ANALYTICAL MODEL FOR POWER PLANT CONDENSER FOR TRANSIENTS AND OFF-NORMAL OPERATING CONDITIONS

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I. Thangamani, Anu Dutta, G. Chakraborty and A.K. Ghosh
Reactor Safety Division
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**Abstract:** A computer code for power plant condenser dynamic analysis has been developed based on a lumped parameter approach considering time dependent mass and energy conservation equations over the control volumes for the shell side as well as tube side fluids. Effects of heat transfer on condenser structure and hot well level transients were considered in the analysis. Suitable heat transfer coefficient recommended by various standards and codes were employed. The code was used to analyze the condenser performance during steady state as well as transient (load rejection or turbine trip) conditions. The condenser performance is predicted in terms of condenser back pressure, shell side steam temperature and tube side coolant exit temperature with respect to time. As a part of parametric studies, the effect of change in tube side coolant flow rate and inlet temperature was also studied. The analysis predicted that up to 47% of rated coolant flow rate on the tube side (for design conditions), the steam dumping can be continued without condenser isolation. The paper describes the detailed methodology adopted for the condenser modeling and presents the results obtained from the different parametric studies and code validation.

**Keywords/Descriptors:** NUCLEAR POWER PLANTS; TRANSIENTS; PERFORMANCE; STEAM CONDENSERS; HEAT TRANSFER; STEADY-STATE CONDITIONS; TURBINES; COMPUTER CODES; VALIDATION

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सारांश

शोल साइड के साथ-साथ दूरब पत्रहस्त हेतु निगमण वास्तुप योग्य में समय निर्धार
ग्रीमण एवं जल्द परिश्रम सर्वाधिक रूप से विचार करते हुए विशिष्ट प्राचीन प्रमाण
के आधार पर विद्युत संयंत्र संख्यात्मक विश्लेषण हेतु एक कंप्यूटर कोड का
विकास किया गया। विश्लेषण में संचालन सर्वना एवं होटेल सेवा लेवल
द्राफ्टिंग से प्रभाव अतिरिक्त के प्रभावों का अध्ययन किया गया। विभिन्न
पानी एवं कोडों द्वारा संचालन उपयुक्त उपभोक्ता अतिरिक्त सुरक्षा का प्रयोग किया
गया। कोड का प्रयोग स्थिर अवस्था के साथ-साथ (पर अस्तित्वता अवधार
ट्राईन ट्रिप) अस्तित्व परिस्थितियों के दौरान संचालन नियमानुसार विश्लेषण हेतु
किया गया। संचालन नियमानुसार पूर्ववर्तमान संचालन यह वि, तथा वि
वाणिज्य के अनुसार एवं नतीजे शीतलक निकासी तापमान के अनुसार लगाया जाता
है। प्राचीन की अध्ययन के रूप में नतीजे शीतलक प्रवाह दर एवं अतिरिक्त तापमान में
परिवर्तन के प्रभाव का भी अध्ययन किया गया। विश्लेषण द्वारा पूर्ववर्तमान लगाया
गया कि नतीजे की ओर पर (डिजाइन परिस्थितियों के लिए) नियम शीतलक
प्रवाह दर के 47% सर्वाधिक के प्रभाव के सीमा वापस नियंत्रण जारी रखा जा
सकता है। लेख में संचालन प्रतिरूपण हेतु अपनाई गई विद्युत प्रदर्शन का वर्णन
eविश्व प्राचीन क्षेत्रों एवं कोड मान्यता से प्राप्त परिणाम प्रस्तुत है।
ABSTRACT

A computer code for power plant condenser dynamic analysis has been developed based on a lumped parameter approach considering time dependent mass and energy conservation equations over the control volumes for the shell side as well as tube side fluids. Effects of heat transfer on condenser structure and hot well level transients were considered in the analysis. Suitable heat transfer coefficient recommended by various standards and codes were employed. The code was used to analyze the condenser performance during steady state as well as transient (load rejection or turbine trip) conditions. The condenser performance is predicted in terms of condenser back pressure, shell side steam temperature and tube side coolant exit temperature with respect to time. As a part of parametric studies, the effect of change in tube side coolant flow rate and inlet temperature was also studied. The analysis predicted that up to 47% of rated coolant flow rate on the tube side (for design conditions), the steam dumping can be continued without condenser isolation. The paper describes the detailed methodology adopted for the condenser modeling and presents the results obtained from the different parametric studies and code validation.
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1. INTRODUCTION

