Simulation of Perturbation Effects on the Air Cooled Steam Condenser Backpressure in a Combined Cycle Power Plant

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Abstract*– The instructions give the basic guidelines for the dynamic behavior of an air-cooled steam condenser (ACSC) backpressure is analyzed in this paper; the inquiry is performed when the equipment suffers changes on the operational conditions. The study is conducted using a simulation technique which consists on first dividing the equipment in modules and then applying the mass and energy conservation equations, to the corresponding control volumes using the concentrated parameters concept, secondly, the set of differential equations for each module are organized in a matrix form and is solved using the commercial software MATLAB-Simulink. Finally, an integral program containing all the individual modules is solved and its solution provides the pressure at the turbine outlet also known as backpressure. Some parameters included in the equations were estimated using empirical expressions, while others were obtained adjusting the employed model using operational data of the Combined Cycle Gas Turbine Plant (CCGT).

The disturbances produced in the simulation took place when the air cooling flow rate and the air temperature in certain section or sections of the ACSC were abruptly changed. Other disturbances were provoked by changing the vapor mass flow rate coming into the ACSC. All effects due to the disturbances are reflected in the turbine backpressure. The response of the model was compared against the power plant steady state operational data for its validation. As supposed, the model presents a better approach for the operational conditions near to the adjustment point, corresponding to the design conditions. In general, the qualitative response of the model for different operational points is logic and acceptable.

***Keywords*-- mathematical modelling, dynamic simulation, air cooled steam condenser, Simulink, combined cycle.**

I. INTRODUCTION

The condenser is an important component in a thermal power plant. Its fundamental role is condensing steam, besides this equipment produces vacuum, which is an important feature for the steam cycle. The lower the condensation pressure, the better the condenser performance, and then, the power plant efficiency is augmented. Since enthalpy of the steam is lower for a small backpressure (pressure at the turbine outlet), there is a bigger difference in the enthalpy between the inlet and exit of the turbine, yielding an important increment of the power delivered. The vacuum produced by the condensation effect is proportional to the rate of heat extracted from the system. It depends on various parameters such as the type of fluid used, the geometry and number of finned tubes that configure the heat exchanger.

Due to its good thermal properties and availability in many places, the cooling fluid typically used in power plants is water. However, there are several arid regions where other types of steam condensation system are required, such a system is known as air-cooled steam condenser (ACSC) that avoids water dependence. These types of condensers are important not only for dry areas but also they are installed even in places where the water resource is not a problem. This is because power plants based on ACSC do not contend against other human activities using water, so the permissions to construct facilities are readily given. Also these systems do not produce environmental damage to rivers and oceans as open and close conventional systems do.

ACSC thermal performance (associated to heat transfer rate) is highly dependent on environmental conditions such as air velocity and temperature [1] and its efficiency is lower than the conventional systems [2]. Also ACSC requires much heat transfer area to work, so that, they are big and expensive. There are also several problems involved on the operation of these systems: recirculation of warm air affects negatively the condenser performance and the wind speed and direction have a great influence on it [3], thermal resistance for fouling appears during operation and it has to be removed from the system [4], air mass flow rate sent by the fans should be regulated to an optimal pressure value [5]. Most of research carried out on ACSC has been directed to solve a specific problem by static models.

A way to analyze general behavior and performance of a system is employing dynamic analysis. Since dynamic response is fundamental for designing control systems, control engineers use the information obtained from this approach to design secure installations and protect the equipment. It also allows having a complete description of any system because models contain multiple inputs and the effects of each one can be rapidly studied through the output variables, so a lot of operation conditions can be run and see the effect on some important output parameters. The dynamic behavior of an ACSC was studied for different disturbances like abrupt reduction on air mass flow rate, air temperature and steam mass flow rate [6]. This approach could be made on design phase or be applied to an installation working analysis. In this work this analysis will be implemented on the ACSC operating in a CCGT located at Puebla, Mexico.

