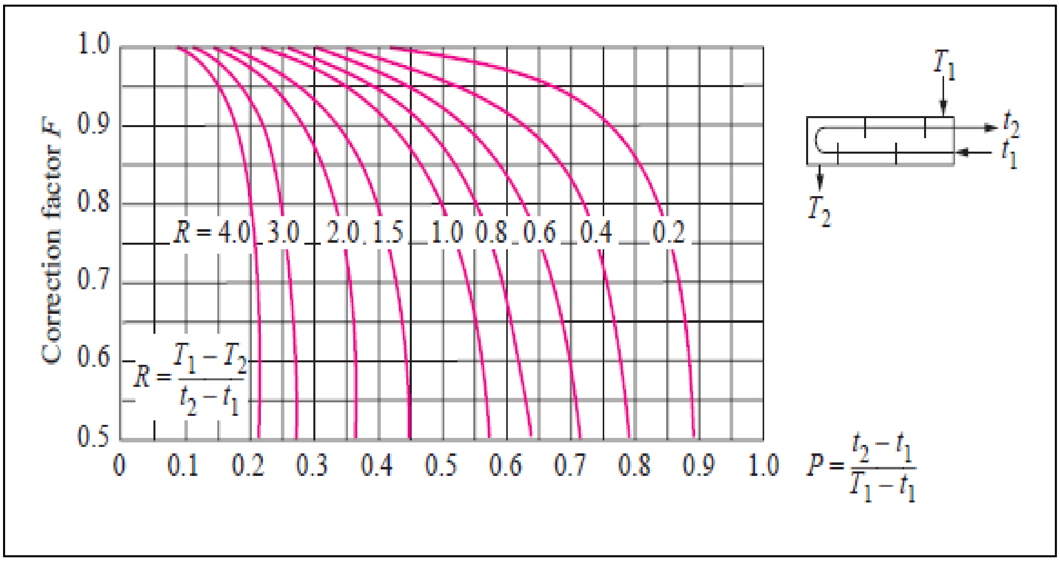

Figure 4: Correction-factor for an exchanger with one shell pass and two, four, or any multiple of tube passes (Cengel, 2002).

## 3.4 The Effectiveness-NTU Method:

This method is the most suitable scheme to determine the performance of a specified heat exchanger. The heat transfer effectiveness "$\varepsilon$" is a dimensionless parameter, which can be defined as:

$$\varepsilon = \frac{Actual\ heat\ transfer\ rate}{Maximum\ possible\ heat\ transfer\ rate} = \frac{q}{q_{max}} \tag{8}$$

The maximum possible heat transfer for the exchanger is expressed as:

$$q_{max} = C_{min}\left(T_{h,in} - T_{c,out}\right) \tag{9}$$

The effectiveness relation for counter flow heat exchangers is presented as follows (Cengel, 2002):

$$\varepsilon = \frac{1 - exp[(-UA/C_{min})(1 - C_{min}/C_{max})]}{1 - (C_{min}/C_{max})exp[(-UA/C_{min})(1 - C_{min}/C_{max})]} \tag{10}$$

where $C_{min}$ is the *smaller* heat capacity ratio and $C_{max}$ is the larger one, and it makes no difference whether $C_{min}$ belongs to the hot or cold fluid.

The quantity ($UA/C_{min}$) is called the number of transfer units *NTU* and is expressed as:

$$NTU = \frac{UA}{C_{min}} \tag{11}$$

The effectiveness relation for condensation process is expressed as:

$$\varepsilon = 1 - e^{-NTU} \tag{12}$$

## 3.5 Terminal Temperature Difference (TTD):

This parameter is defined as the saturation temperature of the extracted steam ($T_{sat}$) minus the feed water outlet temperature ($T_{Fw,out}$). It provides feedback on the performance of the feed water heater relative to heat transfer. An increase in TTD is a sign of a reduction in heat transfer whereas a decrease in TTD indicates an improvement (Hite, 2010; Kushwaha & Koshti, 2015).

