HEFAT2012
9th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
16 – 18 July 2012

Malta

OPTIMIZATION OF A CONDENSER IN A THERMAL POWER PLANT

Gokhan OZDAMAR[1] and Recep OZTURK[2]

[1]University of Firat, Department of Mechanical Engineering, ELAZIG

[2] University of Yildiz Technical , Department of Mechanical Engineering, ISTANBUL

E-mail: ozdamargokhan@hotmail.com

ABSTRACT

In this study, thermal power plant's, based on Ideal Rankine Cycle, steam is condensed while the pressure of the condenser changes between 0.1 bar and 0.02 bar. The effect of this condensation on condenser area and effects to the cost is investigated. Change of rate of condenser area is calculated, relative to the condenser pressure drops. It is concluded that the area of condenser increases when the pressure of condenser decreases, accordingly the cooling water pump power changes. This result is discussed in the impact of energy production costs.

At the optimisation of the condenser, heat transfer is very important because it a kind of heat exchanger. To find the heat transfer area, based on Rankine Cycle, NTU method is mainly used.

Environmental impacts and cost calculations of these systems are also briefly mentioned.

The results obtained within this study shows that if the condenser pressure decreases, then the condenser area and accordingly the cost and power increases. Optimum value of condenser pressure is 0.4 bar subject to some boundry conditions.

NOMENCLATURE

A_{NTU} [m^2]	The area which is needed for cooling water
At_{otal} [m^2]	Total condenser surface area
$c_{p\ water}$ [kj/kg^{0}C]	Specific heat of water
C_{min}	Heat capacity flow rate ($m_{water}.c_{pwater}$)
d_{out} [m]	Condenser outer diameter
ε [-]	Efficiency
h_n $\left[kj/kg\right]$	Enthalpy of the point n
h_{steam} $\left[kj/kg\right]$	The heat coefficient of steam
h_{water} $\left[kj/kg\right]$	The heat coefficient of cooling water
$k_{tube(pipe)}$ $\left[kj/kg\right]$	The heat coefficient of condenser pipe.
$K_{theoric}$	Theorical heat transfer coefficent of all heat transfer area
$K_{counted}$	Counted heat transfer coefficent of all heat transfer area
L_condenser[m]	Condenser pipe length
m_{steam} [kg/s]	Flow rate of steam
m_{water} [kg/s]	Flow rate of cooling water
n_{uni} [-]	Number of pipe in condenser
Ne[kW]	Power produced in the turbine
NP[kW]	Power of the condenser cooling wat pump
NU	Nuselt Number
NTU[-]	Number of transfer unit
P_3[Pa]	Inlet steam pressure of turbine
P_o [bar]	Pressure of condenser
P_{net}	Net Power
P_{pomp}	Power of Pomp
s_3	Inlet entropy
s_4	Outlet entropy
SS	Specific Heat
Re	Reynolds Number
q [kW]	Real heat transfer
q_{max} [kW]	Value of maxsimum heat transfer
Q_b[kW]	Transferred heat from steam to cooling water
Q_s[kW]	Transferred heat from condenser to cooling water
R [-]	Theoric heat transfer coefficient and the counted heat transfer coefficients ratio
v_{water}[m/s]	The velocity of the cooling water
T_{hi} ,$T_{ho,}$[^{0}C]	Inlet and outlet temperatures of steam
T_{ci} ,T_{co} [^{0}C]	Inlet and outlet temperatures of cooling water
T_3 [^{0}C]	The inlet temprature of turbine
t_g [^{0}C]	The temperature of the inlet cooling water
t_c [^{0}C]	The temperature of the outlet cooling water

η_i, η_m, η_p [-]	Efficiency of effectiveness, mechanic and cooling water pump	ρ [kg/m^3]	Density of the cooling water at the temperature of tort
δ [m]	Thickness of the condenser pipes	X	The degree of dryness

INTRODUCTION

For the calculation of an effective heat transfer coefficient, Sadık KAKAC, Bergles and Mayinger, in ‘heat exchangers’[1], define ε *_efficacy*, which is the ratio of the actual heat transfer to the maximum heat transfer at the heat exchanger. Kundakcioglu[4], studied on the subject, by using LMTD method to optimise the condenser. All preliminiers and notations are as in [4], [10] [8], [7], [3]. All calculations made by MAPLE 13. In this study, based on Rankine Cycle, datas are taken from NTU method and then the effected heat transfer coefficent , heat transfer area are calculated.