NOMENCLATURE

A_{isct}	[m^2]	Tube interior area per tube
A_{et}	[m^2]	Total exterior area per tube
A_{iscM}	[m^2]	Interior area of all the tubes per submodule
A_{eM}	[m^2]	Total exterior area per submodule
C_{pa}	[J/kg·K]	Air specific heat
C_{pt}	[J/kg·K]	Tube material specific heat
h	[W/m^2·°C]	Heat transfer coefficient
i	[J/kg]	Specific enthalpy
i_{fg}	[J/kg]	Decrease on specific enthalpy due to condensation
m	[kg]	Mass
$\dot{m}$	[kg/s]	Mass flow rate
n_t	[-]	Number of tubes per module
p	[kPa]	Pressure
R	J/kg·K	Steam constant
T	[°C]	Temperature
V	[m^3]	Volume
Special characters		
η_{se}	[-]	Exterior total area effectiveness
ρ	[kg/m^3]	Density
Subscripts		
aet		Air at tube inlet
ast		Air at tube outlet
aVC		Air in the control volume (between fins)
aVCt		Air in the control volume (between fins) per tube
aVCM		Air in the control volume (between fins) per submodule
c		Condensation
e		Exterior
isc		Interior condensation section
iscM		Interior condensation section per submodule
lsst		Saturated liquid at tube outlet per tube
M		Submodule
t		Tube
tM		All tubes in the submodule
vset		Saturated steam at tube inlet per tube
vseM		Saturated steam at tube inlet per submodule
vsst		Saturated steam at tube outlet per tube
vsVC		Saturated steam in the control volume

II. SIMULATION TECHNIQUE

It is convenient to apply series of consecutive steps shown in Fig.1. The starting point is to analyze pipe and instrumentation diagrams (P&ID's) and its simplification. The former step will depend on the objectives and the deepness of the study. After doing P&ID's simplification, each equipment is considered as a module, according to its function in the system. If one of them requires a detailed analysis or contains internal devices, the module must be divided into submodules inside. This separation permits to see the whole system under analysis as several parts ordered in a hierarchical form. The ACSC under study is configured by 15 fans located in three different rows or sections, so on each one there are 5 fans placed. The module of the ACSC is divided and 15 submodules are determined to do the analysis. Each submodule contains *n* tubes cooled by the same fan. Once the modules or submodules are defined, conservation laws of mass and energy using the *lumped parameters* method are applied.

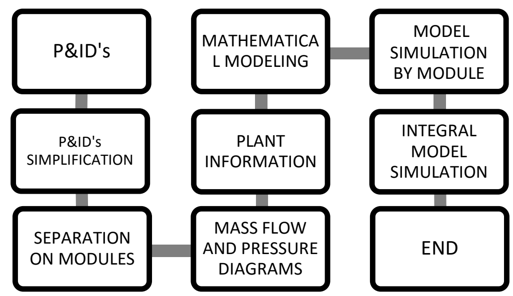

Figure 1. Steps executed to carry out the simulation.

Generally the set of differential equations obtained contains parameters which can be computed through auxiliary equations. The set of equations are solved by simulation software. The former offers an environment where the user can add a drag blocks to create a simulation program whose output is the behavior of the some important parameters throughout the time. Finally, simulation program of each module is coupled along with all the modules of the installation to create an integral simulation model than can be run under various operational conditions and disturbances.

III. MATHEMATICAL MODELING

A. *Individual tube*

Using an individual finned tube as reference shown in Fig. 2 2, mathematical model is obtained. The aim is to analyze any important parameter indicating the performance of the system, the most important parameter is condensation pressure.

The following assumptions were considered for the model:

- Uni-directional flow
- The working fluid is pure
- Tube wall thermal resistance is negligible
- Thermal resistances inside and outside of the tube because of fouling are neglected
- Drop pressure is negligible

- *Steam side*:

Mass balance

$$\frac{d}{dt}m_{vsVC} = \dot{m}_{vset} - \dot{m}_{lsst} - \dot{m}_{vsst} \qquad (1)$$

Saturated steam mass flow rate at the outlet tube is proportional to the quantity of steam mass in the tube,

$$\dot{m}_{vsst} \propto m_{vsVC} \qquad (2)$$

a constant of proportionality is required to turn the relation into an equality,

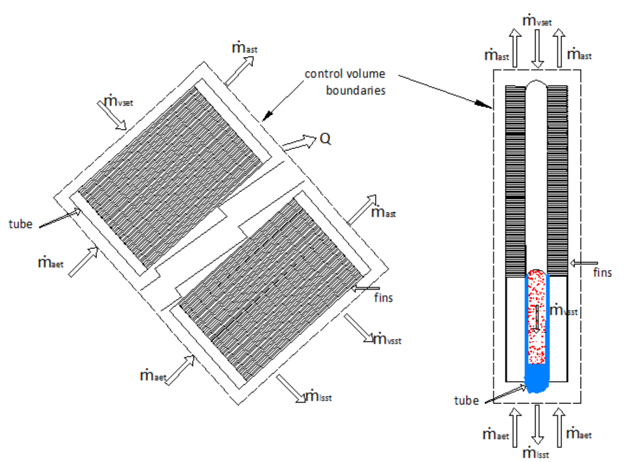

Figure 2. Control volume of an inclined finned tube taken as a reference for balances.

