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The Effect of the Direct Contact Heat Exchanger on Steam Power Plant

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***Abstract*—**An actual power plant, which is the power plant of Iron and Steel Factory at Misurata city in Libya, has been modeled using Matlab in order to compare its results to the actual results of the actual cycle. This paper concentrates on two factors:

a- The comparison between exergy losses in the actual cycle and the modeled cycle.
b- The effect of extracting pressure on temperature water at boiler inlet.

Closed heat exchangers used in this plant have been substituted by open heat exchangers in the current study of the modeled power plant and the required changes in the pressure have been considered. In the following investigation the two points mentioned above are taken in consideration.

***Keywords*—**Steam Power Plant, Contact Heat exchanger, Exergy, Cycle Efficiency.

I. Introduction

REGENERATIVE cycle is one of the methods used in uplifting the thermal efficiency of vapor generating plants, where depletion is done via two types of heat exchangers (heaters) which are mostly closed indirect contact heat exchangers or closed types. In this type of exchangers a part of the steam passing through the turbine is depleted in order to pass in tubes existed inside the heater containing the feeding water that comes out of the condenser. The steam is not mixed with the feeding water and passes out after passing in the pipes from below of the heater joining once more the cycle either in the direction of feeding water movement or in reverse direction at the condenser where shell and tube heat exchangers usually used.

The other type is the direct contact heat exchanger or open heater where direct mixing between apart of the steam passing the turbine and feeding water coming out from the condenser.

The wide use of open heat exchangers is because of their great features comparing to closed exchangers. Currently; factor of main parts efficiency, steam generator and turbine is much closer from its final value, so any efficiency uplifting of the heat/thermal station can be obtained by developing the auxiliary techniques which includes the heat exchangers. The closed exchangers are used widely despite their following defects:

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1- The Reaction that occurs between steam, water and the metal of heat exchanger which leads to contamination fluid in the oxidized metal.
2- Heating water does not reach to saturation degree due to thermal resistance in pipe's metal of heat exchanger metal, also because existence of gasses in water steam which are incondensable.
3- A complex structure, high price and the continual failure in the pipes.

All these reasons, and others; lead to the use of an open heat exchanger, this recent usage is a reason of absence of what is called operation experience, besides; there are no enough studies available about these exchangers. Additionally, use of open heat exchangers is relevant to many operational problems, such as return of steam-water mixture to the lower part of turbine which results in raising the pressure inside the exchanger, furthermore, the decrease of feeding water flow level, as well as the high sensitivity of pressure changes when heat exchangers directly connected.

This study has been carried to evaluate the performance of the both types of heat exchangers, opened and closed heat exchangers in terms of the Exergy losses and the increase in water temperature at the boiler inlet.

II. Methodology

The study based on a mathematical analysis of the equations of heat transfer and thermodynamic that applied on two cycles, an actual cycle (Iron & Steel Co. power steam plant - Misurata) and a proposed cycle. In order to evaluate the open heat exchangers, the actual cycle has been modified by using the open heat exchangers instead of the closed heat exchangers. However, as result of the modifications, the pressure values in the cycle are changed. Fig. 1 below illustrates the cycle of the actual power plant [1].

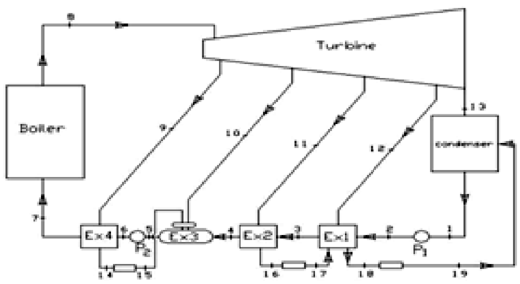


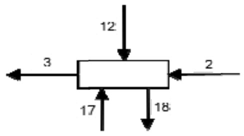
Fig. 1 Actual cycle vapor power plant

In order to analyze the actual cycle numerically, mass flow rate equations can be given as:

$$q = \dot{m} \times (h_{13} - h_1) = \dot{m}_{water} \times c_w \times \Delta t \quad (1)$$

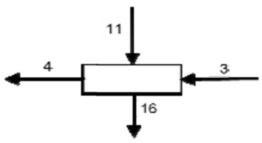
Where: q is heat transfer in KJ/S, m is mass flow rate in KJ/S, h is enthalpy in KJ.