ANALYSIS METHOD

Thermodynamical Analysis

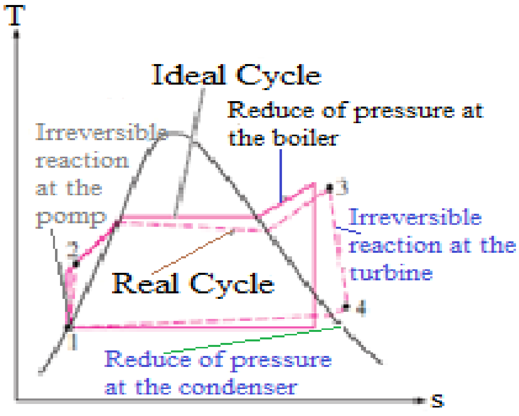

Figure 1 Ideal and Real Rankine Cycle States,[7]

At the 1-2 state; in the boiler pomp, fluid pressured to the boiling level of boiler,
2-3, steam boiler, the glow of the evaporation occurs at constant pressure,
3-4, steam entering to the turbine, expands isentropic,
At the 4-1, at the constant pressure, heat is thrown out, [7].

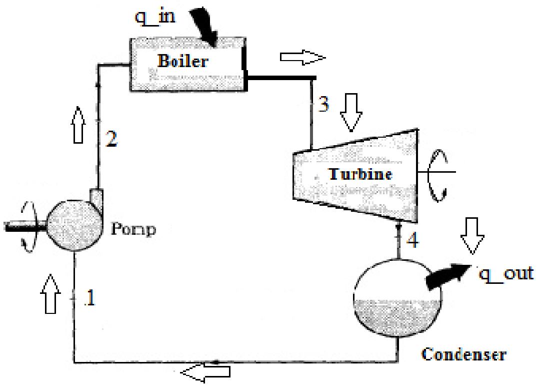

Figure 2 Scheme of power plant steam turbine [5]

In a plant, which has boiler outlet temperature and pressure T_3 and P_3, respectively, also condenser pressure is shown by P_o. According to T_3 and P_3, the inlet enthalpy of the steam which is outed from boiler through the turbine is h_3 (kJ/kg) can be found in the thermodynamic tables, [7] .According to ideal Rankine cycle by the acceptance of isentropic change of state between the points 3 and 4, we can say ,entropy at points 3 and 4 equals to each other ($s_3 = s_4$). At the P_o if we denote saturated water’s (x = 0) enthalpy by h_1 and the turbine efficiency by η_i; then η_i and Δh_t can be formulated as in [4], [10].

$$\eta_i = \frac{\eta_e}{\eta_m} \tag{1}$$

$$\Delta h_t = h_3 - h_4 \tag{2}$$

Thermodynamic law for water feeded pump of boiler, in the ideal Rankine Cycle can be given as in [4], [10].

$$h_1 - h_{2g} = v(P_o - P_3) \tag{3}$$

Where h_1 denotes the enthalpy at the point 1 and the values of it can be taken from Moliere Diagram.
At the real cycle, h_1 can be calculated by using the following equation; [4]

$$h_1 - h_2 = \eta_p(h_1 - h_{2g}) \tag{4}$$

or

$$h_{2g} = h_1 - \frac{(h_1 - h_2)}{\eta_p} \tag{5}$$

In real Rankine cycle, entalpy at the point 4g can be found from;

$$h_{4g} = h_3 - (h_3 - h_4)\eta_i \tag{6}$$

The degree of dryness, shows important reduces at the 100 bar and 400^{0}C. Which is harmfull for the turbine, it causes corrosion and for preventing this effect; the solition is reheating. Symbols h_{3_2} and h_{4_2} represent the inlet entalphy and outlet entalphy of turbine respectively.
Efective specific steam consumption, SS; can be evaluated from the following equation,

$$SS = \frac{3600}{\eta_e \, \Delta h_t}\left[kg/kWh\right] \tag{7}$$

[10], [4]. The flow rate of the steam which is outed from the turbine is;

$$\dot{m}_{steam} = \frac{SS\ \ N}{3600}\left[kg/s\right] \tag{8}$$

At this study, the theorical power of turbine (N) is in kW. It changes from 5 MW to 100MW.
The generated power at the turbine can be calculated from the following

$$Ne = \dot{m}_{steam}\ \ \Delta ht\ \ \eta_e\ \left[kW\right] \tag{9}$$

CALCULATION OF CONDENSER HEAT TRANSFER AREA; [4], [10]