A computer code for nuclear power plant dynamic analysis is presently under development at BARC. For developing such complicated code, it is always preferable to adopt a modular structure and develop individual modules for each of the associated major equipments/subsystems separately and integrate them subsequently after due validation. As a part of development of the nuclear power plant dynamic analysis code, a computer program has been developed to study the steady state as well as the transient behaviour of the condenser. In nuclear power plant, the exhaust steam from the low pressure turbine enters the condenser under steady state. Besides the condenser needs to perform during load rejection/turbine trip where steam from steam generator is directly dumped into the condenser in order to increase reactor availability and to conserve de-mineralized water. Under such condition, large quantity of superheated steam is flowing in to the condenser and hence the condenser pressure starts rising. If the pressure exceeds the permissible value then condenser is isolated and the excess steam is to be discharged to atmosphere. Feasibility of continuous steam dumping without resorting to condenser isolation also depends upon its other operating parameters such as coolant water flow rates and its temperature at the condenser inlet. While the coolant inlet temperature is seasonal variant, the coolant flow rate could be different depending upon number of pumps operating.

Earlier R.S.Silver [1] investigated the phenomena governing the condensation rates and also devised analytical methods for heat and mass transfer and studied the effects of steam velocity, air & other non-condensable gases on heat transfer. Ciechanowicz [2] derived mathematical model for condenser to study its dynamics and developed two-dimensional dynamic model & one-dimensional dynamic model with which computed the distribution of condenser parameters in the unsteady state. Shida et al [3] developed a numerical method for analyzing the flow and heat transfer based on modified FLIC method using triangular mesh pattern and calculations were performed using upwind two-step Lagrangian and Eulerian time-marching techniques. Zhang et al [4] first proposed a two-dimensional numerical procedure for predicting fluid flow and heat transfer and then the work was extended to a quasi-three dimensional numerical procedure and further the model was improved to predict the existence of non condensable pockets along the tube length. Ormiston, Carlucci et al [5] proposed a model based on finite volume method to predict the condenser performance. Most of the condenser analysis was interested in calculating the flow induced vibrations, heat transfer rates, and pressure
losses and also in doing optimization studies (such as tube arrangement, air cooler position, baffle shape) that leads to enhance the condenser unit performance, reliability and service life.

However Botch et al [6] developed a mathematical model to predict the behaviour of industrial shell and tube condenser under steady state and transient conditions. In which, the condenser was divided into many smaller units and defined by baffle locations and conservation equations were applied for these volumes. Condensation was modeled on the basis of diffusion and the non-condensable gases considered in the analysis were slightly high. All the models, discretised the entire condenser into many volumes and mass, momentum and energy equations were solved. Such model needs extremely high computational time and hence is not suitable when it is intended to use for overall plant dynamics studies. Hence the present model for steady state and transient behaviour is based on lumped parameter approach considering time dependent mass and energy conservation equations over a lumped control volume for the shell side as well as tube side fluids. The heat transfer coefficients are taken from standard codes for power plant condensers such as HEI [7], BEAMA [8], etc.

The code developed was first checked for its predictions of steady state data from the design manual [9] for one of the operating nuclear power plants at Kaiga. The agreement was found to be good with the use of heat transfer coefficient as given in the HEI standard. The transient analysis was carried out to obtain the condenser behavior in terms of condenser back pressure, shell side steam temperature and tube side coolant exit temperature with respect to time. As a part of parametric studies, the effect of change in tube side coolant flow rate was also studied.

The present report deals with the mathematical formulation of the processes occurring in steam condenser, details of the analysis by computer program and the results obtained for load rejection condition in Kaiga Nuclear power plant. Parametric studies have been carried out to identify appropriate operating condition.
2. NOMENCLATURE