$$TTD = T_{sat} - T_{FW,out} \tag{13}$$

## 3.6 Drain Cooler Approach Temperature (DCA):

This parameter is used to deduce feed water heater levels based on the temperature difference between the drain cooler outlet ($T_{drains}$) and the feed water inlet ($T_{Fw,in}$). An increasing DCA temperature difference points to the level is decreasing; while, a decreasing DCA specifies a rise in level (Hite, 2010; Kushwaha & Koshti, 2015).

$$DCA = T_{drains} - T_{FW,in} \tag{14}$$

## 3.7 Feed Water Temperature Rise (TR):

Feed water temperature rise is the difference between the feed water outlet temperature ($T_{Fw,out}$ ) and the feed water inlet temperature ($T_{Fw,in}$). A properly performing heater should meet the design specifications (Hite, 2010; Kushwaha & Koshti, 2015).

$$TR = T_{FW,out} - T_{FW,in} \tag{15}$$

## 4. The Case Study and Technical Data of The Feed Water Heater

The working cycle for the considered thermal power plant with feed heating system, which is located in Amravati, Maharashtra, India is shown in Fig. (5) (Bode & Gore, 2016). The feed heating system includes two horizontal high pressure heaters (Number 5 and 6 in Fig. 5). The high pressure heater No. 5 (HPH5) is considered as a case study in this work, and for the sake of simplicity, the drip from heater 6 to heater 5 is ignored.

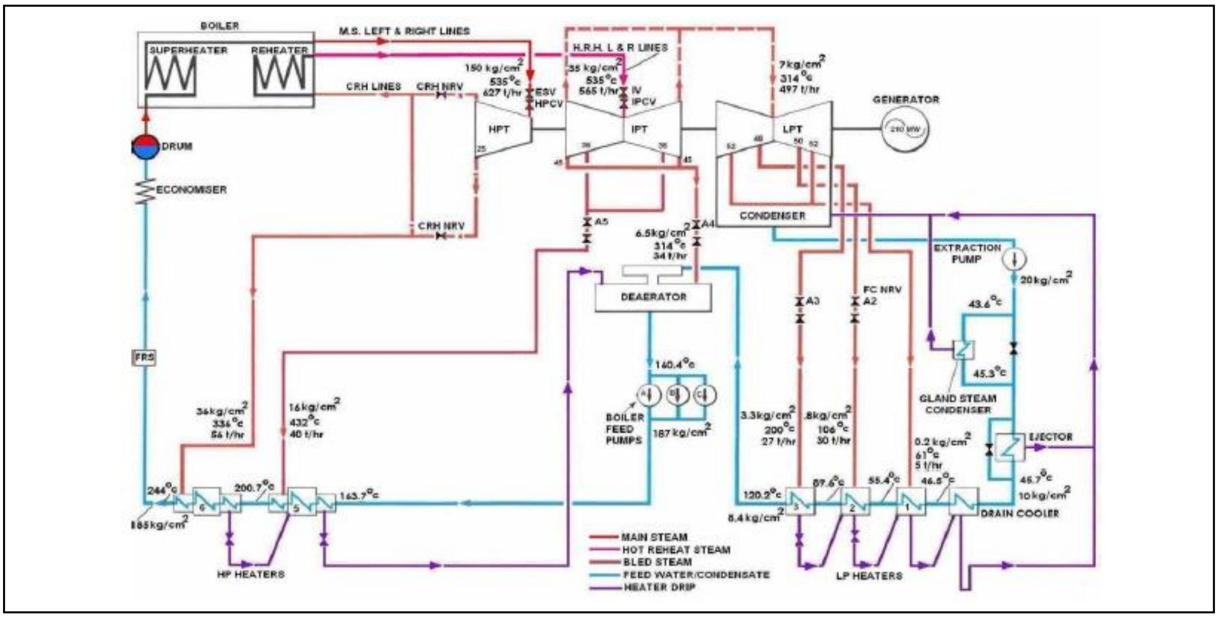

Figure 5: Operating cycle of thermal power plant (Bode & Gore, 2016).

