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DEVELOPMENT OF OFF-DESIGN MODELLINGS FOR DISTINCT HEAT EXCHANGERS COUPLED TO AN INTERNAL COMBUSTION ENGINE

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Abstract. *The present work is part of a Research and Development project, whose objective is to increase net power production of a Wärtsilä 20V34SG engine by cooling and dehumidifying the intake air. In order to accomplish this goal, a cooling coil on the compressor's upstream is proposed, thereby, delivering cooled air with less moisture to the engine. Moreover, a plate heat exchanger is proposed to cool the engine cooling water before it goes to the charge air coolers, and a waste heat recovery heat exchanger drives an absorption chiller. This work presents several off-design simulations for distinct heat exchangers, which are developed in the Engineering Equation Solver software and are based on heat and mass transfers fundamentals. The performance of the thermal system is simulated in function of the different air conditions, according to ambient conditions variation on the region that the power plant is located. Results show that cooling coil outlet air, at critical conditions, has higher humidity and temperature than expected, however, the engine could still operate safely under these conditions. Additionally, the Charge Air Cooler simulations demonstrated that engine cooling water inlet temperature could be further reduced without occurring condensation, providing additional cooling to combustion air in order to mitigate the consequences of higher temperature in the cooling coil air outlet.*

Keywords: *Off-design, Condensation, Heat and Mass Transfer, Simulation, Engineering Equation Solver*

1. INTRODUCTION

In order to analyze the performance of heat exchangers, several methods are known, such as Log-Mean Temperature Difference (LMTD), effectiveness and number of transfer unit (e-NTU) and temperature effectiveness and number of transfer unit (P-NTU). Otherwise, the method of e-NTU is the most frequently used when the objective is to calculate the output temperature of the fluids once their input value is known in the heat exchanger (Shah and Sekulic, 2003).

Several research efforts have been conducted to develop mathematical modeling in order to simulate and analyze the off-design operation of heat exchangers. Chatpozoulou *et al*. (2019) studied off-design behavior of different heat exchangers in an organic Rankine cycle recovering residual heat from an internal combustion engine. Assato *et al*. (2008) made an off-design modelling of the HRSG and compared with the manufacturer data, demonstrating that, for this type of equipment, the deviation of the results with the manufacturers data was only 0.5%.

Campbell *et al.* (2021) simulated a Wärtsilä 20V34SG engine of 8.7 MW by using the GT-POWER software, varying saturated air temperature at the cooling coil outlet and the air temperature at the Charge Air Cooler (CAC) outlet in relation to the dew temperature in the intake manifold. In his work, the engine power output was targeted to 10 MW by decreasing the air temperature. An important result regarding this operating condition is a limitation due the maximum pressure inside the piston cylinders of approximately 186 bar. The temperature of the saturated air at the output of the cooling coil must not reach 13.8°C, while air temperature difference between dry-bulb and dew temperatures in the intake manifold is 8°C. It was verified that, for inlet air temperature above 13.8°C, pressure in the cylinders would rise above the maximum limit.

In order to perform a technical and economic analysis of a thermal systems, Miotto (2021) dimensioned, analyzed and compared two experimental system configurations for engine inlet air conditioning, recovering heat from the engine exhaust gases. The alternative configuration proposed in that work is used in this paper, as well as the design parameters developed of the heat exchangers which are set as input data along the methodology.

In this paper, four heat exchangers are studied by using mathematical modeling. Several simulations, developed in Engineering Equation Solver, are carried out and analyzed separately. The contribution of this work is to study and analyze the off-design performance of each heat exchanger in function of the ambient variation. Some limitations are also investigated, such as condensation formation in the intake air manifold.

2. CASE STUDY

The case study is part of a Research and Development project (ANEEL PD-06483-0318/2018) sponsored by Thermoelectric Viana S.A. (TEVISA) and Linhares Generation S.A (LGSA). The thermoelectric power plant of LGSA, called Luiz Oscar Rodrigues de Melo (UTE LORM) is powered by 24 internal combustion engines of Wärtsilä model 20V34SG and each generator set produces an amount of shaft power around 8.7 MW. The location of this facility is at Linhares, Espírito Santo, Brazil. In summary, an experimental thermal system composed by several heat exchangers is coupled to one engine, as shown in Figure 1. The components of the new thermal system are a single-effect absorption chiller, a cooling coil, an auxiliary heat exchanger (HX), a waste heat recovery (WHR) unit and a cooling tower.

