Mathematical models of air-cooled condensers for thermoelectric units

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Abstract

The present paper deals with the use of air-cooled condensers in thermoelectric units, such as steam power plants or combined cycle units. The paper describes the main features of air-cooled condensers, showing the problems related to their use, such as the presence of non-condensable gases inside the steam flow or the condenser performances variability due to ambient conditions. The present study refers to A-frame air-cooled condensers characterized by single-pass and multiple-row arrangement. A phenomenological study has been done in order to calculate the main physical parameters which describe the condenser operating conditions. In particular, three mathematical models have been developed: while both the first and the second model consider a condenser with tubes characterized by the same finned surface, with or without the use of throttling valves upstream of each tube, the third model examines the behaviour of a condenser with different row effectiveness. The three mathematical models permit one also to investigate the possible vapour back flow inside each row of tubes, by calculating the axial coordinate value along each tube at which the condensing steam mass flow rate becomes null; the third mathematical model has been implemented in the Matlab/Simulink environment in order to couple the simulator of the condenser with the whole plant simulation model.

Keywords: air-cooled condenser, non-condensable gases, simulation model.

1 Introduction

Nowadays air-cooled condensers are more and more installed in thermoelectric units, taking the place of water cooled condensers, because their use allows to

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save a lot of cooling water, in accordance with strict environmental rules, and to build power plants in sites far from rivers or the sea [1]. But, as reported by Fabbri [2–4], Kröger [5] and Larinoff *et al* [6], air-cooled condensers are also characterized by several technical and practical problems: the heat exchange process between the ambient air and the condensing steam needs large heat exchange areas which have to be cleaned periodically; the condenser operating pressure depends on the site conditions such as ambient temperature, humidity and windiness; the large number of fans are characterized by high electricity consumptions and noise. Furthermore one of the major problems of a single-pass air-cooled condenser is the accumulation of non-condensable gases inside the finned tubes, as analysed by Fabbri [2], Kröger [5] and Berg and Berg [7]; this undesired phenomenon determines the reduction of the effective heat exchange area and possible condensate freezing in several finned tubes.

The aim of this paper is that of describing three different simplified mathematical models which have been implemented in order to predict the steady-state behaviour of an A-frame air-cooled condenser, considering different design data and operating conditions. Both the first and the second model have been implemented using the Fortran code and are used to study the condenser from a phenomenological point view while the third simulator, tuned-up in the Matlab/Simulink environment, has been created and validated taking into account the design and operating data of a condenser installed in a 400 MW combined cycle power plant still operative in the Italian electricity market.

2 The air-cooled condenser

The air-cooled condenser is a steam-air heat exchanger composed by several modules such as that sketched by fig. 1.

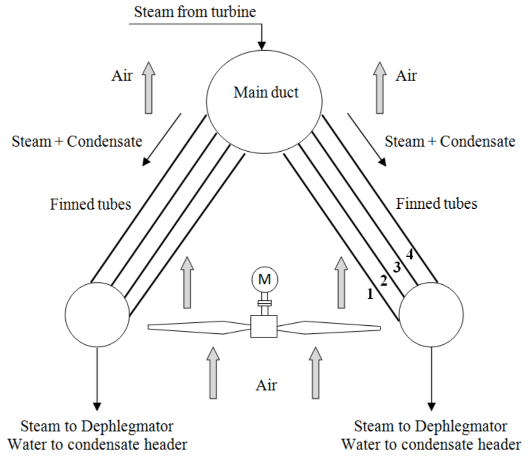

Figure 1: The cross-section of a single module of the air-cooled condenser.

The exhaust steam, leaving the low pressure turbine, comes up to the horizontal main duct and then enters the “parallel-flow” modules flowing down inside banks of finned tubes where its condensation occurs owing to the heat exchange with the ambient air moved by axial fans; the condensate accumulates into two outlet headers, in order to be withdrawn, while the remaining condensing steam goes to the “dephlegmator” characterized by the steam and condensate in counterflow arrangement. The “dephlegmator” modules are also equipped with steam jet air ejectors for the removal of non-condensable gases. Each fan is usually moved by a two-speed motor and the finned tubes are installed in staggered rows; tubes can be round, elliptical or flat and several types of fins are on the market with the aim of optimising the technical performance of condensers subjected to different operating conditions.