Table (1) provides the technical data for the feed water heater (HPH5), which is a , horizontal coil type with integral de superheating, condensing and drain cooling zones (Bode & Gore, 2016).

Table 1: Technical data of high pressure heater No.5 (HPH5) (Bode & Gore, 2016).

| **Physical Parameter** | **Units** | **Value** |
|---|---|---|
| Pressure of extraction steam | MPa | 1.57 |
| Enthalpy of extraction steam | KJ/kg | 2794 |
| Saturation temperature | $^{O}$C | 201.4 |
| Quantity of extraction steam | Kg/s | 11.11 |
| Steam temperature | $^{O}$C | 432 |
| Enthalpy of steam | KJ/kg | 3327 |
| Drain temperature | $^{O}$C | 172.2 |
| Enthalpy of Drain | KJ/kg | 728.73 |
| Quantity of Drain flow | Kg/s | 45.77 |
| Temperature of feed water at inlet | $^{O}$C | 163.7 |
| Temperature of feed water at outlet | $^{O}$C | 201 |
| Quantity of feed water | Kg/s | 221.11 |

## 5. Results and Discussion

## 5.1 Determination of the Thermal Profile of Working Fluids within The Feed Water Heater:

The first task is to determine the thermal profile of both the steam and feed water within the heater. The second task is the estimation of the surface area of each zone in the considered feed water heater. Based on the data from Table (1) and using LMTD method, the intermediate temperatures, heat transfer area of each zone, the effectiveness of each zone, TTD and DCA are computed and the results are presented in Table (2) and Fig.(6). These data are considered as a reference data for further performance studies.

Table 2: Results from LMTD method to calculate Area percentage, Effectiveness, inlet and outlet temperature for each zone:

| De-superheating Zone | | Condensing Zone | | Sub-cooling Zone | |
|---|---|---|---|---|---|
| Shell | Tube | Shell | Tube | Shell | Tube |
| $T_{steam} = 432\,^{o}C$ | $T_{fwo} = 201\,^{o}C$ | $T_{sat} = 201.4^{o}C$ | $T_{fw2} = 194.44^{o}C$ | $T_{sat} = 201.4\,^{o}C$ | $T_{fw1} = 172.8\,^{o}C$ |
| $T_{sat} = 201.40\,^{o}C$ | $T_{fw2} = 194.44^{o}C$ | $T_{sat} = 201.4^{o}C$ | $T_{fw1} = 172.8\,^{o}C$ | $T_{Dr} = 194.4^{o}C$ | $T_{fwin} = 163.7\,^{o}C$ |
| Percentage of Area ($A_1$) = 4.75 % | | Percentage of Area ($A_2$) = 73.13% | | Percentage of Area ($A_3$) = 22.12% | |
| Effectiveness ($\varepsilon_1$) = 97.30% | | Effectiveness ($\varepsilon_2$) = 77.61% | | Effectiveness ($\varepsilon_3$) = 77.45% | |