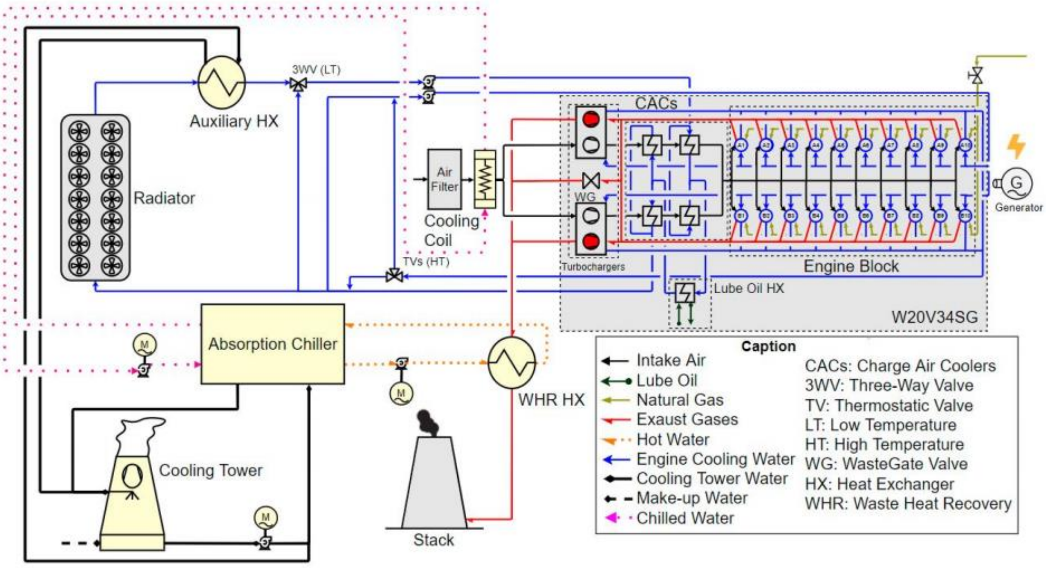

Figure 1. Flowsheet of the experimental thermal system.

3. METHODOLOGY

The methodology of this work is based on mass and heat transfer fundamentals and thermodynamic principles. Moreover, the Charge Air Cooler (CAC), the auxiliary HX and the WHR HX are modeled with e-NTU methods presented by Shah and Sekulic (2003), whereas the cooling coil is modeled with the methodology from Stoecker and Jones (1985), whose main assumption is a total wet surface. However, the proposed methodology to predict condensation in the CAC is a combination of the models proposed by Shah and Sekulic (2003) and by Mansour and Hassab (2012).

For the off-design modelling, the heat transfer area of each heat exchanger was calculated and posteriorly treated as a fixed value. Then, with the heat transfer area defined, the thermodynamics states at the outlet of each flow stream were obtained for each input condition. It is important to notice that the global heat transfer coefficient, the convection coefficient, the LMTD, the heat transferred and the effectiveness were calculated parameters and are not considered fixed on the results, except for the convection coefficient for the cooling coil modelling.

The following assumptions are set to simplify the modelling: (i) steady state, (ii) kinetic and potential energy are negligible, (iii) adiabatic heat exchanger, (iv) temperature of the fluids are constant along the transversal direction along the flow, (v) thermal resistance of the wall neglected, (vi) coefficient of heat transfer constant along HX, (vii) area of heat transfer uniform and (viii) velocity and temperature at the entrance of HX is uniform along the transversal section.

