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Exergetic Optimization of Inlet Cooling Water Temperature of Cross Flow Steam Condenser

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ABSTRACT : Condenser is one of the major components of the power plant, so it is necessary to operate the condenser efficiently under the various operating condition to increase the overall efficiency of the power plant. The present study focuses on the optimum inlet temperature of cooling water during condensation of saturated water vapor within a shell and tube condenser, through minimization of exergy destruction. First, the relevant exergy destruction is mathematically derived and expressed as a function of operating temperatures and mass flow rates of both vapor and coolant. The optimization problem is defined subjected to condensation of the entire vapor mass flow and it is solved using the program in MAT Lab. To find out the overall heat transfer coefficient, Bell Delaware method is used. Optimum inlet temperature of cooling water is calculated under the various operating condition of the steam condenser to determine the variation of exergy destruction and exergy efficiency at different condenser pressures, various mass flow rates of steam and cooling water with effect of atmospheric temperature is also considered. It is found that optimum cooling water temperature decrease with decrease of condenser pressure. As the upstream mass flow rate increase, the optimum coolant temperature and exergy efficiency decreases. It is found that there is no effect of atmospheric temperature on the optimum temperature of cooling water. With increase of cooling water flow rate, higher exergy efficiency can be obtained. Pressure drop is also increased with increase of cooling water flow rate, so pressured drop is also calculated and compared with permissible limit.

Keyword : Condenser, exergy efficiency, exergy destruction.

I. INTRODUCTION

The method of exergy analysis is well suited for furthering the goal of more effective energy resource use, for it enables the location, cause, and true magnitude of waste and loss to be determined which can be used in the design of new energy efficient thermal system or increasing the efficiency of existing systems. Exergy is a measure of the departure of the state of the system from that of the environment [1]. Exergy analysis also provides insights that elude a purely first-law approach. The exergy balance is similar to an energy balance but has fundamental difference that, while the energy balance is a statement of the conservation of energy, the exergy balance may be looked upon as a statement of the law of degradation of energy (irreversibility) in terms of second law analysis [2].

Exergoeconomic analysis of condenser was carried out by Ahmet Can and exergy losses of heat exchanger, investment and operation expenses related to this were determined, with functions of steam mass flow rate and cooling water exit temperature at constant values of thermal power, cold water mass flow rate and temperature [3]. In addition to this Haseli *et al.*, carried out the analysis of the shell and tube condenser with respect to exergy and evaluated the optimum cooling water temperature during condensation of saturated water vapour within a shell and tube condenser, through minimization of exergy destruction and its expression as a function of temperature of cooling water and solved by using the sequential quadratic programming method [4]. Lerou *et al.*, carried out optimization of counterflow heat exchanger geometry through minimization of entropy generation [5].

There are many investigations carried out to improve the optimum design of shell and tube heat exchangers [6]. But few papers are available on optimization of the inlet condition of cross flow condenser with minimization of the exergy destruction. For this reason, the present study focuses on it as objective function which is subjected to condensation of entire mass flow rate of steam.

II. EXERGETIC OPTIMIZATION

Genetic algorithms method for optimization of shell and tube heat exchanger, the Bell-Delaware method for the description of the shell side flow and optimize the major geometrical parameters such as number of tube-passes, standard internal and external tube diameters, tube layout and pitch, type of head, fluids allocation, number of sealing strips, inlet outlet baffle spacing, and shell side and tube-side pressure drops [6].

A. Optimization Procedure

The aim is to establish the optimal inlet cooling water temperature through minimization of exergy destruction for a known condensation temperature and a given heat transfer area. Thus, the exergy destruction function becomes the objective function in the optimization problem, subject to the energy balance between cold and hot fluid streams as well as the governing het transfer equation representing the heat released due to the condensation of vapor, removed to the coolant by convective heat transfer. Symbolically, it may be expressed as,

$$Q = m_s h_{fg} = m_c c_{pc} (T_{c2} - T_{c1}) \quad \text{... (1)}$$

where, $T_{c1} \geq 10°\text{C}$ U_0 and A_{eff} denote the overall heat transfer coefficient and the effective heat transfer area respectively. Also, ΔT_{1m} represents logarithmic mean temperature.

Hence, the final form of the optimization problem can be written as,

Minimize $E_d = f(T_{c1})$

Subject to $m_s h_{fg} = U_0 A_{\text{eff}} \Delta T_{1m}$ ... (2)

It is necessary to calculate exergy destruction (E_d), exergetic (η_{ex}) efficiency, overall heat transfer coefficient (U_0), and effective heat transfer area (A_{eff}).