By applying the thermal balance equation on first closed heat exchanger



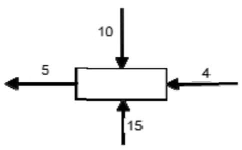
$$\begin{aligned}&\left[\left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_2\right) + (\dot{m}_{12} \times h_{12}) + (\dot{m}_{11} \times h_{17})\right] = \\ &\left[\left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_3\right) + \left((\dot{m}_{12} + h_{11}) \times h_{18}\right)\right]\end{aligned} \quad (2)$$

By applying the thermal balance equation on second closed heat exchanger



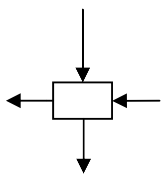
$$\begin{aligned}&\left[\left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_3\right) + (\dot{m}_{11} \times h_{11})\right] = \\ &\left[\left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_4\right) + (\dot{m}_{11} \times h_{16})\right]\end{aligned} \quad (3)$$

By applying the thermal balance equation on open heat exchanger



$$\begin{bmatrix}\left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_4\right) + (\dot{m}_{10} \times h_{10}) \\ + (\dot{m}_9 \times h_{15})\end{bmatrix} = \left[(\dot{m}_8 \times h_5)\right] \quad (4)$$

By applying the thermal balance equation on third closed heat exchanger



$$\left[(\dot{m}_8 \times h_6) + (\dot{m}_9 \times h_9)\right] = \left[(\dot{m}_8 \times h_7) + (\dot{m}_9 \times h_{14})\right] \quad (5)$$

Now by applying the heat transfer equation on the heat exchanger.

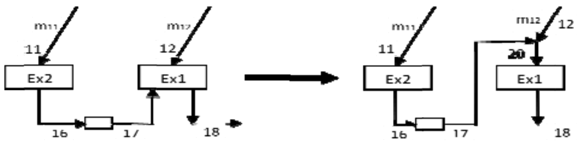
$$\dot{Q} = (\dot{m}_i) \times (h_e - h_i) = U \times A \times \Delta T \quad (6)$$

$$(\dot{m}_i) \times (h_e - h_i) - U \times A \times \Delta T = 0 \quad (7)$$

Where

$$\Delta T = \frac{\Delta T_i - \Delta T_e}{\ln \dfrac{\Delta T_i}{\Delta T_e}}$$

In closed heat exchanger No. 1



Where

$A = 80m^2$ $\quad U = 6.25\, kW/m.c$

$$\left[\left((\dot{m}_{11} + \dot{m}_{12}) \times (h_{20} - h_{18})\right)\right] = \left[\left[\left(500 \times \left(\frac{(t_{20} - t_2) - (t_{18} - t_3)}{\ln\left(\dfrac{t_{20} - t_2}{t_{18} - t_3}\right)}\right)\right)\right]\right] \quad (8)$$

$$\left[\left(\dot{m}_{11} \times (h_{11} - h_{16})\right)\right] = \left[\left[\left(915.057 \times \left(\frac{(t_{11} - t_3) - (t_{16} - t_4)}{\ln\left(\dfrac{t_{11} - t_3}{t_{16} - t_4}\right)}\right)\right)\right]\right] \quad (9)$$

$$\left[\left(\dot{m}_9 \times (h_9 - h_{14})\right)\right] = \left[\left[\left(589.12 \times \left(\frac{(t_9 - t_6) - (t_{14} - t_7)}{\ln\left(\dfrac{t_9 - t_6}{t_{14} - t_7}\right)}\right)\right)\right]\right] \quad (10)$$

III. Newton Method

Non-linear equations can be solved by several numerical methods, but Newton method is wide used. This method can be illustrated as follows:

$$f_1(x, y) = 0$$
$$f_2(x, y) = 0$$

$$\begin{bmatrix} x_{n+1} \\ y_{n+1} \end{bmatrix} = \begin{bmatrix} x_n \\ y_n \end{bmatrix} - \begin{bmatrix} fx_1 & fy_1 \\ fx_2 & fy2 \end{bmatrix}^{-1} \times \begin{bmatrix} f1 \\ f2 \end{bmatrix} \quad (11)$$

Where

$$fx_1 = \frac{\partial f_1}{\partial x} \;,\; fy_1 = \frac{\partial f_1}{\partial y} \;,\; fx_2 = \frac{\partial f_2}{\partial x} \;\&\; fy_2 = \frac{\partial f_2}{\partial y}$$

Where the functions are:

$$F(\dot{m}_{12}, \dot{m}_{12}, \dot{m}_{12}, \dot{m}_{12}, h_3, h_4, h_7) = 0 \quad (12)$$