3.1 Cooling Coil

The cooling coil is necessary in order to reduce dry-bulb temperature and absolute humidity of inlet air at the entrance of the compressor. Firstly, it is necessary to define the design point of this thermal equipment. The nominal operating conditions of the working fluids (air and water) were set by Miotto (2021), considering to attend 99% of the annual demand, as it can be seen in Table 1. Secondly, the heat transfer area is calculated for this design point, which is treated as a fixed parameter during the off-design simulations.

Table 1. Design parameters of Cooling Coil.

Design Parameters	Values
Inlet temperature of the chiller water (°C)	7.0
Outlet temperature of the chiller water(°C)	12.6
Flowrate of the chiller water (kg/s)	22.91
Dry bulb temperature of the inlet air (°C)	31.1
Inlet design air enthalpy (kJ/kg$_{dry\ air}$)	88.41
Dry bulb temperature of the outlet air (°C)	12.5
Flowrate of the air (kg/s)	16.16

With the design parameters defined, it is necessary to introduce the mathematical modeling that is utilized. Stoecker and Jones (1985) divide the cooling coil in several increments, applying equations of heat transfer fundamentals combined with thermodynamic principles to each increment while considering the wet surface assumption. Thereby, the following equations show the mass and energy balance, by considering Q_i as the transfer heat of each increment, m_{air} and m_w is the mass flow rate of the air and water, respectively, $h_{air;i}$ and $h_{sm;i}$ the enthalpies of humid air and saturated humid air at the wet surface, in that order. The infinitesimal heat transfer area A_i is calculated for each increment, while the temperature of the water $T_{w;i}$ and the temperature of the saturated air at the wet surface $T_{sm;i}$ are calculated by Eq. (1) to (4), where h_c and h_r are the convection coefficient at the air side and water side, respectively, and c is the specific heat capacity for the water side.

$$Q_i = \dot{m}_{air} \cdot (h_{air;i+1} - h_{air;i}) \tag{1}$$

$$Q_i = \frac{h_c \cdot A_i}{c_p} \cdot \left[\frac{(h_{air;i+1} + h_{air;i})}{2} - \frac{(h_{sm;i+1} + h_{sm;i})}{2}\right] \tag{2}$$

$$Q_i = \frac{h_r \cdot A_i}{A_{ratio}} \cdot \left[\left(\frac{T_{sm;i+1} + T_{sm;i}}{2}\right) - T_w\right] \tag{3}$$

$$Q_i = \dot{m}_w \cdot c_p \cdot (T_{w;i+1} - T_{w;i}) \tag{4}$$

With the mathematical model and the design condition defined, it is possible to calculate the design area, using the proposed method. Thereby, the total transfer area is 917.72 m², considering a cooling coil with the area ratio A_{ratio} of 23 and the convection coefficient on the air side of 60 W/m²-K. After the design parameters were defined, as well as the geometry of the cooling coil, it is necessary to characterize the off-design parameters. Therefore, a typical behavior of cooling coil shown by Stoecker and Jones (1985) is used, where for a same inlet enthalpy, the outlet enthalpy will be always constant, independent of the air relative humidity. Thereby, data from INMET (2020) is used and organized in a graphic of enthalpy in function of dry bulb temperature as shown in Figure 2. Note that the maximum enthalpy this year was 88.68 kJ/kg and the minimum was 56.83kJ/kg for relative humidity at around 80% and 60%, respectively.

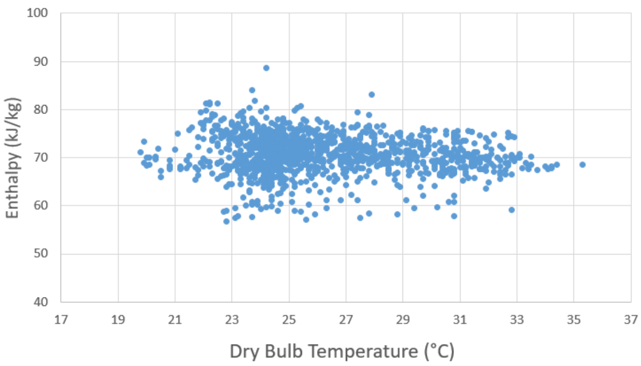

Figure 2. Enthalpy in function of dry bulb temperature for Linhares, ES, from 01/26/2019 to 01/25/2020.