$$E_d = m_s \left\{ \begin{array}{l} c_{pv}\left[(T_{v1} - T_{cond}) - T_0 \ln\left(\frac{T_{v1}}{T_{cond}}\right)\right] \\ + h_{fg|T_{cond}} - T_0 s_{fg|T=T_{cond}} \end{array} \right\} - m_c c_{pc}\left[(T_{c2} - T_{c1}) - T_0 \ln\left(\frac{T_{c2}}{T_{c1}}\right)\right] \quad \text{... (3)}$$

$$\eta_{ex} = \frac{m_c c_{pc}\left[(T_{c2} - T_{c1}) - T_0 \ln\left(\frac{T_{c2}}{T_{c1}}\right)\right]}{m_s \left\{ \begin{array}{l} c_{pv}\left[(T_{v1} - T_{cond}) - T_0 \ln\left(\frac{T_{v1}}{T_{cond}}\right)\right] \\ + h_{fg|T_{cond}} - T_0 s_{fg|T=T_{cond}} \end{array} \right\}} \quad \text{... (4)}$$

$$\frac{1}{U_0} = \frac{d_0}{d_i}\frac{1}{h_t} + \frac{d_0}{d_i} R_{ft} + \frac{d_0 \ln\left(\frac{d_0}{d_i}\right)}{2k_b} + R_{f0} + \frac{1}{h_g} \quad \text{... (5)}$$

III. GEOMETRICAL CHARACTERISTICS

Objective of this section is to outline surface geometrical characteristics that are used in the determination of actual heat transfer coefficient and actual pressure drop. Andre and Eduardo [7] carried out the design optimization of shell and tube heat exchangers for the minimization of capital cost through minimization of the thermal surface area for a certain service, involving discrete decision variables. The Bell-Delaware method is used for shell side film coefficient and pressure drop evaluation. Important geometrical characteristics are the heat transfer area, minimum free-flow area, frontal area, hydraulic diameter, and flow length on each fluid side of the exchanger. The ratio of free-flow area to frontal area is needed for the determination of entrance and exit pressure losses. Model for various streams of flow was originally proposed by Tinker [8]. Plate baffle is used to improved heat transfer performance [9-10]. But in conventional shell-and-tube heat exchanger various clearances (tube-to-baffle hole clearance, bundle-to-shell clearance, and baffle-to-shell clearance) are required and also consider in present optimization for the construction of the exchanger.

Total shell side pressure drop is,

$$\Delta p_s = \Delta p_{cr} + \Delta p_w + \Delta p_{i-0}$$

$$= |(N_b - 1)\Delta p_{b,id}\varsigma_b + N_b \Delta p_{w,id}| \; \varsigma_1 + 2\Delta p_{wb,id}\left(1 + \frac{N_{r,cw}}{N_{r,cc}}\right)\varsigma_b \varsigma_s \quad \text{... (6)}$$

Total tube side pressure drop is,

$$\Delta p_{total,\,t} = \frac{G_t^2}{2g_c p_t}\left[\frac{1.5}{N_p} + \frac{f.L1}{d_i \phi_t^r} + K_c + K_e + 4\right] N_p \quad \text{... (7)}$$

All important geometrical characteristics associated with flow bypass and leakages for shell-and-tube condenser are presented. The geometrical information needed for rating such an exchanger. Bejan [11] demonstrated the use of irreversibility as a criterion for evaluation of the efficiency of heat exchanger to minimize the wasted energy.

Table 1: Geometrical Dimensions of Condenser.

Shell side inside diameter	D_s	5.1 m
Tube-Side Outside dia.	d_0	25.4 mm
Tube-Side Inside Diameter	d_i	22.9 mm
Tube Length	L	10 m
Total Number of Tubes	N_t	17,350
Number of Tube Passes	N_p	2
Tube Material		Cu/Ni 90/10
Thermal Conductivity of wall	K_b	45W/m K

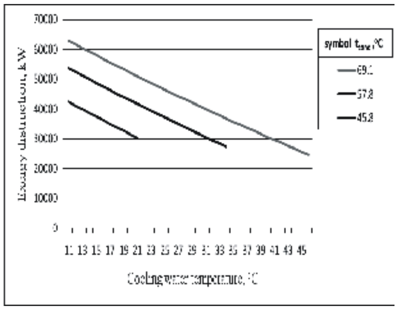

Fig. 1. Temperature at Different condenser pressures.

Fig. 2. Variation of Exergy Destruction with Cooing Water Temperature at Different condenser pressures.

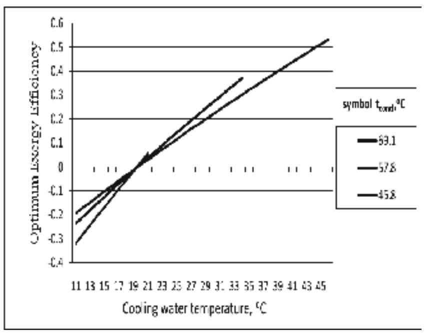

Fig. 2. Variation of Exergy Destruction with Cooling Water.

Table 2 : Operating parameters of Condens.