\[ A \] : Heat Transfer Area (m^2)
\[ c \] : constant
\[ Cp \] : Specific Heat (J/kg-K)
\[ F \] : Correction factor
\[ g \] : Acceleration due to gravity (m/sec^2)
\[ H \] : Total Enthalpy (J)
\[ h \] : Heat transfer coefficient (W/m^2K)
\[ \bar{h} \] : Specific Enthalpy (J/kg)
\[ k \] : Thermal Conductivity (W/mK)
\[ L \] : Length (m)
\[ m \] : Condensation (or) mass flow rate (kg/sec)
\[ m \] : Mass (kg)
\[ P \] : Pressure (Pa)
\[ Q \] : Heat transfer rate (W)
\[ R \] : Gas constant for steam (J/kg-K)
\[ Re \] : Reynold number
\[ t \] : Time (sec)
\[ T \] : Temperature (K)
\[ U \] : Overall heat transfer coefficient (W/m^2K)
\[ \bar{V} \] : Condenser shell volume (m^3)
\[ V \] : Velocity (m/sec)
\[ W \] : Mass fraction of non condensable
\[ x \] : Steam quality
\[ \mu \] : Viscosity (Pa-sec)
\[ \rho \] : Density (kg/m^3)
\[ \tau \] : Time constant (sec)

**Subscripts**

\[ l \] : Tube material
\[ 2 \] : CW inlet temperature
\[ 3 \] : Cleanliness factor
\[ amb \] : Free stream or bulk
\[ c \] : Condensation
\[ cep \] : Condensate extraction pump
\[ cnd \] : Condenser
\[ cor \] : Correction
\[ cs \] : Condenser structure
\[ cw \] : Cooling water
\[ fg \] : Latent
\[ fwh \] : Feed water heater
\[ hot \] : Hot well
\[ L \] : Liquid
\[ o \] : Overall
\[ cs,i \] : Condenser structure, inside
\[ sat \] : Saturated
\[ stm \] : Steam
\[ str \] : Structure
\[ t \] : Tube
\[ uc \] : Uncorrected
3. SYSTEM DESCRIPTION

In Nuclear Power Plants, condenser is an integral component of secondary side steam cycle and it condenses the steam, in saturated conditions, from LP turbine after expansion. The purpose of condenser in steam cycle is to maximize the efficiency of the cycle, to retain the working fluid within the system instead of throwing away hot demineralized water, to conserve some sensible heat present in the condensate and to convert the two phase working fluid to single phase condensate which benefits less pumping power.

![Diagram of power plant condenser](image)

Fig.1: Schematic diagram of power plant condenser.

The condenser, as shown in figure-1, is a single shell, single pass surface condenser arranged transversely to turbine centerline [9]. Normally condenser works at sub atmospheric pressure i.e. operates under vacuum conditions. Air extraction equipment is used for taking away the accumulated air and other non-condensible to maintain the required vacuum. Overall heat transfer is mainly controlled by cooling water velocity and other parameters such as fouling factor, tube material, condensate etc.

3.1 Steam dumping arrangement

An arrangement of dumping the live steam bypassing the turbine has been provided in case of faults in Turbo Generator (TG) or grid failure etc. This steam dumping practice averts the reactor from unnecessary reactor trips and increases reactor availability and also conserves DM water instead of preferring to discharge steam to atmosphere.
Dumping arrangement has been designed to dump about 70% of full power steam generation into the condenser. Before dumping, the high-pressure steam is depressurized and desuperheated to some extent. Incoming steam enthalpy, pressure and mass flow rate in these conditions are also higher than the normal full power operation. This results in increase of steam pressure, temperature, and circulating water outlet temperature in the condenser. Dumping the steam into the condenser is continued if class IV power (i.e. grid power or station power) is available and condenser back pressure is not more than permissible value [9,10]; otherwise condenser gets isolated from the circuit and steam is discharged into atmosphere. A code is developed to predict the condenser back pressure and other parameters during this scenario.

4. MATHEMATICAL MODELLING

4.1 Mass and energy equation for condenser steam space.

The governing mass and energy equations for the steam space under normal operation are as follows.

\[
\frac{dm_{\text{end}}}{dt} = m_{\text{sim}} - m_{\text{s}} - m_{\text{cs}} \quad \text{(Eq. 1)}
\]

\[
\frac{dH}{dt} = [m_{\text{sim}} \bar{h}_{\text{sim}} - (Q_{\text{str}} + Q_{c} + m_{\text{s}} \bar{h}_{s})] + \bar{V}dP_{\text{end}} \quad \text{(Eq. 2)}
\]

where \( m_{\text{end}} \) is the mass of the steam accumulated in the condenser and \( H \) is the total enthalpy of the steam accumulated in the condenser. \( m_{\text{sim}} \) is the incoming steam mass flow rate and \( \bar{h}_{\text{sim}} \) is the steam enthalpy. \( Q_{\text{str}} \) is the heat transferred to the condenser structure and \( Q_{c} \) is the condensation heat transfer. \( m_{\text{s}} \) and \( m_{\text{cs}} \) is the condensation rate on tube banks and condenser structure respectively.