3.2 Auxiliary Heat Exchanger

The auxiliary HX is a plate type heat exchanger whose objective is to deliver the outlet cooling engine water at 4°C below the dew point of the air at the exit of the low temperature CAC. Similarly to the cooling coil, the design parameters are also necessary in order to define a heat transfer area, based on Miotto (2021), as shown in Table 2.

Table 2. Design parameters for the auxiliary heat exchanger.

Design Parameters	Values
Inlet temperature of the cooling engine water (°C)	48.09
Outlet temperature of the cooling engine water (°C)	33.55
Flow rate of the cooling engine water (kg/s)	22.91
Inlet temperature of the tower water (°C)	28.36
Outlet temperature of the tower water (°C)	33.9
Flow rate of the tower water (kg/s)	68.37

Thereby, the mathematical model of plate heat exchanger is used, considering the Chevron angle of the heat exchanger of 45°C and e-NTU method (Kakaç *et al.*, 2012). The Nusselt number (Nu) and the convection coefficient h may be calculated for both fluids sides, using the Reynolds number (*Re*) and Prandtl number (*Pr*), using Eq. (5) and Eq. (6).

$$Nu = C_h \cdot Re^n Pr^{1/3} \tag{5}$$

$$Nu = \frac{\mathrm{h} \cdot \mathrm{D_h}}{\mathrm{k}} \tag{6}$$

Where D_h is the hydraulic diameter, which, along with n, are auxiliary coefficients dependent on the Reynolds number and Chevron angle. With the Nusselt number and the convection coefficient defined, it is possible to calculate the global coefficient and the NTU. Then, the effectiveness ε can be determined by Eq. (7).

$$\varepsilon = \frac{exp\left[\left(1 - \frac{C_{min}}{C_{max}}\right) \cdot NTU\right] - 1}{exp\left[\left(1 - \frac{C_{min}}{C_{max}}\right) \cdot NTU\right] - \frac{C_{min}}{C_{max}}} \tag{7}$$

With the design parameters and the proposed model delineated, the heat transfer area is calculated equal to 33 m² with a total of 27 plates. In addition, it is necessary to set the off-design parameters at entrance of the auxiliary heat exchanger.

Firstly, it is important to define approach as the difference between of the outlet temperature of water in the cooling tower and the wet bulb temperature of ambient air (Vedavarz *et al.*, 2007). However, the cooling tower was not modeled in this work, and it is mandatory to understand its behavior since the auxiliary heat exchanger inlet water is linked to the cooling tower. Therefore, an assumption was made to consider the cooling tower's approach as constant and equal to 3ºC. Finally, cooling tower's outlet water temperature, between 01/26/2019 and 01/25/2020, was calculated considering data of ambient air conditions from INMET (2020), with results shown in Figure 3. The outlet water temperature ranges between 16.6°C and 31ºC, as seen in Figure 3. During the simulations, the tower's cooling water flow is also varied in order to analyze the sensitivity of thermodynamic states, for a range of 60 to 70 kg/s.

3.3 Waste Heat Recovery Heat Exchanger

The waste heat recovery heat exchanger is a tube matrix with continuous fins and its goal is to utilize exhaust gas heat to produce hot water, in order to power the absorption chiller. Considering the design parameters set by Miotto (2021) and with a model based on Shah and Sekulic (2003), it is possible to define the heat transfer area to be used in off-design simulations. Table 3 presents the design conditions of the waste heat recovery heat exchanger.