$$\dot{m}_{vsst} = a\, m_{vsVC} \qquad (3)$$

Units of a are s^{-1}

Energy balance

It is assumed that specific enthalpy of the saturated steam is constant along the tube.

$$\frac{d}{dt}m_{vsVC}i_{vsVC} = \dot{m}_{vset}i_{vsVC} - \dot{m}_{lsst}i_{lsVC} - \dot{m}_{vsst}i_{vsVC} - h_{isc}A_{isct}(T_c - T_t) \qquad (4)$$

Considering the change on specific enthalpy of saturated steam over time is negligible

$$\frac{d}{dt}m_{vsVC}i_{vsVC} = i_{vsVC}\frac{d}{dt}m_{vsVC} \qquad (5)$$

Conservation of energy in the tube wall

$$\frac{d}{dt}m_tC_{pt}T_t = h_{isc}A_{isct}(T_c - T_t) - h_e\eta_{se}A_{et}(T_t - T_{aVC}) \qquad (6)$$

- *Air side*:

Mass balance

$$\frac{d}{dt}m_{aVCt} = \dot{m}_{aet} - \dot{m}_{ast} \qquad (7)$$

If the mass of air in the duct between the fins per tube does not accumulate,

$$\dot{m}_{aet} = \dot{m}_{ast} = \dot{m}_{at} \qquad (8)$$

Energy balance

$$\frac{d}{dt}m_{aVCt}i_{aVC} = \dot{m}_{aet}i_{ae} - \dot{m}_{ast}i_{as} + h_e\eta_{se}A_{et}(T_t - T_{aVC}) \qquad (9)$$

$$\frac{d}{dt}m_{aVCt}C_{pa}T_{aVC} = \dot{m}_{at}C_{pa}(T_{ae} - T_{as}) + h_e\eta_{se}A_{et}(T_t - T_{aVC}) \qquad (10)$$

taking the specific heat coefficient constant and due to the mass of air remains also constant,

$$\frac{d}{dt}m_{aVCt}C_{pa}T_{aVC} = m_{aVCt}C_{pa}\frac{d}{dt}T_{aVC} \qquad (11)$$

It is assumed that the temperature of the air between fins is the average between the input and exit of the finned tube.

$$T_{aVC} = \frac{T_{ae} + T_{as}}{2} \qquad (12)$$

Algebraic process is done over the above set of equations to obtain the following system of differential equations per tube,

$$\frac{d}{dt}m_{vsVC} = -am_{vsVC} + \frac{h_{isc}A_{isct}}{i_{fg}}T_t + \dot{m}_{vset} - \frac{h_{isc}A_{isct}}{i_{fg}}T_c \qquad (13)$$

$$\frac{dT_t}{dt} = -\left(\frac{h_{isc}A_{isct}}{m_tC_{pt}} + \frac{h_e\eta_{se}A_{et}}{m_tC_{pt}}\right)T_t + \frac{h_e\eta_{se}A_{et}}{m_tC_{pt}}T_{aVC} + \frac{h_{isc}A_{isct}}{m_tC_{pt}}T_c \qquad (14)$$

$$\frac{d}{dt}T_{aVC} = \frac{h_e\eta_{se}A_{et}}{m_{aVCt}C_{pa}}T_t - \frac{(2\dot{m}_{at}C_{pa} + h_e\eta_{se}A_{et})}{m_{aVCt}C_{pa}}T_{aVC} + \frac{2\dot{m}_{at}}{m_{aVCt}}T_{ae} \qquad (15)$$