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$$F_1 = \left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_2\right) + (\dot{m}_{12} \times h_{12}) + (\dot{m}_{11} \times h_{17}) - \left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_3\right) - \left((\dot{m}_{12} + h_{11}) \times h_{18}\right) = 0 \quad (13)$$

$$F_2 = \left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_3\right) + (\dot{m}_{11} \times h_{11}) - \left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_4\right) - (\dot{m}_{11} \times h_{16}) = 0 \quad (14)$$

$$F_3 = \left((\dot{m}_8 - \dot{m}_9 - \dot{m}_{10}) \times h_4\right) + (\dot{m}_{10} \times h_{10}) + (\dot{m}_9 \times h_{15}) - (\dot{m}_8 \times h_5) = 0 \quad (15)$$

$$F_4 = (\dot{m}_8 \times h_6) + (\dot{m}_9 \times h_9) - (\dot{m}_8 \times h_7) - (\dot{m}_9 \times h_{14}) = 0 \quad (16)$$

$$F_5 = \left((\dot{m}_{11} + \dot{m}_{12}) \times (h_{20} - h_{18})\right) - \left(500 \times \left(\frac{(t_{20} - t_2 - t_{18} + t_3)}{\ln\left(\frac{(t_{20} - t_2)}{(t_{18} - t_3)}\right)}\right) \right) = 0 \quad (17)$$

$$F_6 = \left(\dot{m}_{11} \times (h_{11} - h_{16})\right) - \left(915.057 \times \left(\frac{(t_{11} - t_3 - t_{16} + t_4)}{\ln\left(\frac{(t_{11} - t_3)}{(t_{16} - t_4)}\right)}\right) \right) = 0 \quad (18)$$

$$F_7 = \left(\dot{m}_9 \times (h_9 - h_{14})\right) - \left(589.12 \times \left(\frac{(t_9 - t_6 - t_{14} + t_7)}{\ln\left(\frac{(t_9 - t_6)}{(t_{14} - t_7)}\right)}\right) \right) = 0 \quad (19)$$

By applying Newton method, we assume a symbol for each matrix in the above formula as follows [2]:

$$ss = aa - \left((jj)^{-1} \times gg\right) \quad (20)$$

Where

$$ss = \begin{bmatrix} ss_1 \\ ss_2 \\ ss_3 \\ ss_4 \\ ss_5 \\ ss_6 \\ ss_7 \end{bmatrix} \quad aa = \begin{bmatrix} \dot{m}_{12} \\ \dot{m}_{11} \\ \dot{m}_{10} \\ \dot{m}_9 \\ h_3 \\ h_4 \\ h_5 \end{bmatrix} \quad gg = \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \\ f_7 \end{bmatrix} \quad (21)$$

$$jj = \begin{bmatrix} f_1\dot{m}_{12} & f_1\dot{m}_{11} & f_1\dot{m}_{10} & f_1\dot{m}_9 & f_1h_3 & f_1h_4 & f_1h_7 \\ f_2\dot{m}_{12} & f_2\dot{m}_{11} & f_2\dot{m}_2 & f_2\dot{m}_9 & f_2h_3 & f_2h_4 & f_2h_7 \\ f_3\dot{m}_{12} & f_3\dot{m}_{11} & f_3\dot{m}_{10} & f_3\dot{m}_9 & f_3h_3 & f_3h_4 & f_3h_7 \\ f_4\dot{m}_{12} & f_4\dot{m}_{11} & f_4\dot{m}_{10} & f_4\dot{m}_9 & f_4h_3 & f_4h_4 & f_4h_7 \\ f_5\dot{m}_{12} & f_5\dot{m}_{11} & f_5\dot{m}_{10} & f_5\dot{m}_9 & f_5h_3 & f_5h_4 & f_5h_7 \\ f_6\dot{m}_{12} & f_6\dot{m}_{11} & f_6\dot{m}_{10} & f_6\dot{m}_9 & f_6h_3 & f_6h_4 & f_6h_7 \\ f_7\dot{m}_{12} & f_7\dot{m}_{11} & f_7\dot{m}_{10} & f_7\dot{m}_9 & f_7h_3 & f_7h_4 & f_7h_7 \end{bmatrix} \quad (22)$$

The actual cycle has been modified by replacing the closed heat exchangers by the open heat exchangers with considering the change in pressure values. The proposed cycle is shown in Fig. 2.

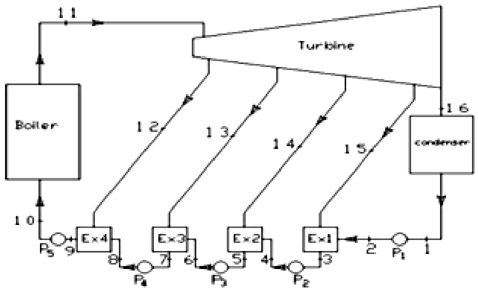

Fig. 2 Proposed cycle vapor power plant

IV. Results and Discussion

A. The comparison between exergy losses in actual cycle and modeled cycle.

Exergy analysis for the main components of the modeled and actual power plants has been done at ambient temperature 25 °C and pressure 101.134 kPa [3]. Simulation includes two options, one option is to consider the intermediate pressure as an equal segment between extracting stages. i.e.

$$\Delta p = \frac{(p_i - p_0)}{number\, of\ stages}$$

The other option is to consider the maximum efficiency of the cycle in calculation of extracting pressure values in the simulation [4].