4.1.1 Normal operation

The heat transfer to the cooling water and condensation rate can be calculated from the following equations during normal operation of the condenser.

\[
Q_{c} = u_{c} A_{c} (T_{\text{sat}} - T_{cw}) \quad \text{(Eq. 3)}
\]

\[
m_{\text{s}} = \frac{Q_{c}}{\bar{h}_{fl}} \quad \text{(Eq. 4)}
\]
Condenser structures (such as plates, tube sheets etc.) are of considerable quantity and it affects the heat transfer during transients. Heat transfer to condenser structure and condensation takes place on the condenser structure can be evaluated from the following equations.

\[ Q_{str} = h_{c,i} \cdot A_{str} \left( T_{sat} - T_{str} \right) \]  
\[ \text{Eq.5} \]

\[ \frac{n \cdot h_{cs}}{\overline{n}_{fs}} = Q_{str} \]  
\[ \text{Eq.6} \]

where \( T_{sat} \) is the average steam saturation temperature of the condenser, \( T_{str} \) is the average temperature of condenser structure and \( T_{cw} \) is the average condenser cooling water temperature. \( U_c \) is the overall condensation heat transfer coefficient for tube banks and \( h_{c,i} \) is the heat transfer coefficient for condensation of steam on the structure. The heat transfer coefficients are discussed separately following section.

4.1.2 Steam dumping condition

During turbine trip or load rejection conditions, the steam from steam generator is dumped into condenser in superheated form. The condensation heat transfer can be calculated using equation 3 and condensation rate can be calculated from the following equation.

\[ \frac{n \cdot h_{cs}}{\overline{n}_{fs}} = Q_{str} \]  
\[ \text{Eq.7} \]

Similarly Equ-5 can be used for heat transfer with condenser structure and condensation on structures during steam dumping conditions can be calculated from the following equation

\[ \frac{n \cdot h_{cs}}{\overline{n}_{fs}} = Q_{str} \]  
\[ \text{Eq.8} \]

4.1.3 Condenser Isolation condition

If the condenser pressure, during steam dumping, exceeds the permissible limit then condenser gets isolated and the trapped superheated steam condenses by free convection. If \( T_{sat} > T_{cw} \) then condensation takes place. The heat transfer rate and condensation rate can be calculated from equation 3 and 4 if steam is saturated or from equation 3 and 7 if steam is in superheated.

If \( T_{sat} < T_{cw} \) then condensation will not takes place. In such conditions only heat transfer takes place, by single phase convection, from condenser to cooling water provided the steam is in superheated state. The heat transfer rate can be calculated from equation 3.
4.2 Energy equation for condenser structure
The heat transfer with condenser structure has been represented by the following transient energy equation.

\[
\frac{dT_{str}}{dt} = \frac{Q_{str} - Q_{amb}}{m_{str}C_{p_{str}}} \quad \text{(Eq.9)}
\]

where \( Q_{amb} \) is the heat transferred out of the condenser structure to ambient by convection and radiation. \( m_{str} \) is total mass of the condenser structure.

4.3 Energy equation for condenser cooling water
The energy equation for condenser cooling water can be given as

\[
\frac{dT_{cw}}{dt} = \frac{Q_{loss}}{m_{cw}C_{p_{cw}}} \quad \text{(Eq.10)}
\]

where \( Q_{loss} \) is the heat rejected by the cooling water to ambient.

\[
Q_{loss} = 2\dot{m}_{cw}C_{p_{cw}}(T_{cw} - T_{in}) \quad \text{(Eq.11)}
\]

\( m_{cw} \) is the mass of the cooling water present in condenser tubes and \( \dot{m}_{cw} \) is the mass flow rate of the cooling water.