B. Submodule

A submodule of n tubes takes into account the number of finned tubes which are cooled by the same fan. To model a submodule, in addition to assumptions made for an individual tube, the flow of air sent by the fan and the steam flow will be taken uniform for each tube. The set of differential equations per submodule (which have been already ordered) is,

$$\frac{d}{dt}m_{vsVC} = -\frac{a}{n_t}m_{vsVC} + \frac{h_{isc}A_{isct}}{i_{fg}}T_t + \frac{1}{n_t}\dot{m}_{vseM} - \frac{h_{isc}A_{isct}}{i_{fg}}T_c \quad (16)$$

$$\frac{dT_t}{dt} = -\left(\frac{h_{isc}A_{iscM}}{m_{tM}C_{pt}} + \frac{h_e\eta_{se}A_{eM}}{m_{tM}C_{pt}}\right)T_t + \frac{h_e\eta_{se}A_{eM}}{m_{tM}C_{pt}}T_{aVC} + \frac{h_{isc}A_{iscM}}{m_{tM}C_{pt}}T_c \quad (17)$$

$$\frac{d}{dt}T_{aVC} = \frac{h_e\eta_{se}A_{eM}}{m_{aVCM}C_{pa}}T_t - \frac{(2\dot{m}_{aV}C_{pa} + h_e\eta_{se}A_{eM})}{m_{aVCM}C_{pa}}T_{aVC} + \frac{2\dot{m}_{aV}}{m_{aVCM}}T_{ae} \quad (18)$$

where *a* will be computed through steady estate values of ACSC.

In the model the steam is considered as an ideal gas,

$$p_cV = m_{vsVC}RT_c \quad (19)$$

Also, some expressions for the thermodynamic relation between saturation pressure and temperature were considered. These expressions will calculate the condensation temperature by using the pressure value got with the ideal gas model above.

C. Steam Duct at the turbine outlet

Mass balance

$$\frac{dM_v}{dt} = \dot{m}_{vst} - \dot{m}_{01} - \dot{m}_{02} - \dot{m}_{03} \quad (20)$$

Considering that the pressure drop inside the tubes is very small, mass flow rates can be expressed as a function of the pressure drop inside the tubes,

$$\dot{m}_{ij} = A(p_i - p_j) \quad (21)$$

Where A is the admittance according to [7]. Using again the equation (19) for the steam, the equation (20) becomes,

$$\frac{dp_0}{dt} = -\frac{(A_{01} + A_{01} + A_{01})}{\tau_0}p_0 + \frac{1}{\tau_0}\dot{m}_{vst} + \frac{A_{01}}{\tau_0}p_1 + \frac{A_{02}}{\tau_0}p_2 + \frac{A_{03}}{\tau_0}p_3 \quad (22)$$

In the last equation τ_0 is,

$$\tau_0 = \frac{V}{RT} \quad (23)$$

It will be considered, for simplicity, that pressure inside each of the submodules, which integrates one row is the same, so pressure can be different between rows (sections), depending on the operation conditions. Pressures p_1, p_2 and p_3 on equation (22) are the pressure at rows 1, 2 and 3 respectively while p_0 is the presssure inside the steam duct at the turbine exit.

IV. SIMULATION

A simulation program was implemented in block Simulink environment. Some parameters (coefficients in the equations) were adjusted by power plant data during one steady state using the design ambient air temperature and steam flow mass rate, while others were obtained by empirical relations available in heat exchange equipment literature.

Fifteen submodules (integrated in three rows) and the steam duct were coupled in a general dynamic simulation program which represents the ACSC under study. Finally, the general program was utilized to simulate the ACSC dynamic response and its performance under different operation conditions and disturbances.

- *Steady state ACSC performance under distinct operational conditions*

On these tests, we will analyze the value of the backpressure for different conditions at steady state.

Table 1 shows the results of four tests carried out during the simulation. Air mass flow rate and inlet air temperature vary between rows for each test.

Test one is done using design operation conditions, which are the same for each row. In test two the air mass flow rate was reduced to 2000 on row 3 and pressure is increased to 24.3 kPa. Test three was practiced to know the inlet air temperature effect, this situation can occur due to the recirculation of hot air, because of that the average temperature at each row is different. Finally, test four indicates the ACSC performance if the steam flow rate is reduced.

A lot of different operation conditions can be performed to see how they impact over the backpressure, which is a parameter that reflects the ACSC performance.

- *Validation*

The model was validated by comparing its output values versus empirical data obtained from the power plant. Three different steady states were used to determine the error of the model as presented in Table 2. Model results are closed to the power plant empirical data values when the input data are closed to the ones that were used for the model adjustment. However for points far from the conditions used for this purpose, the error increases, which means that we need to improve the model to take into account this fact.