3 The mathematical models

The developed mathematical and simulation models are based on the “NTU effectiveness method” which permits to determine the condenser operating parameters beginning from the calculation of the rows effectiveness and the condenser effectiveness, as suggested by Fabbri [2]:

$$E_{row\_i} = \frac{T_{i+1} - T_i}{T_s - T_i} \quad , \quad E_{condenser} = \frac{T_{air\_out} - T_{air\_in}}{T_s - T_{air\_in}} \tag{1}$$

where T_i is the air temperature upstream the ith row while T_s is the condensation temperature. Furthermore the system thermal balance equation is:

$$\dot{M}_{air} \cdot c_{p_{air}} \cdot \left(T_{air\_out} - T_{air\_in}\right) = \dot{M}_s \cdot r \cdot x_s \tag{2}$$

where $\dot{M}_{air}$ and $\dot{M}_s$ denote the air and steam flow rates, $c_{p_{air}}$ is the air average specific heat at constant pressure, r the condensation latent heat and x_s the steam quality at the condenser inlet. In accordance with the model proposed by Fabbri [2], the steam condensation rate per unit of tube length is assumed constant along each tube:

$$\dot{m}_{s_i} = \frac{\dot{M}_{air} \cdot c_{p_{air}} \cdot \left(T_{i+1} - T_i\right)}{r \cdot x_s \cdot L} \tag{3}$$

where L denotes the tubes length.

The first two models analyse the steady-state behaviour of a single module of the condenser, coupled with the corresponding fan, and characterized by four rows of finned tubes while the third one refers to a condenser composed of 21 modules (18 parallel-flow and 3 dephlegmator) with three tubes rows. The first step of the analysis has been the design calculation of the condenser main operating parameters: the steam mass flow rate entering each row, the pressure drop inside the tubes, the air intermediate temperatures T_i, the rows effectiveness and the axial coordinate value a_i along each tube at which the condensing steam mass flow rate becomes null and the pressure reaches the minimum value. All models assume the pressure drop per unit of tube length proportional to the square of the current steam flow rate and consider the same pressure drop between the inlet and the outlet of each finned tube, the main duct and the lower headers being in common.

3.1 The first model

The main inputs of the first simulator are: the air inlet temperature, the exhaust steam flow rate at the turbine outlet, the pressure in the main duct and the geometrical data of the condenser. The model considers four tubes rows having the same effectiveness. For this case, in absence of non-condensable gases, the first rows, which are those in contact with the coldest air, have pressure minimum values inside the tubes instead of at their outlet ($a_i < L$); as a consequence, there is vapour back flow from the other rows to the first ones. If the presence of non-condensable gases is considered, these gases accumulate along the first rows from the a_i axial coordinate to the tubes outlet and determine the reduction of the heat exchange effectiveness.

The following considerations refers to the simulation of a parallel-flow module belonging to a condenser characterised by the next design data:

- Total thermal flux = 218.05 MW_{th};
- Total steam mass flow rate = 97.51 kg/s;
- Nominal condensation temperature = 41.53 °C;
- Steam quality at the condenser inlet = 0.9308;
- Ambient temperature = 15 °C, ambient pressure = 100.4 kPa;
- 20 modules each composed of 6 bundles;
- 4 rows per bundle;
- Condensing steam mass flow rate per bundle = 0.81 kg/s.

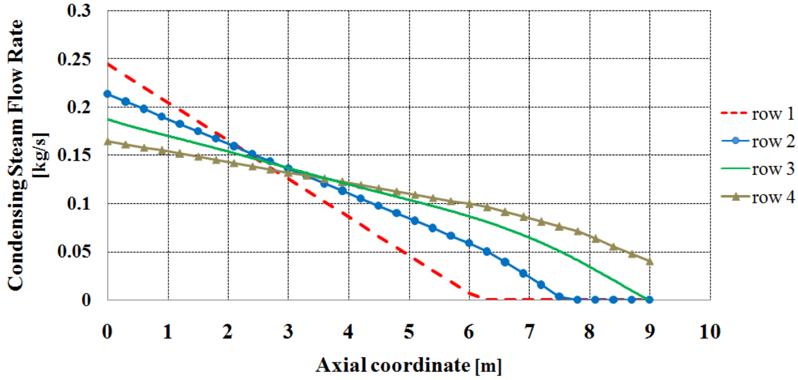

Figure 2: Condensing steam flow rates (E_1=E_2=E_3=E_4).