The value of transferred heat, from condensing steam to cooling water is represented by Q_b [kW]. And it can be formulated as;

$$Q_b = \dot{m}_{steam}(h_1 - h_{4g}) \tag{10}$$

Also the transferred heat from condenser to cooling water is important and the next equation shows the value of it, [6];

$$Q_s = \dot{m}_{water} \;\; c_{p\_water} \, \Delta T \tag{11}$$

ΔT ,is the difference between the entry and output temperatures of cooling water when it enters and goes out from the condenser. That can be writen more openly

$$Q_s = \dot{m}_{water} \;\; c_{p\_water}(t_c - t_g) \tag{12}$$

At the steam turbine plants, ΔT generaly varies between 6^0C and 10^0C. In this study it is 8^0C. The theorical heat transfer coefficient, $K_{teorik,}$ has been accepted between 2.000 and 4000W/m^{2}K [10], [4].

Calculating the heat transfer area by NTU method; [8]

Logarithmic temperature difference method can be used when the entry and the output tempratures of the fluid are known or they can be calculated by the help of energy conservation equation. In this case LMTD (logarithmic temperature difference method) is very easily applicable. When there is no data about the output temperature of the fluid NTU (the number of transfer unit) method is much more usefull, [3].

$$q_{max} = C_{min}(T_{hi} - T_{ci}) \tag{13}$$

After that information, the ratio of real heat transfer and the possible maximum heat tranfer is called ε - efficacy;

$$\varepsilon = \frac{q}{q_{\max}} \tag{14}$$

[8] ,or equivalently

$$\varepsilon = \frac{C_h(T_{hi} - T_{ho})}{C_{\min(}T_{hi} - T_{ci})} \tag{15}$$

Heat transfer can be determined by using efficacy

$$q = \varepsilon . C_{\min} . (T_{hi} - T_{ci}) \tag{16}$$

NTU, the common usefull method of analysing the heat exchangers, is a dimensionless parameter

$$NTU \equiv \frac{UA}{C_{\min}} \tag{17}$$

From that formula, area can be found easily, and it is called A_{NTU}. Total condenser cooling area is the sum of A_{NTU} and the 10 % of $A_{NTU,}$ [2], [3] [9] ;

$$A_{total} = A_{NTU} + (0{,}1)A_{NTU} \tag{18}$$

After finding the A_{total}, the condenser tube (pipe) length can be selected from the standard condenser tube length table.

One of the important data to be found is m_{water} because when m_{water} is known, to find the v_{water} is possible from the equation

$$V_{water} = \frac{4.m_s}{\pi . d_i^2 . n . \rho}(m/s) \tag{19}$$

V_{water} is important to calculate the value of Reynolds number (Re), which gives an idea about the flow character, turbulance or laminar. The flow is turbulanced in this study and fluid gets heat. All of these datas are for finding heat transfer coefficient for steam. The usage of Nu and Re number is the main way to determine this coefficient. Now, there is 3 layers in condenser parallel to each other , cooling water flow, tube , steam. The general heat transfer coefficient $K_{counted}$ can be found from the following

$$\frac{1}{K_{counted}} = \frac{1}{h_{\mathrm{water}} *} + \frac{\delta}{k_{tube}} + \frac{1}{h_{steam}} \tag{20}$$

k_{tube} is the heat transfer coefficient of conduction for tube, h_{water} is the convection coefficient like h_{steam} [4]. If all parameters are known then we can check the cycle of R by the formula;

$$R = \frac{K_{theoric} - K_{counted}}{K_{theoric}} \tag{21}$$

R, is a kind of ratio and the acceptable value for R is 0.01.

Net power;

$$P_{net} = Ne - P_{pump} \tag{22}$$

gives an idea about the efficiency of power plant [5]. When condenser pressure drops, turbine power and the used power by pump increases [10].

ECONOMICAL ANALYSIS

To find an approximate value for the cost of condenser, the tube (pipe) length and the material of this tubes necessary, [10].

$$\left.\begin{array}{l} L\_total\_tube\_lenght = L.\,(n\_unit) \\ Con_{\cos t} = \left(L\_total\_tube\_lenght\right).(unitprice) \end{array}\right\}$$

(23)

ASSUMPTIONS

Mechanical efficiency, efficacy efficiency, cooling water pump efficiency are $\eta_m = 0.99$, $\eta_e = 0.80$ $\eta_p = 0.80$ respectively.