Table 1. Tests for steady state under several operation conditions

OPERATION CONDITIONS					PROGRAM OUTPUTS	
Test number	Steam mass flow rate (kg/s)	ACC row	Air mass flow rate (kg/s)	Inlet air temperature (°C)	Backpressure (kPa)	Steam saturation temperature (°C)
1	126	1	3315	25	19.88	60.01
		2	3315	25		
		3	3315	25		
2	126	1	3315	25	24.31	64.43
		2	3315	25		
		3	2000	25		
3	126	1	3315	30	22.42	62.63
		2	3315	25		
		3	3315	28		
4	100	1	3315	25	14.19	52.91
		2	3315	25		
		3	3315	25		

Table 2. Values of model and power plant data for model validation

Parameter	Test 1		Test 2		Test 3	
	Power plant data	Model	Power plant data	Model	Power plant data	Model
Ambient air temperature (°C)	26.17	26.17	22.31	22.31	15.99	15.99
Steam mass flow rate(kg/s)	123.12	123.12	123.08	123.08	122	122
Condensation water pressure (kPa)	21	20.22	19	16.93	16	12.37
Error on backpressure		3.71 %		10.89 %		22.69 %

- *ACSC response to disturbances (transient response)*

All following tests begin at steady state with the next design operation conditions: air mass flow rate 3315 kg/s per row, inlet air temperature 25° C and steam mass flow rate 126 kg/s. When the disturbance occurs the ACSC goes to another state, the abrupt change produced is explained next.

Air mass flow rate disturbance.
The disturbance is produced when the air mass flow rate is suddenly reduced to half of its design maximum value at row one. Fig. 3 illustrates how the pressure changes until a new value is reached and kept.

Steam mass flow rate disturbance. An abrupt drop on the steam mass flow rate is created to analyze its influence on backpressure. The drop goes from 100 % to 85% of the design value. This could occur when the steam flow from the turbine is reduced using a valve to control the flow. Fig. 4 shows how the back pressure is reduced.

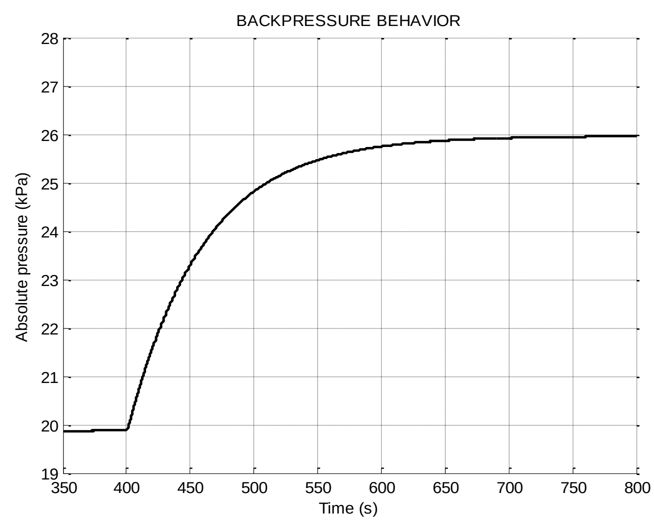

Figure 3. Backpressure behavior due to an abrupt reduction on air mass flow rate at row one.

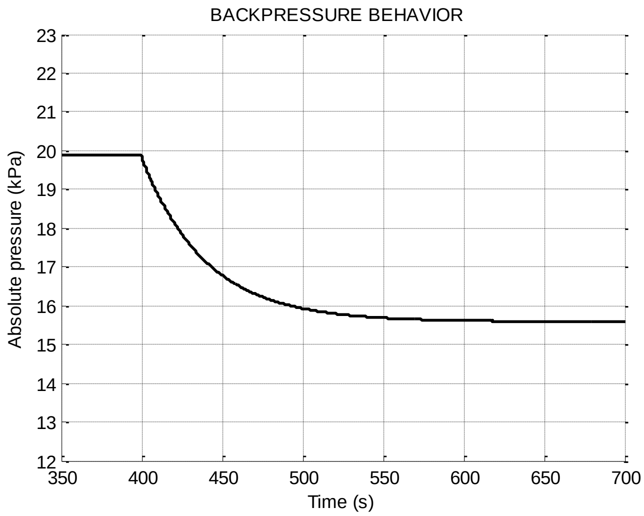

Figure 4. Back pressure behavior after a drop on steam mass flow rate occurs.