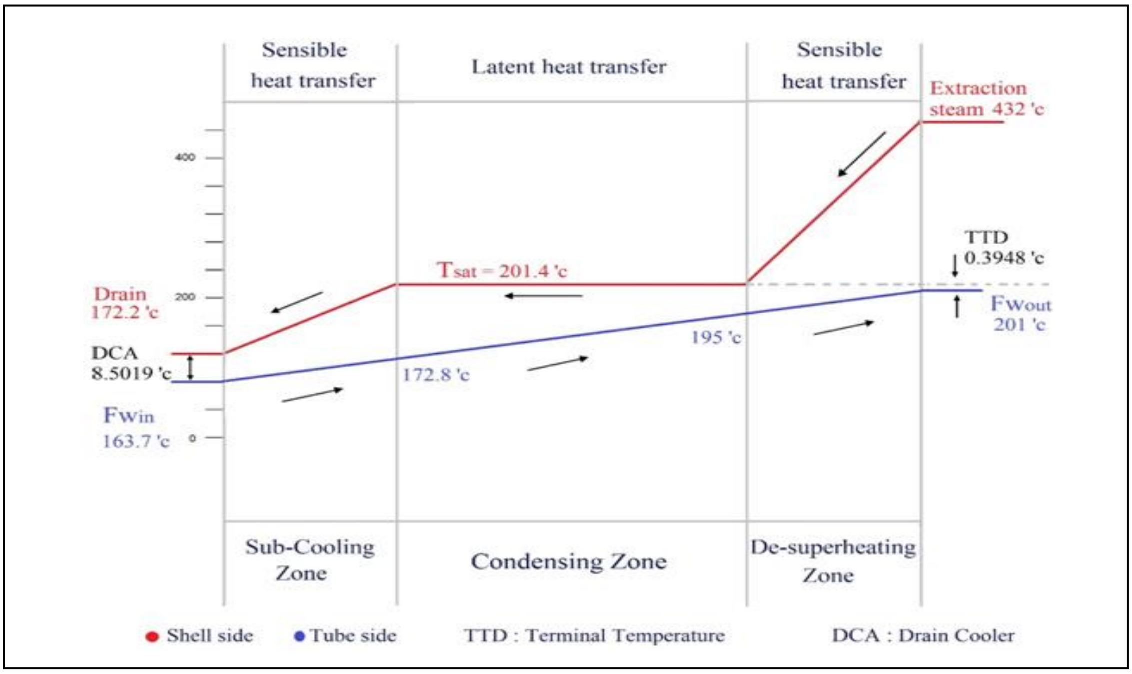

Figure 6: Thermal profile in different zones of the feed water heater.

## 5.2 Verification of The Effectiveness – NTU Code:

This task has been performed using the effectiveness – NTU method, where the program has been verified by comparing the present results with the previous data that were obtained using the LMTD method. Excellent agreement between both results are achieved. Summary of the verification results are illustrated in Table (3).

Table 3: Results from Effectiveness - NTU method to calculate the effectiveness, inlet and outlet temperatures for each zone (in this case $T_{Dr}$ and $T_{fwo}$ are unknown)

| Sub-cooling Zone | | Condensing Zone | | De-superheating Zone | |
|---|---|---|---|---|---|
| Shell | Tube | Shell | Tube | Shell | Tube |
| $T_{sat}$ = 201.4 °C | $T_{fw1}$ = 172.8 °C | $T_{sat}$ = 201.4 °C | $T_{fw2}$= 195 °C | $T_{steam}$ = 432 °C | $T_{fwo}$ = 163.7 °C |
| $T_{Dr}$ = 172.2°C | $T_{fwin}$ = 163.7 °C | $T_{sat}$ = 201.4 °C | $T_{fw1}$= 172.8 °C | $T_{sat}$ = 201.4 °C | $T_{fw2}$= 195 °C |
| Effectiveness ($\varepsilon_3$) = 77.45 % | | Effectiveness ($\varepsilon_2$) = 77.61 % | | Effectiveness ($\varepsilon_1$) = 97.30 % | |

## 5.3 Effect of Feed Water Flow Rate at Constant Steam Inlet Parameters:

After the verification of the code, a performance test using the effectiveness – NTU method is performed to investigate the effect of feed water mass flow rate ($\dot{m}_{fw}$) on the temperature distribution, effectiveness, TTD, DCA, and TR of the feed water heater at constant steam inlet temperature, pressure and flow rate.

The results of the computations are presented in Table (4). It can be seen that increasing the feed water mass flow rate by 5% to 10% leads to decrease in the intermediate temperatures, outlet temperature of feed water, DCA and TR. This is associated with the increase in TTD. Also, it can be observed that increasing the feed water mass flow rate by 10% causes a slight increase in the effectiveness of the sub-cooled drain zone and slight decrease in the effectiveness of the condensing zone, whereas the effectiveness of the de-superheating zone almost remains constant.