Cooling water flow rate	m_c	27000 m^3/hr
Cooling water inlet temperature	t_{c1}	30°C

Table 3 : Variations of Cooling Water Temperature, Exergy Destruction and Exergy Efficiency with Steam Mass Flow Rate at Three Different Environmental Temperatures p_{cond} = 0.18 bar (t_{cond} = 57.8).

m_s(%)	t_c(°C)	η_{ex}			E_d(kW)		
		t_0=25°C	t_0=30°C	t_0=35°C	t_0=25°C	t_0=30°C	t_0=35°C
70	41	0.4997	0.4305	0.3365	1.52E+04	1.51E+04	1.49E+04
80	39	0.4653	0.3901	0.2882	1.86E+04	1.84E+04	1.82E+04
90	36	0.4063	0.3212	0.2056	2.32E+04	2.31E+04	2.29E+04
100	34	0.371	0.2798	0.156	2.74E+04	2.72E+04	2.70E+04
110	32	0.3353	0.238	0.106	3.18E+04	3.17E+04	3.15E+04
120	29	0.2743	0.1666	0.0204	3.79E+04	3.78E+04	3.77E+04
130	27	0.2377	0.1237	–	4.31E+04	4.30E+04	4.29E+04
140	24	0.1751	0.0504	–	5.02E+04	5.02E+04	5.02E+04

p_{cond} = 0.18 bar (t_{cond} = 57.8°C)

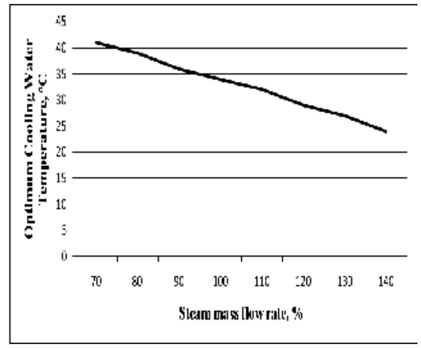

Fig. 3. Variation of Optimum Cooling Water Temperature with Steam Mass Flow Rate.

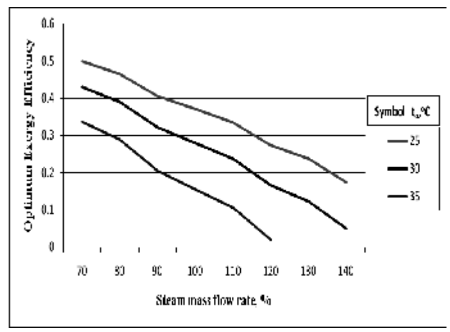

Fig. 4. Influence of Environment Temperature on Optimum Exergy.

p_{cond} = 0.30 bar (t_{cond} = 69.1°C)

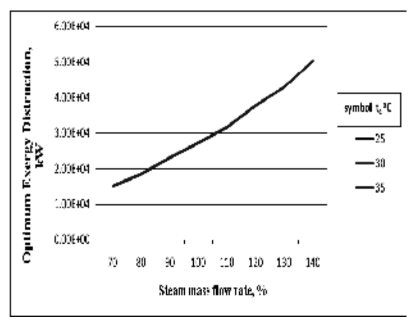

Fig. 5. Variation of Minimum Exergy Destruction with Steam Mass Flow Rates at Three Ambient Temperatures.

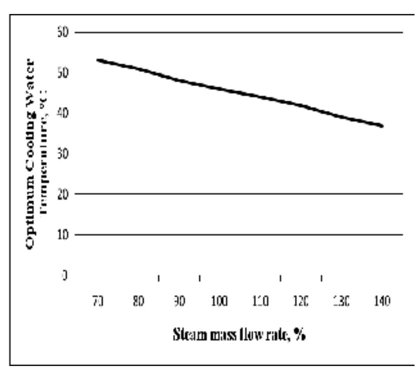

Fig. 6. Variation of Optimum Cooling Water Temperature with Steam Mass Flow Rate.

IV. CONCLUSION

The exergy destruction and exergy efficiency depend upon several parameters, such as the inlet temperature of cooling water, condensation pressure (temperature) and atmospheric temperature (dead state). From the present study following points are concluded.

1. With decrease of condenser pressure from 0.18 bar to 0.10 bar optimum cooling water temperature is also decrease from 34 to 21 , and exergy destruction at 21 decreases from 41755 kW to 30263 kW. So it is better to operate the condenser at as low as possible pressure, as expected.
2. For given condenser pressure, higher exergy efficiency can be obtained by increasing the cooling water temperature, but it may results in incomplete condensation of steam. At higher temperature of cooling water, some amount of steam may remains uncondensed. So, temperature of coolant is increased up to limits, such that all the steam is condensed. For the given operating condition operating temperature of cooling water cannot be increased more than 34. At this temperature, exergy destruction is 27350kW and exergy efficiency is 37.1%.
3. As the upstream mass flow rate increases by 20% of initial mass flow rate, the optimal inlet cooling water temperature required decreases from 34 to 29. .
4. Higher exergy efficiency is obtained with increase of cooling water flow rate, but it results in higher pressure drop. As pressure drop should not be more than 50kPa, so mass flow rate of cooling water can be increased up to 10% of the initial flow, at which exergy destruction is 26.4 MW.

Similarly exergoeconimic analysis can be carried out to optimize overall cost as well as exergy destruction of the condenser, in which initial capital cost, operating cost, and exergy destruction during operation of the condenser is minimized using the exergy method.

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