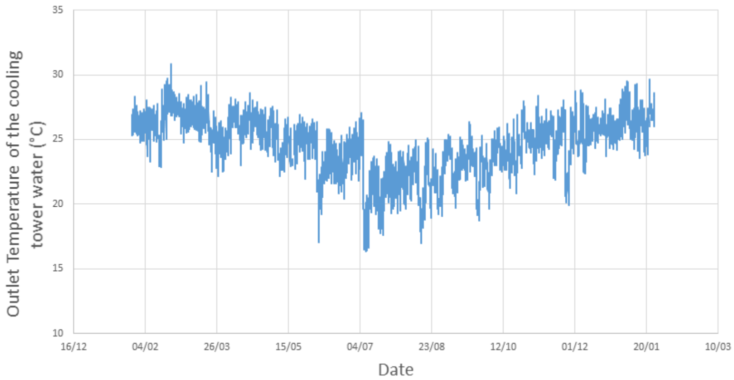

Figure 3. Outlet temperature of the cooling water tower for Linhares, ES 26, from 01/26/2019 to 01/25/2020.

Table 3. Design parameters for the waste heat recovery heat exchanger

Design Parameters	**Values**
Inlet temperature of the hot water (°C)	88
Outlet temperature of the hot water (°C)	98
Flow rate of the hot water (kg/s)	28
Inlet temperature of the gases (°C)	390
Outlet temperature of the gases (°C)	113.5
Flow rate of the gases (kg/s)	4

With the objective of modelling the fins of the heat exchanger, it is mandatory to calculate the fin efficiency. For this purpose, Shah and Sekulic (2003) demonstrate a method that divides a section of the area along the fin and calculate the effectiveness and area of each infinitesimal element considering an annular fin.

Considering the staggered arrangement for WHR, it is necessary to define the efficiency of one fin n_f in function of the infinitesimal areas A_i and A_j, as shown by Eq. (8). The efficiency of an annular fin n_i and n_j are dependent on the radius $r_{e,j}$ and $r_{e,i}$ radius. The coefficient *a* is an auxiliary variable equal to 2 for staggered arrangement. In order to determine the global efficiency n_0 of fin arrangement, it is necessary to define the fin area A_f and total area A_t, consequently it is calculated using Eq. (9).

$$\eta_f = \frac{\sum A_i \cdot \eta_i + a \cdot \sum A_j \cdot \eta_j}{\sum A_i + a \cdot \sum A_j} \tag{8}$$

$$\eta_0 = 1 - \eta_f \cdot \left(\frac{A_f}{A_t}\right) \tag{9}$$

Additionally, to determine outlet temperatures using inlet conditions as input, it is crucial to calculate the convection coefficient of the gases and water. The Colburn coefficient *j*, shown in Eq. (10), is used for the model, where *Nr* is the number of tube rows, Re_{dc} is the Reynolds number, p_f is fin pitch, D_h is the hydraulic diameter and X_t is the transverse tube pitch. The coefficients *c3*, *c4*, *c5* and *c6* are described in Shah and Sekulic (2003). With the Colburn coefficient defined, the convection coefficient *h* can be calculated by Eq. (11), in function of flow rate *G*, thermal capacity c_p and the Prandtl number *Pr*.

$$j = 0{,}086(Re_{dc})^{c_3}(\mathrm{Nr})^{c_4}\left(\frac{p_f}{d_c}\right)^{c_5}\left(\frac{\mathrm{p}_f}{D_h}\right)^{c6}\left(\frac{p_f}{X_t}\right)^{-0{,}93} \tag{10}$$

$$h = \mathrm{j} \cdot \mathrm{G} \cdot \mathrm{cp} \cdot (\mathrm{Pr})^{\left(-2/3\right)} \tag{11}$$

Finally, it is possible to calculate the effectiveness of the heat exchanger using Eq. (12), where C_{max} and C_{min} are the maximum and minimum capacitances of the fluids.

$$\varepsilon = 1 - \exp\left[\left(\frac{C_{max}}{C_{min}}\right) \cdot NTU^{0{,}22} \cdot \left(\exp\left(-\left(\frac{C_{min}}{C_{max}}\right) \cdot NTU^{0{,}78}\right) - 1\right)\right] \tag{12}$$

With the mathematical model described as well as the design parameters, it is possible to obtain the heat transfer area, which, for this case, is equal to 954.3 m², assuming tubes of 10.7 x 9.4 mm diameters and a horizontal pitch of 24.5 mm.