4.4 Mass equation for hotwell
The hotwell level transients can be obtained by the mass conservation equation

\[
\frac{dm_{hot}}{dt} = \dot{m}_{c} + \dot{m}_{ct} + \dot{m}_{fwh} - \dot{m}_{cep} \quad \text{(Eq.12)}
\]

where \( \dot{m}_{fwh} \) is the mass flow rate of condensate from the feed heaters are finally added to the condenser hotwell and \( \dot{m}_{cep} \) is mass flow rate through Condensate Extract Pump (CEP). During steam dumping condition, the feed heaters are deprived of bleed steam and hence condensate will not flow from feed heaters to condenser. The following equation can be used to calculate the transients mass flow rate through CEP during steam dumping conditions.

\[
\frac{d\dot{m}_{cep}}{dt} = \frac{\dot{m}_{ct} - \dot{m}_{cep}}{\tau} \quad \text{(Eq.13)}
\]

where \( \tau \) is the time constant.
In an iterative manner, condenser pressure and temperature can be calculated from the mass and energy equations (i.e. Equ 3 & 4). If the steam is in two phase then the pressure can be calculated by the following equation.

\[ P_{\text{cond}} = \frac{x_{\text{cond}}RT_{\text{cond}}}{V} \]  

(Eq.14)

The above equation considers only vapor mass contributing to pressure because of its high specific volume. If the steam is in superheated state then the pressure and temperature can be obtained from equation of state and total enthalpy (obtained from energy equation).

5. HEAT TRANSFER COEFFICIENT

Heat transfer coefficients correlations used in the modeling are described in this section and which involves heat transfer from steam to cooling water through tube and from steam to condenser structure. Heat transfer coefficients used for normal operating conditions, steam dumping conditions and condenser isolation conditions are given below.

5.1 Normal operating conditions

5.1.1 Tube banks

In power plant condensers, high vapor velocities are employed and condensation is also complex. Hence correlations for overall heat transfer coefficients are taken from the standards such as HEI, BEAMA were used in the analysis and these overall heat transfer coefficients are primarily a function of the cooling water velocity through tubes. HEI 5th and 9th Edition [11] give almost same overall heat transfer coefficients without design corrections. But the design correction factors are different for both editions. The equation given by HEI standard given below was used.

\[ U_c = 5.678263c F_1 F_2 F_3 V_{cw}^{0.5} \]  

(Eq.15)

where \( c \) is the constant, \( F_1, F_2, F_3 \) is the tube material correction factor, cooling water inlet temperature correction factor and cleanliness factor respectively.

From open literature, the following correlation by HEI and BEAMA were also examined for analysis.

For HEI

\[ U_c = 2.7 F_1 F_2 F_3 V_{cw}^{0.5} \left( 0.5707 + 0.0274 T_{\text{cond}} - 0.00036 T_{\text{col}}^2 \right) \]  

(Eq.16)
For BEAMA

\[ U_c = 2.15 F_1 F_2 F_3 V_{ce}^{0.5} (0.7586 + 0.0135 \frac{T_{sat} - 0.0001 T_{sat}^2}{T_{sat}}) \]  
\[ \text{(Eq.17)} \]

Recommended design correction factors from HEI 5th Edition for tube material, inlet cooling water temperature and cleanliness were used to get the corrected overall heat transfer coefficient.

5.1.2 Condenser Structure

Condenser structure was considered as a vertical plate (refer figure) and condensation was calculated from the correlation given by Shekriladze et al [12]. The correlation is applicable for the saturated steam under forced convection conditions.

\[ h_{as} = \left( \frac{k_s^2 \rho_s V_{st}}{\mu_s L} \right)^{0.5} \left( \frac{\sqrt{2}}{3} \left( 2 + \left(1 + 16 F_L \right)^{0.5} \right) \right) \left( \frac{1 + \left(1 + 16 F_L \right)^{0.5}}{1 + \left(1 + 16 F_L \right)^{0.5}} \right) \]  
\[ \text{(Eq.18)} \]

where

\[ F_L = \frac{g L \mu_L \overline{F}_L}{k_L V_{st}^2 \Delta T} \] & \[ \Delta T = (T_{sat} - T_{str}) \]

The obtained heat transfer coefficient has been corrected for the presence of non-condensable gases. For steam dumping conditions, the heat transfer coefficient was corrected for

5.2 Steam dumping conditions

During steam dumping conditions the steam is flowing in superheated condition. The heat transfer ratio of superheated and saturated steam under forced convection is slightly greater than one for \( W = 0.005 \) and 100°F degree of superheat [except at lower (\( T_{sat} - T_{cw} \)) values] [13]. The degree of superheat in this analysis is not high, so the correlation used for normal operation can be used for steam dumping conditions also.