To find the condenser heat transfer area the $K_{theoric}$ is selected between 2500 W/m^{2}K – 4000 W/m^{2}K. $h_{steam} = 12000$ kcal/m^{2}hK

The difference between the cooling water inlet temperature and the outlet temperature is $\Delta T = 8$ °C. the density of water for this temperature is $\rho = 1.005 \times 10^3$ kg/m^3. For the BWG 18 (Birmingham Wire Gauge) standards, thickness of the tubes are 1,24 mm. the used material is Admiralty. For the economical analysis the unit price is taken as 15 TL/m, [10].

CONCLUTIONS

This study is calculated in MAPLE 13, the graphs drawn by that program . It is clear that when pressure drops, the cooling surface area of the condenser increases, tube length increases, and cooling water pump power changes.

Cost , one of the important criteria , increases when condenser pressure decreases under the 0.4 bar. [10]

Table.1 Condenser Pressure and Condenser Surface Areas Data at 60 bar pressure 400 ^{0}C temperature with a flow rate 40 kg /s ;

Condenser pressure (bar)	0.1	0.09	0.08	0.07	0.06	0.05	0.04	0.03	0.02
Conderser Area(m^2)	448.96	503.53	531.97	630.08	711.49	830.55	1166.01	2756.70	5562.48

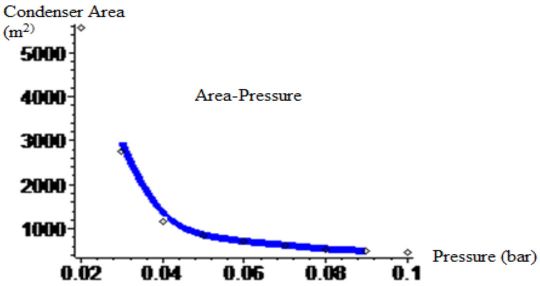

Figure 3 The change of condenser area and condenser pressure[10]

Accompanying figure 3. suggest that when the condenser pressure decreases, the area (Area= π .L.n.(d$_{out}$) [4] increases. Because when the pressure at the condenser drops the length of the pipes icreases. Pipes are the main heat transfer areas for the condenser.

Table.2 Condenser Pressure and Turbine Power Data at 60 bar pressure 400 ^{0}C temperature with a flow rate 40 kg /s

Pressure of condenser (bar)	0.1	0.09	0.08	0.07	0.06	0.05	0.04	0.03	0.02
Turbine Power(kW)	34505.60	34985.60	35785.60	36265.60	36745.60	37385.60	38345.60	38985.60	39945.60

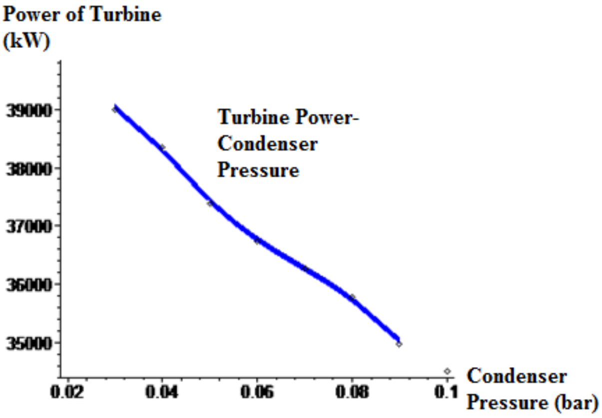

Figure 4 Condenser pressure and turbine power changes[10]

The main theme for the turbine is getting the steam dry as posibble it is important because the m$_b$ is important for the condenser

.

Table.3 Condenser Pressure and Counted Heat Transfer Coefficient Data at 60 bar pressure 400 ^{0}C temperature with a flow rate 40 kg /s

Pressure (bar)	0.1	0.09	0.08	0.07	0.06	0.05	0.04	0.03	0.02
$K_{Counted}$ (W/m^{2}K)	2772.08	2671.65	2805.08	2649.67	2694.61	2808.85	2618.24	2220.67	2101.99

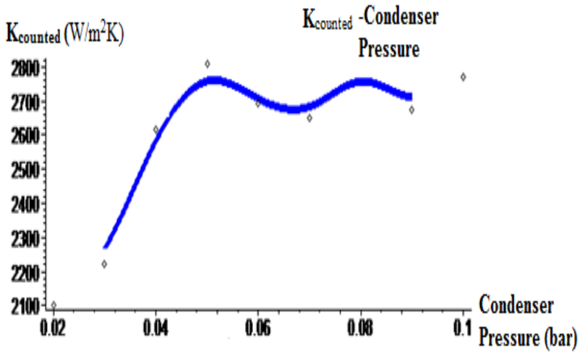

Figure 5 The counted heat transfer coefficient and the condenser pressure changes [10]

As shown at the figure 5 when condenser pressure increases ,the heat coefficient is also decreases. [10]This coefficient, effects the character of the flow which depends on Re and Nu

.