On the other hand, it can be seen that the decrease in the feed water mass flow rate by 10% leads to increase in the intermediate temperatures, outlet temperature of feed water, DCA, and TR. However, the terminal temperature difference TTD decreases with the decrease of the mass flow rate. Decreasing the feed water mass flow rate by 10% causes a slight decrease in the effectiveness of the sub-cooled drain zone and slight increase in the effectiveness of the condensing zone, whereas the effectiveness of the de-superheating zone almost remains constant.

Table 4: Results from NTU method with changes in mass flow rate of feed water heater:

| $\dot{m}_{fw}$ *kg/s* | $T_{fwin}$ °C | $T_{fw1}$ °C | $T_{fw2}$ °C | $T_{fwo}$ °C | *TTD* °C | *TR* °C | *DCA* °C | $\varepsilon_1$ % | $\varepsilon_2$ % | $\varepsilon_3$ % |
|---|---|---|---|---|---|---|---|---|---|---|
| 198.99 | 163.7 | 173.81 | 198.47 | 205.15 | -3.752 | 41.4521 | 8.744 | 97.28% | 81.04% | 76.81% |
| 210.05 | 163.7 | 173.27 | 196.64 | 202.96 | -1.569 | 39.2695 | 8.616 | 97.29% | 79.31% | 77.15% |
| **221.11** | **163.7** | **172.80** | **195.00** | **201.00** | **0.3948** | **37.3052** | **8.502** | **97.30%** | **77.61%** | **77.45%** |
| 232.16 | 163.7 | 172.36 | 193.51 | 199.22 | 2.1704 | 35.5296 | 8.399 | 97.31% | 75.96% | 77.72% |
| 243.22 | 163.7 | 171.97 | 192.15 | 197.61 | 3.7861 | 33.9139 | 8.306 | 97.32% | 74.35% | 77.97% |

## 5.4 Effect of Inlet Temperature of Feed Water:

The purpose of this performance test is to investigate the effect of the inlet feed water temperature on the thermal performance of the heater. Accordingly the inlet feed water temperature is changed by a percentage of approximately 9.5% and intermediate temperatures, TTD, DCA, TR and the outlet feed water temperature $T_{fwo}$ have been computed. The results are presented in Table (5) and Fig. (7). Table (5) presents the values of outlet feed water temperature $T_{fwo}$, TTD, DCA and TR as a function in the inlet temperature of the feed water. One can notice that by decreasing the inlet temperature, all the performance parameter values increase except the $T_{fwo}$ and by increasing the inlet temperature, all the performance values decrease with the exception of the $T_{fwo}$. Figure (7) shows a plot of the temperature distributions along the feed water heater as a function in the feed water inlet temperature. It is observed that the outlet temperature of feed water $T_{fwo}$ increases with the increase in the inlet temperature of feed water $T_{fwin}$. Whereas all other performance temperatures TTD, TR, and DCA decrease with the increase in the inlet feed water temperature.

Table 5: Results from the effectiveness-NTU method with changes in the inlet temperature of the feed water:

| Feed Water Inlet Temperature | Feed Water Outlet Temperature | Performance Parameters | | |
|---|---|---|---|---|
| $T_{fwin}$ (°C) | $T_{fwo}$ (°C) | *TTD* (°C) | *TR* (°C) | *DCA* (°C) |
| 150.00 | 190.58 | 10.8113 | 40.588 | 11.5914 |
| 158.00 | 196.69 | 4.7291 | 38.6709 | 9.7879 |
| **163.70** | **201.00** | **0.3948** | **37.3052** | **8.5019** |
| 170.00 | 205.77 | - 4.3700 | 35.77 | 7.0812 |
| 176.50 | 210.69 | - 9.2962 | 34.1962 | 5.6153 |