5.3 Condenser isolation

5.3.1 Tube banks

Condenser gets isolated if the pressure exceeds the permissible limits thereafter steam starts condense by free convection. Under such conditions the overall heat transfer coefficient can be calculated by conventional method of combining the individual heat transfer resistances.
\[
\frac{1}{U_{sc}} = \frac{1}{h_{st}} + \frac{1}{h_t} + \frac{1}{h_{cw}}
\]  
(Eq.19)

where \(h_{st}\) is the heat transfer coefficient for condensation of steam on tube banks [14] and \(h_{cw}\) is from tube inner surface to cooling water.

\[
h_{st,n} = 1.51 \left( \frac{\mu_s^2}{g \rho_s \kappa_s} \right)^{1/3} Re^{-1/3}
\]  
(Eq.20)

and \(h_{st} = F_{cor} h_{st,n}\)  
(Eq.21)

\(F_{cor}\) is the factor that accounts for the condensation in presence of non condensable gases and degree of superheat [15]. The term \(h_{cw}\) is the heat transfer coefficient between inner tube and cooling water and it can be calculated using Dittus-Boelter equation. The term \(h_t\) is the tube wall coefficient can be obtained from the tube material and dimensions [5].

The corrected overall heat transfer coefficient is

\[
U_c = F_{cor} U_{sc}
\]  
(Eq.21)

5.3.2 Condenser Structure

Condensation on the structures can be calculated from the following equation and the obtained value has been corrected for degree of superheat and presence of non condensable gases.

\[
h_{st,n} = 1.13 \left( \frac{g h_{fg} \rho_{fg}^2 \kappa_{fg}}{\mu_{fg} L \Delta T} \right)^{1/4}
\]  
where \(\Delta T = (T_{sat} - T_{str})\)  
(Eq.22)

and \(h_{st} = F_{cor} h_{st,n}\)  
(Eq.23)

6. METHOD OF ANALYSIS

The major inputs required for the condenser performance analysis are incoming steam conditions, condenser details and condenser cooling water details. The incoming steam conditions include steam mass flow rate, steam enthalpy and steam pressure. Arbitrary condenser pressure (under saturated condition) was given as initial condition and with the above inputs the mass and energy equations were solved by fourth order Runge-Kutta method.
Steady state performance behaviour of the condenser under normal operating conditions was established first. Subsequently transient (i.e. turbine trip or full load rejection transient) was initiated at 100 seconds by changing the steam mass flow rate and prevailing steam properties for dumping conditions in stepwise manner. During transient analysis when the condenser back pressure exceeds the permissible value, the condenser isolation is initiated by reducing steam flow to zero value.

7. RESULTS AND DISCUSSION

One of the important objectives of this exercise was to choose the appropriate correlation for overall heat transfer coefficient from among the different correlations described earlier. In the absence of actual plant data for condenser performance, the reported data in the design manual [9] of condenser and air removal equipment were used as reference values. Figures 2, 3 and 4 show the condenser performance obtained using different standards and correlations at various conditions (such as cleanliness factor, cooling water inlet temperature, cooling water flow rate respectively) and its comparison with design manual data. From this it was found that HEI 5th Edition agrees well with design manual data and therefore it was employed for further analysis.

Transients are initiated after establishing the steady state operating conditions. The analysis was carried out for constant cleanliness factor ($F_3$) of 0.95 and cooling water inlet temperature ($T_1$) of 29.5°C. Figure 5 shows the condenser pressure variation with time for different coolant flow rates through tube side and transients were initiated at 100 seconds. Corresponding to rated flow condition (100% flow) a rise in pressure is noted after 100 seconds which is due to increased steam mass flow rate. The condenser pressure is noted to quickly stabilize at the new value and continues. Thus steam dumping under new stabilized conditions is feasible without any need of condenser isolation. However, it can be seen that for cooling water flow rate below 47.2% (refer Fig-11), steam dumping would lead to condenser isolation. Thus for 47% cooling water flow rate, the condenser gets isolated at 128 seconds as shown in figure 5.

The trapped superheated steam, after condenser isolation, condenses by free convection and continues up to 163 seconds. During this period, the pressure decreases fast at initial time and it decreases exponentially with time. This decrease is due to high condensation rate at initial time and condensation decreases with time and continues until the saturation temperature ($T_{sat}$) is greater than the average temperature ($T_{cw}$) of cooling water. It has observed that the condensation
is not taking place after the saturation temperature reaches the average temperature of cooling water (i.e. after 163 seconds). Further the heat transfer to cooling water is governed by single phase convection, hence the condenser temperature decreases and therefore the pressure decreases.