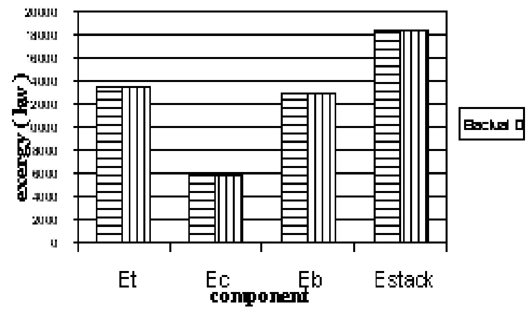

Fig. 3 Exergy losses of the components of the two cycles

Figs 3 and 4 show the exergy losses in the main components of the modeled and actual cycle at equal pressure segment in the steam turbine (option 1).

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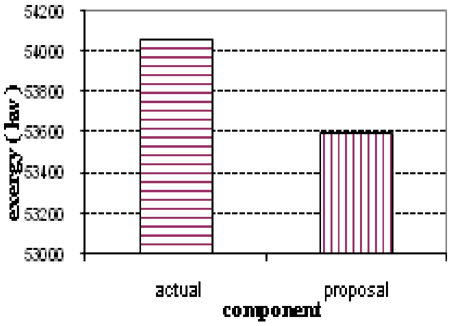

Fig. 4 Exergy losses of the other components of the two cycles

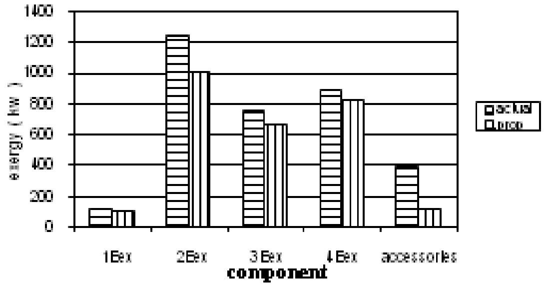

Fig. 5 Exergy losses of both cycles at the same extracting pressure

Fig. 5 shows a significant difference in exergy losses between the modeled and actual cycle, while Fig. 3 shows the exergy losses in the heat exchangers and Fig. 4 illustrates, that there is a slight difference in the other component.

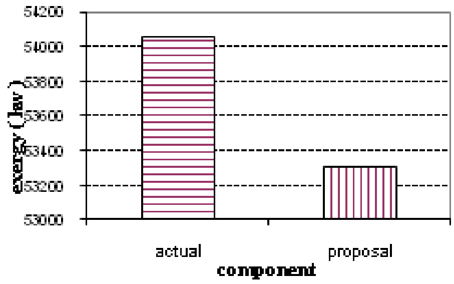

Fig. 6 Exergy losses of both cycles at the abstract extracting pressure

The simulation of the exergy losses for the whole components of the proposed and actual cycle under the equal segment pressure and optimum values, are given in Figs. 5 and 6. It can be seen that there is a significant difference between the proposed and actual cycle in the given figures. Maximum difference of about 8000 kW (14%) can be attained

B. The Effect of Extracting Pressure on Water Temperature at Boiler Inlet

Fig. 7 shows the effect of extracting pressure on water temperature. It can be seen from the diagram that when the extracting pressure increases, the water temperature will increase, but there is a slight improve in the modeled cycle comparing with the actual cycle.

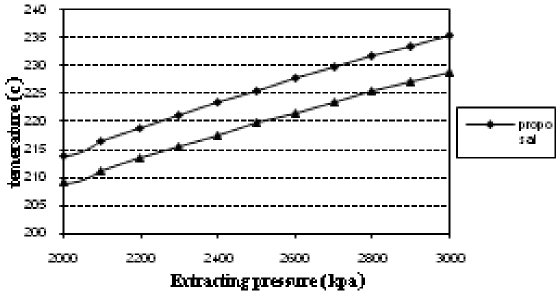

Fig. 7 The effect of the extracting pressure on the temperature of the inlet water to the boiler

However, it can be observed that water temperature at boiler inlet in modeled cycle increases by approximately (6-8) C^o compared with the actual cycle, which is considerable enhancement, which reflects in decreasing of the fuel consumption.

V. Conclusion

1- The implementation of exergy analysis on the components of the both cycles shows the decrease in exergy reaches 800 kW and it is greater by around 14% for the actual cycle.

2- The increase in extracting pressure lead to an increase in temperature of water inlet to the boiler by about (6 °C to 10 °C), which result in decrease in burning fuel.

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