Assuming that each row has an effectiveness equal to 35%, the following air intermediate temperatures have been calculated by means of eqn. (1): T_2 = 24.2 °C, T_3 = 30.31°C, T_4 = 34.24°C, T_5 = 36.79°C. The condensing steam mass flow rate and pressure along each tube row are plotted by fig. 2 and fig. 3, considering the presence of non-condensable gases inside the flow. It is possible to notice that the steam condenses completely inside both row 1 and row 2, which operate in better conditions in terms of air lower temperatures and higher steam mass flow rates; the condensing steam flow rate becomes null at the end of the 3rd row while the steam which enters the 4th row does not condensate completely inside the tubes and goes to the dephlegmator unit. If non-condensable gases had not been considered, the steam exiting the 4th row would have gone back into the tubes of the first two rows. So it is clear that the vapour back-flow does not occur inside the tubes where non-condensable gases accumulate.

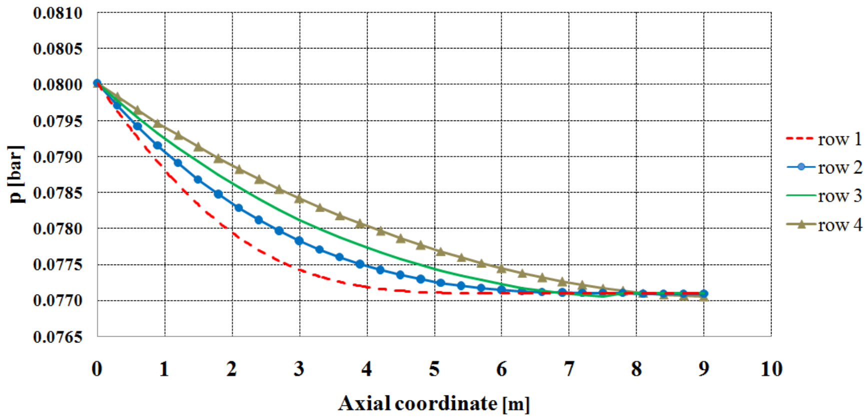

Figure 3: The pressure along each tube (E_1=E_2=E_3=E_4).

Being the operating conditions depicted by fig. 2 and fig. 3 not optimal, in terms of heat exchange effectiveness and steam flow rate distribution between the tubes rows, it has been necessary to find better design data for the condenser in order to optimise its technical performance; it would be better to have the same flow rate at each tube inlet, the same air temperature increase through each tubes row and the steam complete condensation at the end of tubes ($a_i \geq L$), in order to avoid any vapour back flow and non-condensable gases accumulation.

3.2 The second model

Different solutions have been adopted in order to optimise the condenser operating conditions. The first is relative to a condenser with tubes characterized by the same finned surface and consists in installing a gauged throttle upstream of each tube with the aim of completing the steam condensation not before the tube end. The condensing steam mass flow rate and pressure along each tube row are plotted by fig. 4 and fig. 5, considering gauged pressure drops upstream of each tube and row effectiveness equal to 35%.