Figure 6 shows condenser steam mass variations after initiation of transient for different cooling water flow rates. At 100 sec the condenser steam mass increases sharply is due to initiation of transient and it has been simulated by giving step input of increased steam mass flow rate and its conditions. As the condenser steam mass increases, the saturation temperature ($T_{\text{sat}}$) also increases which leads to increase in temperature difference ($T_{\text{sat}} - T_{cw}$). Condensation rate starts increasing because of increasing temperature difference ($T_{\text{sat}} - T_{cw}$) and exceeds beyond the incoming steam mass flow rate. Hence the condenser mass starts decreasing which leads to decrease in saturation temperature and results in low condensation rate. Finally the condenser steam mass stabilizes to new value because of attaining equilibrium between the condensation rate and incoming steam flow rate. At 47% CW flow rate, the rapid decrease in condenser mass is due to isolation of condenser.

Condenser steam temperature variation with time for different flow rates is depicted in figure -7. The steam temperature after 100 second increased due to initiation of transient where superheated steam is dumped in to condenser. At 47% CW flow rate, the condenser temperature starts decreasing steadily after 128 seconds (after condenser isolation) because of condensation of superheated steam. The temperature starts decreasing rapidly after 163 seconds because of single phase convection and then it stabilized to new value.

The figure 8 depicts the mean cooling water outlet temperature variations for different flow rates. Inlet CW temperature given for the analysis was 29.5°C. The sharp rise at 100 seconds is due to initiation of transient and then the temperature gets stabilized to new value. For 47% CW flow rate, the cooling water temperature starts decreasing after 128 seconds is due to isolation of condenser.

Figure 9 and 10 shows the hot well mass and level transients for different CW flow rates. Decrease in CW flow rate affects the overall heat transfer coefficient which further affects the condensation rate and hence the hot well mass decreases first and then stabilizes. Hot well volume was calculated based on the hot well mass and saturated liquid density obtained for the condenser pressure. For low CW flow rates, the condenser pressure raises hence the
corresponding saturated liquid density decreases. Due to decrease in liquid density the condenser hot well level is showing high for low CW flow rates. For 47% CW flow rate, the hot well mass and level starts decreasing after condenser isolation and gets stabilized. The stabilized hot well level is well above the low level limit (242 mm) prescribed for condenser.

The figures 5 to 10 were obtained for constant inlet cooling water temperature of 29.5°C and cleanliness factor of 0.95.

In order to account for the cooling water inlet temperature variations during different seasons, the transient analysis was carried out for different inlet temperatures. Figure 11 shows the limiting cooling water rate for different inlet temperature. If the operating conditions were above this curve then continuous steam dumping is possible and if it is below this curve then steam dumping will lead to condenser isolation.

Thus for, the design conditions, the inlet CW temperature of 29.5°C and cleanliness factor 0.95 it has been found that the condenser is capable to condense the incoming steam continuously even when the CW flow rate was decreased up to 47.2%. For the inlet CW temperature of 35°C, say in summer, it has been found that the condenser is capable to condense the incoming steam continuously for CW flow rate reduction up to 64.14%.

Fig-2: Condenser performance comparison at various cleanliness factors.
Fig-3: Condenser performance comparison at various cooling water inlet temperatures.

Fig-4: Condenser performance comparison at various cooling water flow rates.
**Fig-5:** Condenser back pressure variations for different cooling water flow rates.

**Fig-6:** Condenser steam mass variations for different cooling water flow rates.
-7: Condenser steam temperature

Fig-8: Cooling water outlet temperature variations for different cooling water flow rates.
Fig-9: Hot well mass variations for different cooling water flow rates.

Fig-10: Hot well level variations for different cooling water flow rates.
Fig-11: Limiting CW flow rate for different CW inlet temperatures to avoid condenser isolation.

8. CONCLUSION

Analytical model has been developed for predicting the steady state and transient behaviour of power plant condenser. Various correlations for calculating overall heat transfer coefficients recommended by different guides/open literature sources were also cross compared. The transient analysis carried out for turbine trip condition confirmed the adequacy of condenser performance under normal operating conditions. Regions in which condenser would get isolated due to build up of condenser back pressure beyond the permissible value were also identified.

9. REFERENCE

7. Heat exchanger Institute, Standard for Steam Surface Condensers, Fifth Edition
14. Macabe
ANNEXURE – I : Condenser Details

**Type:** Single shell, single pass, non divided water box, surface condenser and arranged transversely to the turbine centre line. The following parameters have been selected at MCR conditions (220 MWe).