Finally, the gases flow rate will be increased, ranging from 3.5 to 4.5 kg/s, due to a damper allocated downstream of the WHR. In addition, the temperature of the gases and the outlet hot water will be fixed for the off-design modelling due a control system that will be installed to maintain these conditions.

3.4 Charge Air Cooler

The CAC, distinct from the others heat exchangers, is already integrated to the engine. Fin and tube geometry were previously measured on a heat exchanger in the power plant. Moreover, the CAC has 2 circuits, the Low Temperature (LT) and the High Temperature (HT). Cooling engine water goes into the LT CAC first, cooling the air directly before the combustion. Furthermore, after the first stage, the water is used to cool the engine oil, before flowing to the HT CAC, where it will cool air flowing directly after the compressor. Therefore, intake air goes first into the HT CAC, following to the LT CAC.

Since the objective of the project includes cooling the air in this heat exchanger, it is very important to understand its behavior in function of the engine cooling water inlet temperature. Initially, is necessary to define the design parameters on Table 4.

Table 4. Design parameters for the charge air cooler.

Design Parameters	**Values**
Inlet temperature of the cooling engine water on CAC LT (°C)	35.55
Outlet temperature of the cooling engine water on CAC HT (°C)	72
Flow rate of the cooling engine water (kg/s)	22.91
Inlet temperature of the air on CAC HT (°C)	203.7
Outlet temperature of the air CAC LT (°C)	44
Flow rate of the air (kg/s)	16.16
Area HT (m^2)	84.39
Area LT (m^2)	111.2

With the design parameters defined, the same methodology proposed by the WHR heat exchanger are also set to the CAC. However, according to Shah and Sekulic (2003), the Colburn correlation used has limitations, and the analyzed CAC has a distance between tube columns of 19.3 mm, being one millimeter smaller than the minimum limit. Moreover, the CAC LT has 12 columns and HT side has 18 columns, while the correlation limitations is of 6 columns.

However, Xie *et al.* (2009) demonstrated that for heat exchangers of 6 or more columns, the Nusselt number does not increase significantly and consequently, the convection coefficient remains approximately constant. While the distances between columns are also out of range, initial comparisons of simulation results and performance data available from the power plant showed that, for this case study, the model presented an error of 3.7%, according to Miotto (2021).

Additionally, to predict if condensation will occur, it is necessary to study the heat transfer in more detail. For this purpose, the convection coefficient is calculated by the methodology proposed by Shah and Sekulic (2003) for flat fin over a staggered arrangement, and these results are implemented in the model of cooling coil partially dry proposed by Mansour and Hassab (2012). If air inlet temperature at each row is lower than the dew temperature, the surface is considered wet and Eq. (13) and Eq. (14) are used, where (*ΔNTU0*) is the number of transfer unit for each row.

$$T_{air;i+1} = \frac{1-\frac{\Delta NTU0}{2}}{1+\frac{\Delta NTU0}{2}} \cdot T_{air;i} + \frac{\Delta NTU0}{1+\frac{\Delta NTU0}{2}} \cdot T_{sm;i} \tag{13}$$

$$T_{sm;i} = \frac{-(R+1{,}44)+\sqrt{(R+1{,}44)^2+0{,}184\cdot(h_{air;i}+R\cdot T_{w;i}-10{,}76)}}{0{,}092} \tag{14}$$

Otherwise, the surface of the heat exchanger is considered dry, using Eq. (15) and Eq. (16).

$$T_{sm;i} = \frac{T_{air;i+R\cdot T_{w;i}}}{R+1} \tag{15}$$

$$Q_i = \dot{m}_{air} \cdot c_p \cdot (h_{air;i+1} - h_{air;i}) \tag{16}$$

Since the design parameters and the mathematical model were defined, it is substantive to obtain the off-design parameters. Firstly, the air flow rate will be increased from 14 to 16.16 kg/s, due to an increase in shaft power. Also, the temperature of the cooling engine water at entrance of the CAC LT will be reduced from 35.55ºC until condensation starts to occur. Thereby, it will be possible to predict the minimum reach of the engine cooling water temperature that will ensure a safe engine operation.