The results show that the steam flow rate at the tubes inlet is not the same for all rows and it is higher for row 1 in contact with cold air; as a consequence, this solution does not seem to be the most adequate. It appears clear that the best

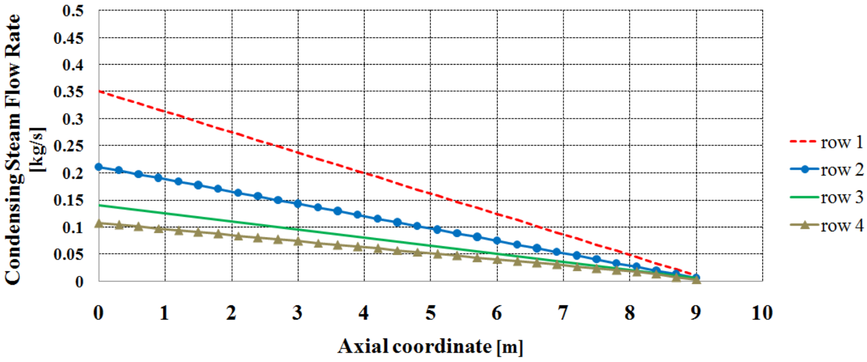

Figure 4: Condensing steam flow rates (E_1=E_2=E_3=E_4, gauged pressure drops).

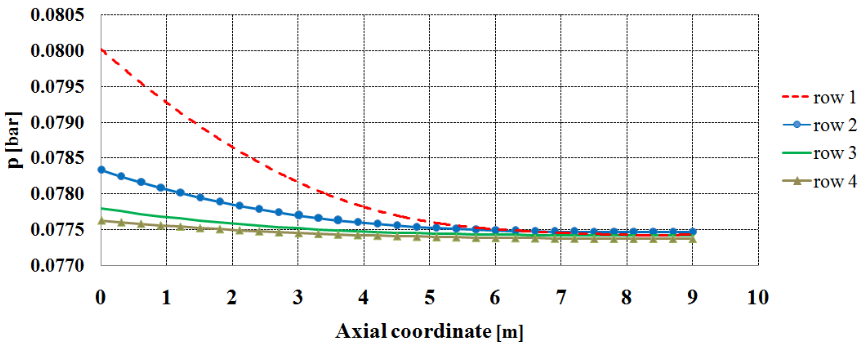

Figure 5: The pressure along each tube (E_1=E_2=E_3=E_4, gauged pressure drops).

solution is that of designing condensers with tubes rows characterized by different effectiveness values, as nowadays proposed by manufacturers.

3.3 The rows effectiveness optimization

From the considerations drawn in 3.1 and 3.2 it derives that a well-designed multi-row air-cooled condenser needs differently finned tube rows; in particular, it is necessary to increase the fin surface of the more external rows which are those exposed to a hotter air flow. The optimization of the rows effectiveness for the condenser described in 3.1 has determined the following best values: E_1 = 20.7%, E_2 = 26.2 %, E_3 = 35.5 %, E_4 = 55.0%. Considering these nominal values, the condensing steam mass flow rate and pressure along each tube become the same for all rows, as shown by fig. 6. Furthermore, there is not vapour back flow and the air temperature increase is the same through each row.

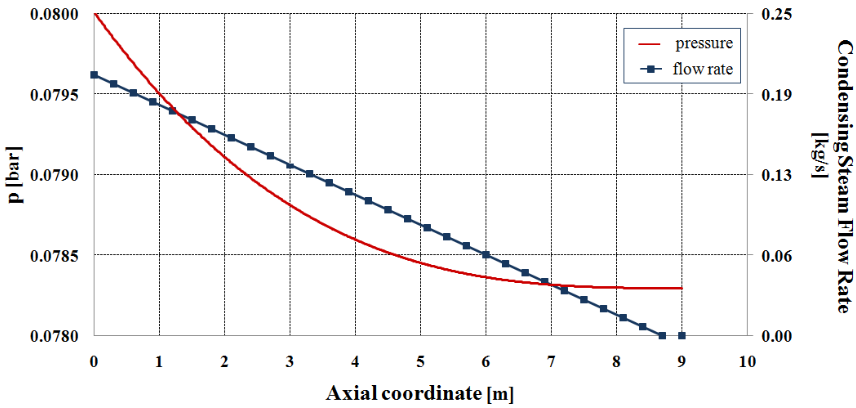

Figure 6: The pressure and steam flow rate along each tube ($E_1 \neq E_2 \neq E_3 \neq E_4$).