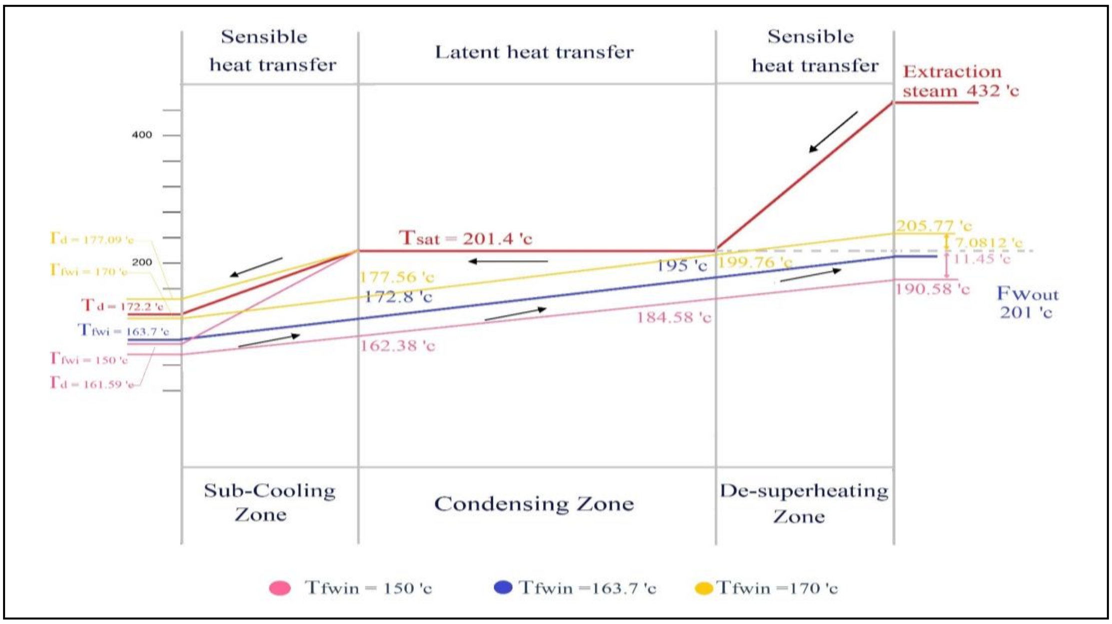

Figure 7: Thermal profile in different zones of the feed water heater with different inlet temperature for feed water.

## 6. Conclusions

In this paper, the prediction and analysis of the thermal performance of three zones feed water heaters were performed successfully using computer. A MATLAB code was developed based on the well known LMTD and Effectiveness-NTU methods. The program was tested and verified then applied to study a practical case study. The following conclusions are obtained:

- The effectiveness of the de-superheating zone is the highest among the three zones of the feed water heater. The effectiveness of condensing and sub-cooling zones may be lower, however the surface area of the condensing zone is the largest and the area of the sub-cooling zone comes the second.

- The feed water temperature rise (TR) and drain cooling approach (DCA) are inversely proportional to mass flow rate of feed water heater, whereas the terminal temperature difference (TTD) is directly proportional to mass flow rate of feed water heater.
- The effectiveness of condensing zone is inversely proportional to mass flow rate of feed water heater and the effectiveness of sub-cooling zone is directly proportional to mass flow rate of feed water heater. However, the mass flow rate of feed water heater has almost no effect on the effectiveness of the de-superheating zone.
- The feed water temperature rise (TR), terminal temperature difference (TTD), and drain cooling approach (DCA) are inversely proportional to the inlet temperature of feed water heater.

Finally, the following recommendations are suggested to extend this work:

- Extensions to the developed program to include data base for the physical properties and the enthalpy of steam for a wide range of temperatures and pressures. This allows extending the study to include other parameters such as changing steam side operating parameters.
- Extending the study to include other feed water heaters especially, those located in thermal power plants in Libya.

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