4. RESULTS AND DISCUSSION

In this section, the thermodynamics states of the work fluids of each heat exchanger are presented and discussed in function of the off-design parameters and of the effects of these results on engine performance.

4.1 Cooling Coil

Considering the ambient data and the methodology of cooling coil totally wet proposed by Stoecker and Jones (1985), the results are presented of the cooling coil in function of inlet enthalpy of the air and relative humidity, as demonstrated in Figure 4.

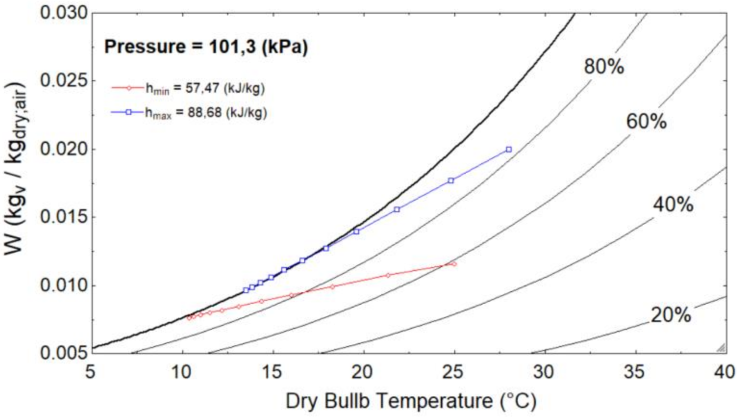

Figure 4. Cooling coil thermodinamics states in function of the maximum and minimum inlet enthalpy of the air registred and the relativy humidty.

Although the coil is designed to condition the ambient air to a wet-bulb temperature of 12.5°C, observe that the air outlet condition depends directly on air inlet conditions. Considering the ambient data from INMET (2020), outlet air absolute humidity varies between 0.007648 and 0.009657 $kg_v/kg_{dry\text{-}air}$, with dry bulb temperatures of 10.5°C and 13.49°C respectively. Besides that, Campbell *et al.* (2021) showed that the engine will operate under safe conditions for the air saturated at 13.49ºC of dry-bulb temperature at the exit of coil, since the operational limit for the engine operating at 10 MW is 13.8ºC, considering saturated air and air temperature difference between the dry-bulb and the dew temperatures in the intake manifold of 8°C.

4.2 Auxiliary Heat Exchanger

Using the plate heat exchanger modelling demonstrated by Kakaç *et al.* (2012) and the off-design parameters demonstrated, is possible to obtain the results of inlet temperature of the cooling engine water in the auxiliary heat exchanger, as shown in Figure 5.

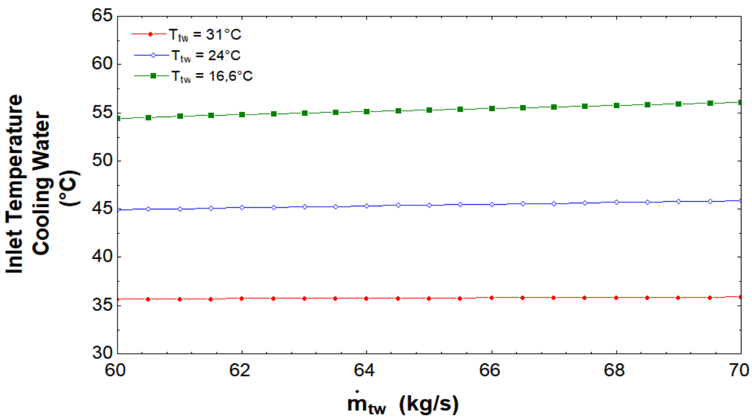

Figure 5. Inlet temperature of the cooling engine water in function of the flow mass and the inlet temperature of the tower water at auxiliary HX.

It is possible to notice that the radiator needs to deliver cooling engine water to the auxiliary heat exchange at about 55°C and 35ºC, depending on the ambient conditions. If the radiator outlet water temperature is greater than the demonstrated temperatures, then the auxiliary HX outlet water temperature will be greater than 33.55ºC, compromising the cooling of combustion air in the CAC and, consequently, operational conditions of the engine.