3.4 The third model

The third simulation model, implemented in the Matlab/Simulink environment, is called "operative" because it is going to be linked to the dynamic simulator of the whole combined cycle power plant; it is used to calculate off-line the condenser performance parameters in off-design operating conditions. In this case the steam flow rate is an important datum while the pressure downstream the turbine is unknown. In the following the main equations used to simulate a single module of the condenser are reported.

The main inputs of the simulator are:

- the ambient temperature (T_1) and pressure (p_1);
- the exhaust steam mass flow rate and quality at the condenser inlet;
- the condenser design operating parameters: condensation temperature ($T_{s\text{-}nom}$ = 37.90°C), ambient temperature and pressure ($T_{1\text{-}nom}$ = 15°C, $p_{1\text{-}nom}$ = 101.6 kPa), exhaust steam mass flow rate and quality ($\dot{m}_{s-nom}$ = 5.52 kg/s, $x_{s\text{-}nom}$ = 0.9256), air volume flow rate and density ($\dot{Q}_{air-nom}$ = 581 m^3/s, $\rho_{air\text{-}nom}$ = 1.228 kg/m^3), air intermediate temperatures ($T_{2\text{-}nom}$ = 20.74°C, $T_{3\text{-}nom}$ = 26.48°C , $T_{4\text{-}nom}$ = 32.22 °C), rows effectiveness ($E_{1\text{-}nom}$ = 25.07%, $E_{2\text{-}nom}$ = 33.45%, $E_{3\text{-}nom}$ = 50.27%), condenser global effectiveness ($E_{condenser}$ = 75.2%);
- the condenser geometrical data.

The heat balance equation for the ith row can be written as:

$$\dot{M}_{air} \cdot c_{p_{air}} \cdot \left(T_{i+1} - T_1\right) = U_i \cdot S_i \cdot \Delta T_{lm\_i-i+1} \tag{4}$$

where U_i is the global transmittance, S_i the heat exchange area and ΔT_{lm} the log-mean temperature difference. Since the NTU (Number of Transfer Units) corresponds to the ratio [8]:

$$NTU_i = \frac{U_i \cdot S_i}{\dot{M}_{air} \cdot c_{p_{air}}}, \tag{5}$$

after a little algebra one gets:

$$E_{row\_i} = 1 - e^{-NTU_i} . \tag{6}$$

Considering constant the average values of U_i, S_i and $c_{p\ air}$, it derives that the product ($\dot{M}_{air} \cdot NTU_i$) must be kept constant and so the effectiveness values can be calculated from the current air mass flow rate, which depends on the fan rotational speed, as follows:

$$E_{row\_i} = 1 - e^{-\frac{\dot{M}_{air-nom} \cdot NTU_{i-nom}}{\dot{M}_{air}}} . \tag{7}$$

The next step is the calculation of the condensation temperature T_s: it is determined by eqn. (2), after having expressed the condensation latent heat as a polynomial function of T_s and the air outlet temperature T_4 as a function of E_1, E_2, E_3, T_1 and T_s. The T_s calculated value permits to determine the condensation pressure, by means of the water-steam properties, and the air intermediate temperatures T_2, T_3 and T_4, by means of eqn. (1). As a consequence the steam condensation rates per unit of tube length can be determined by eqn. (3).

As proposed by Fabbri [2], it is necessary to distinguish the tubes characterized by vapour back flow (bf) from those where the steam does not condense completely (nbf). In the first case, the steam mass flow rate at the axial coordinate x along the ith tube is calculated by:

$$\left(\dot{M}_{s_i}\right)_{bf} = \dot{m}_{s_i} \cdot \left(a_i - x\right) \tag{8}$$

where a_i is the length of the tube segment in which the steam flows from the main duct to the lower header. The steam flow rate crossing the outlet end of each tube is equal to:

$$\dot{\Pi}_{s_i} = -\dot{m}_{s_i} \cdot \left(L - a_i\right) . \tag{9}$$

On the other hand the steam flow rate at the axial coordinate x along the tubes without back flow is calculated by:

$$\left(\dot{M}_{s_i}\right)_{nbf} = \dot{m}_{s_i} \cdot \left(L - x\right) + \dot{\Pi}_{s_i} . \tag{10}$$

The calculation of the pressure drop between the inlet and the outlet of each tube has been done referring to the model reported by Fabbri [2]. In particular, for the tubes with back flow the following equation has been adopted:

$$\left(\Delta p_i\right)_{bf} = \frac{k_i \cdot \dot{m}_{s_i}^2}{3} \cdot \left(2a_i^3 - 3a_i^2 L + 3a_i L^2 - L^3\right) \tag{11}$$

k_i denoting a friction coefficient which depends on the tubes material.