Inlet air temperature disturbance. This simulation allows analyze what would happen if some recirculated hot air enters in a row and fans send it back for cooling the tubes. Temperature at row one is increased from 25°C to 30°C. Fig. 5 shows the way the backpressure increases.

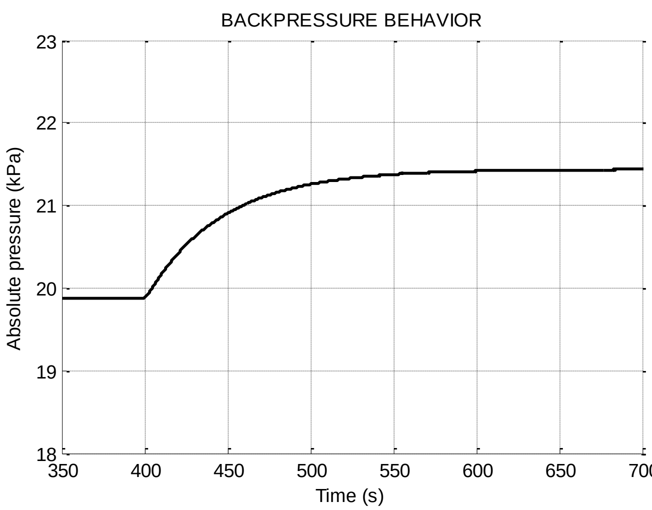

Figure 5. Back pressure behavior after increasing the inlet air temperature at one row.

CONCLUSIONS

The dynamic response of the model is good, when different operation conditions are introduced, the output values of several parameters meets what logically would occur in actual ACSC. The model is accurate for input values around design operation conditions, but when the inputs are far from that values, the precision decreases. This is because the model was adjusted using power plant data closed to design conditions. This dynamic model functions as a computational laboratory to analyze different conditions, it can predict which will be the maximum and minimum condensation pressure, depending on extreme ambient air temperature during a year, so this can help to determine whether the condenser will meet the steam turbine critical specifications. It also helps to identify which are the most important parameters on the performance of the condenser to focus on these ones during the design stage. Due to the modular structure, model can be improved by adding equipment and more complex and precise models.

Model can predict the transitory behavior of the backpressure if any reduction of cooling air take place. Another transient problem associated with the operation of ACSC, the recirculation of warm air, can be analyzed if there is a previous study to determine certain average inlet air temperature for each row. The model requires this temperature to determine its impact on backpressure. Further investigation should be made to create a model whose response meets with actual plants at any operation point and can be useful as a reference for control engineers.

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REFERENCES

[1] M. Pieve and G. Salvadori. “Performance of an Air Cooled Steam Condenser for a Waste-to-Energy Plant over its whole operating range”. *Science Direct, Energy Conversion and Management* 52, 2011; p.p. 1908-1913.

[2] D. Huifang and R. Boehm. “An Estimation of the Performance Limits and Improvement of Dry Cooling on Trouhg Solar Thermal Plants”. *Science Direct, Applied Energy* 88, 2011; p.p. 216-223

[3] Z. Wanli and L. Peiqing. “Effect of Wind Recirculation of Direct Air Cooled Condenser for a Large Power Plant”. *IEEE Trans.,* 2009

[4] Z. Hongbin and C. Ling. “Study on Heat Transfer Coefficient of Direct Air Cooled Condenser”. *2010 International Conference on Advances in Energy Engineering. IEEE*, p.p. 235-238.

[5] Z. Honbin and C. Ling. “Study on the Optimal Back pressure of Direct Air Cooled Condenser in Theory”. *IEEE* 2009.

[6] D. Shuangmei and L. Jianmin. “Simulation Analysis and Mathematics Model Study of Direct Air- cooling Condenser”. *IEEE*, September 2008

[7] P. O’Kelly. “Computer Simulation of Thermal Plant Operations”. *Springer*, 2013.

[8] H. Klee, “Simulation of Dynamic Systems with MATLAB and SIMULINK”. 2007, *Ed. CRC Press*.

[9] R. K. Shah, and D. Sekulic. “Fundamentals of Heat Exchanger Desing”. *Ed.John Wiley and Sons*. 2003.