Table.4 Condenser Pressure and Condenser Cost Data at 60 bar pressure 400 ^{0}C temperature with a flow rate 40 kg /s

Condenser pressure (bar)	0.1	0.09	0.08	0.07	0.06	0.05	0.04	0.03	0.02
con_cost (TL)	223387.25	250543.05	224254.04	265611.12	257083.13	233413.79	290084.99	457216.37	864912.43

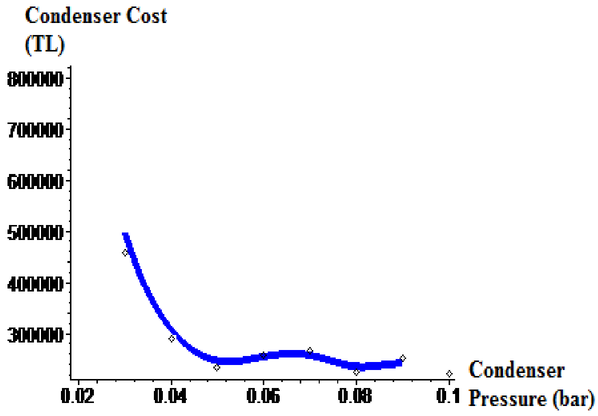

Figure 6 Condenser cost and Condenser Pressure Changes [10]

At the equation(23) there is the relation between pipe(tube) length and the cost.The optimum point is important, that point is ;between 0,04-0,05 bar. After that slope of the figure goes up that means the cost is increasing. [10]

Table 5 Condenser Pressure and Condenser Cost Data at 60 bar pressure 400 ^{0}C temperature with a flow rate 40 kg /s [10]

K_{cost}	2772.08	2671.65	2805.08	2649.67	2694.61	2808.85	2618.24	2220.67	2101.99
h_{water}	6457.26	6134.94	6565.19	6065.60	6207.80	6577.60	5967.22	4793.65	4467.00

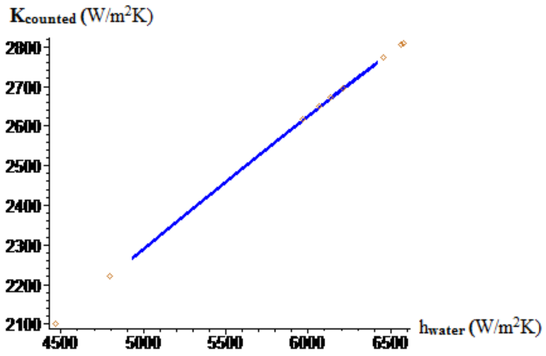

Figure 7 The counted general heat transfer coefficient and the cooling water convection coefficienti [10]

h_{water} is the heat transfer coefficient of cooling water which is related with Nu number can be find from (20). Figure 7 shows the effect of hwater at the total counted heat transfer coefficient. As a result when the cooling water heat transfer coefficient increases relatively $K_{counted}$ increases too. [10]

REFERENCES

[1] Kakac, S., A.E. Bergles, F. Mayinger, Eds.,‘ Heat Exchangers’, Hemissshere Publishing, New York, 1981.

[2] Ozturk, R., ‘MARINE TECHNOLOGY’, Vol.41,No.1,pp17-21, 2004.

[3] Genceli F.O., ‘Isı Degistiricileri’, Birsen Publishing, ISTANBUL, 2005.

[4] Kudakçıoğlu, E.D.,Optimisation of a Condenser in a Thermic Power Plant, MsC thesis, Yıldız Technical University, ISTANBUL, 2001.

[5]http://www.scribd.com/doc/54185767/Rankine-cevrimi.

[6] Ozturk,R. Energy plant and businesses , Lecture Notes ,YTU, ISTANBUL, 1998.

[7]Cengel,Y., ‘Thermodynamics An Engineering Approach’, ISBN:978-975-6240-26-7,5th Edition, Izmir Guven Publishing, 2008.

[8]Incropera F.P., Dewitt D.P., ’ Fundamentals of Heat and Mass Transfer , Literatur Publishing, Istanbul, 2000.

[9]Peselik, I. , Hydrolic Machines, Lecture Notes, YTU, 1997.

[10] Ozdamar, G., ‘Optimisation of a Condenser In the Power Plants and Flow Analisys’, Yıldız Technical University, ISTANBUL, 2011.