On the other hand, the tubes without back flow are characterized by a pressure drop equal to:

$$\left(\Delta p_i\right)_{nbf} = \frac{k_i \cdot \dot{m}_{s_i}^2}{3} \cdot \left(L^3 - 3a_i L + 3a_i^2 L\right) . \tag{12}$$

Since all tubes rows have the inlet and outlet plena in common, the pressure drops have to be the same:

$$\left(\Delta p_i\right)_{bf} = \left(\Delta p_i\right)_{nbf} \quad \forall i . \tag{13}$$

In order to calculate pressure drops within the Matlab/Simulink model efficiently, eqn. (12) and (13) have been implemented in non-dimensional form, as done by Fabbri [2] by proper choice of reference variables, as a function of the steam flow rates defined by eqn. (8) and (10). Assuming the steam total flow rate at the exit of tubes without back flow equal to the one flowing back to the other tubes, that is equivalent to not considering a dephlegmator module downstream, the model calculates the pressure drop between the inlet and outlet plena. Once this value has been determined, it is possible to solve eqn. (11) and (12) in order to calculate the axial coordinate a_i along each tube at which the condensing steam mass flow rate becomes null.

As a consequence, the steam flow rate at each row inlet can be evaluated by the following equation:

$$\dot{M}_{s_i} = \dot{m}_{s_i} \cdot a_i \,. \tag{14}$$

In order to consider the presence of a dephlegmator unit downstream of the parallel-flow module, it is necessary to modify the tubes length as follows:

$$L^* = \alpha^{-1} L \quad , \quad \alpha < 1 \tag{15}$$

where the coefficient α is connected with the steam flow rate to the dephlegmator:

$$\dot{M}_{s_i\_dph} \propto (1-\alpha) \cdot \dot{M}_{s_i} \,. \tag{16}$$

Consequently, the steam total flow rate at the parallel-flow module inlet is equal to:

$$\sum_i \dot{M}_{s_i} = \sum_i \dot{m}_{s_i} \cdot a_i = \sum_i \dot{M}_{s_i\_dph} + \sum_i \dot{M}_{s_i\_c} \tag{17}$$

where:

$$\sum_i \dot{M}_{s_i\_c} = \alpha \cdot L^* \cdot \sum_i \dot{m}_{s_i} \tag{18}$$

is the steam flow rate which condenses inside the parallel-flow module.

The rows length increase is equivalent to impose the steam complete condensation inside the system composed of one parallel-flow module and one dephlegmator unit connected in series.

3.5 The condenser performance curves

The aim of the present analysis has been that of determining, by the third simulator, the condenser performance curves which show, on the pressure – steam flow rate plane, how the condensation pressure p_s is influenced by both the air inlet temperature T_1 and the fans rotational speed.

Figure 7 reports the performance curves considering all fans running at full speed. It is clear that, for the same value of steam flow rate, the higher is T_1 the higher is the condenser pressure; on the other hand, considering the same temperature T_1, the lower is the steam flow rate the lower is the condenser pressure. The curves in fig. 7 are not parallel lines but their slope increases with the air inlet temperature.

Since this type of air-cooled condensers are equipped with fans running at two different speed values, respectively 100% and 50% of the nominal rotational speed, it has been interesting to calculate the performance curves also for the case of all fans at half speed, that is an operative condition to which the condenser is subjected both during startup and cold days. The results are plotted in fig. 8: the curves have the same shape as those depicted by fig. 7 but are characterized by a higher slope. In fact in this case, considering the same values of air inlet temperature T_1 and steam flow rate, the air flow rate is lower and so the air temperature T_4 at the condenser outlet is higher; as a consequence, the condensation temperature increases as well as the pressure.