<table>
<thead>
<tr>
<th>Type of condenser</th>
<th>: Horizontal surface type, shell and tube construction &amp; arranged transversely to the turbine centre line.</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of tube passes</td>
<td>: Single pass</td>
</tr>
<tr>
<td>No. of water boxes</td>
<td>: Four</td>
</tr>
<tr>
<td>Total Heat Transfer Area</td>
<td>: 19892.63 m² surface area including 414 tubes as margin for plugging.</td>
</tr>
<tr>
<td>Effective length of tube</td>
<td>: 13.5 m</td>
</tr>
<tr>
<td>Tube size and material</td>
<td>: 22.225 mm OD, 0.711 mm wall thickness, Stainless steel ASTM A-249-Gr TP 316L</td>
</tr>
<tr>
<td>Water velocity in tube</td>
<td>: 2.2 m/sec</td>
</tr>
<tr>
<td>Design cooling water inlet temperature</td>
<td>: 29.5°C</td>
</tr>
<tr>
<td>Cleanliness factor</td>
<td>: 0.95</td>
</tr>
<tr>
<td>Cooling water flow (constant irrespective)</td>
<td>: 55740 m³/hr</td>
</tr>
<tr>
<td>Available margin for tube plugging</td>
<td>: 2% (i.e. 414 tubes)</td>
</tr>
<tr>
<td>Design back pressure</td>
<td>: 63.5 mm of Hg (a) at 29.5°C cooling water inlet temperature and full power operation.</td>
</tr>
</tbody>
</table>

**Steam conditions at condenser inlet for normal operating conditions.**

<table>
<thead>
<tr>
<th>Mass flow rate of steam to condenser</th>
<th>Steam enthalpy at condenser inlet</th>
<th>Steam pressure at condenser inlet</th>
<th>Steam quality at condenser inlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>858.0 TPH</td>
<td>2346.789 kJ/kg</td>
<td>8466 Pa</td>
<td>0.9</td>
</tr>
</tbody>
</table>

**Steam conditions at condenser inlet for steam dumping conditions.**

<table>
<thead>
<tr>
<th>Mass flow rate of steam to condenser</th>
<th>Steam enthalpy at condenser inlet</th>
<th>Steam pressure at condenser inlet</th>
<th>Steam quality at condenser inlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>966.0 TPH</td>
<td>2698.55 kJ/kg</td>
<td>1.0 bar</td>
<td>superheated</td>
</tr>
</tbody>
</table>
ANNEXURE – II : Design data

<table>
<thead>
<tr>
<th></th>
<th>Shell side</th>
<th>Tube side (Water Box side)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid handled</td>
<td>Moist steam from turbine exhaust.</td>
<td>Cooling water from river</td>
</tr>
<tr>
<td>Working pressure (kg/cm² – abs)</td>
<td>0.085</td>
<td></td>
</tr>
<tr>
<td>Design pressure (kg/cm² – abs)</td>
<td>1.0 &amp; Full vacuum</td>
<td>2.0</td>
</tr>
<tr>
<td>Hyd. Test pressure</td>
<td>Water filling upto one foot above final joint of condenser exhaust neck.</td>
<td>3.0</td>
</tr>
<tr>
<td>Design temperature (°C)</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Corrosion allowance</td>
<td>1.5 / 3.0</td>
<td>3.0</td>
</tr>
<tr>
<td>Designed to</td>
<td>HEI std 8th Edition &amp; ASME Sec VIII, Div 1</td>
<td>ASME Sec VIII, Div 1</td>
</tr>
<tr>
<td>Cooling water flow (m³/hr)</td>
<td></td>
<td>55740</td>
</tr>
<tr>
<td>Hot well capacity</td>
<td>Normal level 51.5 m³</td>
<td></td>
</tr>
<tr>
<td></td>
<td>At High level 71 m³</td>
<td></td>
</tr>
<tr>
<td></td>
<td>At Low level 31.5 m³</td>
<td></td>
</tr>
<tr>
<td>Surface area</td>
<td></td>
<td>19500 m²</td>
</tr>
<tr>
<td>Heat load – operating condition</td>
<td>4.452 x 10⁸ kCal/hr</td>
<td></td>
</tr>
<tr>
<td>Heat load – turbine bypass condition</td>
<td>5.890 x 10⁸ kCal/hr</td>
<td></td>
</tr>
</tbody>
</table>