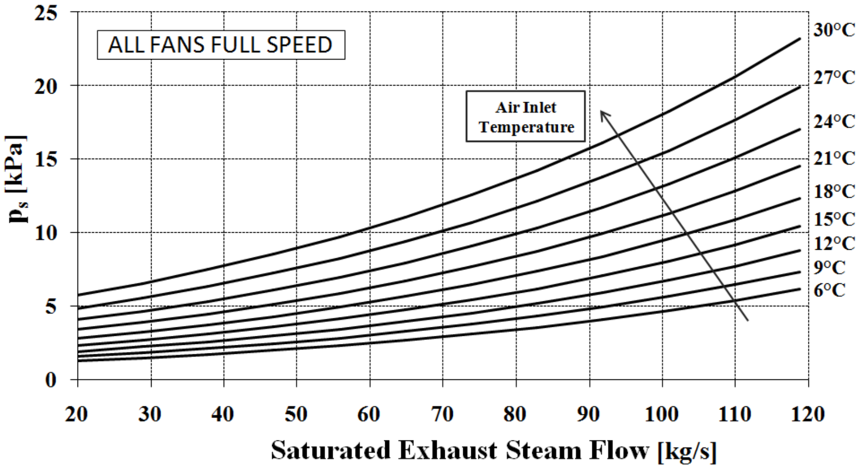

Figure 7: The condenser performance curves with all fans at full speed.

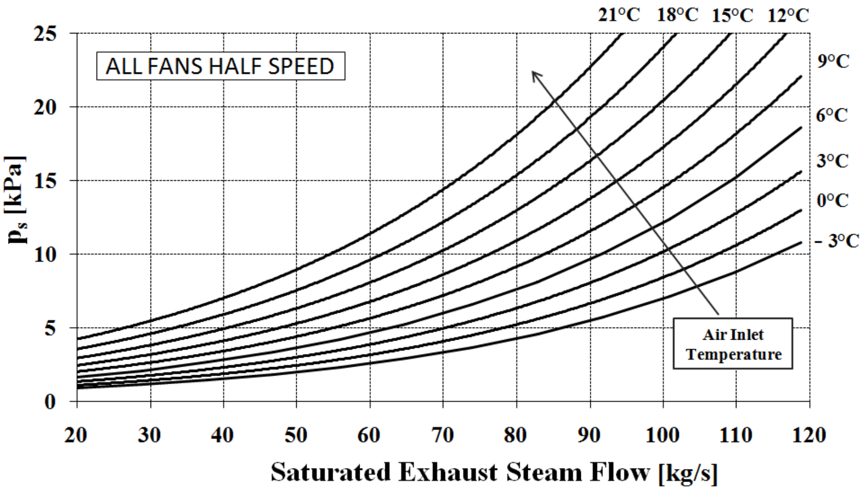

Figure 8: The condenser performance curves with all fans at half speed.

The curves obtained by the third simulation model have been compared with those supplied by Ansaldo Energia SpA, an Italian company which installs air-cooled condensers in its power plants sites. The model validation has been done with reference to different steady-state operating conditions and it is important to remark that the calculated performance curves are in good agreement with manufacturers ones.

4 Conclusions

The mathematical and simulations model described in this paper have been developed to study a multi-row A-frame air-cooled condenser for combined cycle power plants. The analysis has been focused on the simulation of the condenser behaviour under different steady-state operating conditions with the aim of evaluating the influence of the air inlet temperature on the system performance. The problem of the non-condensable gases accumulation inside the finned tubes rows has been analysed from a phenomenological point of view and some technical solutions have been proposed in order to reduce the risk of their formation, as well as the vapour back flow inside the tubes.

The condenser performance curves have been determined considering two different fans operating conditions, full or half speed, and varying both the air inlet temperature and the saturated exhaust steam flow rate. The calculated curves validation phase has confirmed the model validity over a wide range of operating conditions.

The next step of the study is going to be the development of a dynamic simulation model to study the condenser behaviour under different transient conditions; this model will be integrated with the combined cycle power plant dynamic simulator and validated with experimental data coming from the plant.

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