4.3 Waste Heat Recovery Heat Exchanger

Using the methodology proposed by Shah and Sekulic (2003) for the off-design model, considering a tube matrix in a fin matrix with staggered arrangement, inlet temperature of the hot water is calculated, in function of the gases flow rate, as shown in Figure 6.

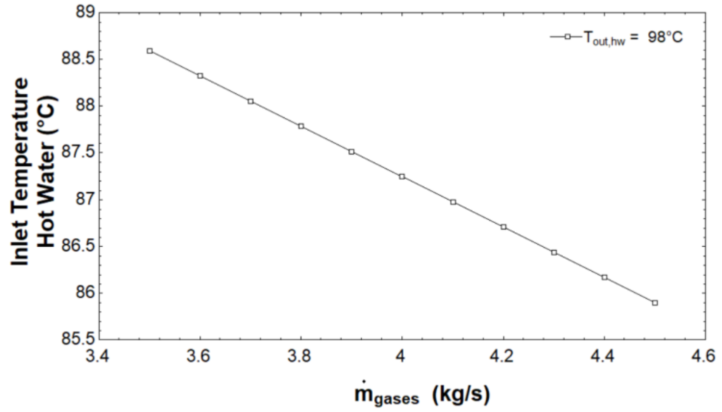

Figure 6. Inlet temperature of the hot water in function of the flow mass of the gases at WHR HX.

Analyzing Figure 6, it is possible to notice that the water temperature at the inlet of the TCR ranges between 85.9 to 88.59°C. This inlet difference of the temperature has to be balanced for the control system in order to maintain the heat transferred in the WHR, without affecting the absorption chiller.

4.4 Charge Air Cooler

Using the methodology proposed by Shah and Sekulic (2003), and considering the CAC as a tube matrix in a fin matrix with staggered arrangement, as well as the off-design parameters in function of the increases of air flowrate, it is possible to calculate the difference of the temperature between the air and the dew point (ΔT_{CA}) at the outlet of the low temperature circuit as shown in Figure 7.

Note that the CAC would be able to cool the air to a ΔT_{CA} about 5.7°C for a flow of 14 kg/s of air and an inlet cooling water temperature of 35.55°C. For air flow of 16.16 kg/s, it is noted that the difference of the temperature between the air and the dew point, at the CAC air outlet, reaches 7.5°C. Since the limitation due the maximum pressure inside the piston cylinders of 186 bar is reached when ΔT_{CA} approaches 8°C, with saturated air at 13.8ºC as cooling coil outlet, it is possible to verify that the engine will operate safely.

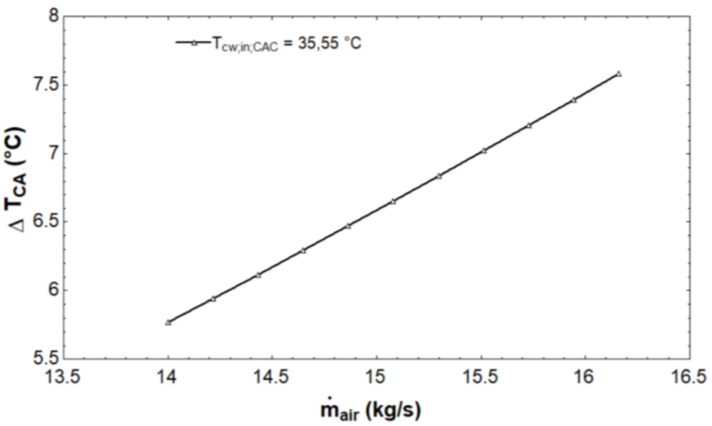

Figure 7. Difference of the temperature of the air at air collector and the dew point in function of the flow mass of the air at CAC outlet.

The thermodynamic states of the air in CAC, considering the predicted maximum absolute humidity of 0.009657 $kg_v/kg_{dry\text{-}air}$, are shown in Figure 8, considering the engine output of 10 MW and inlet air flow engine of 16.16 kg/s.

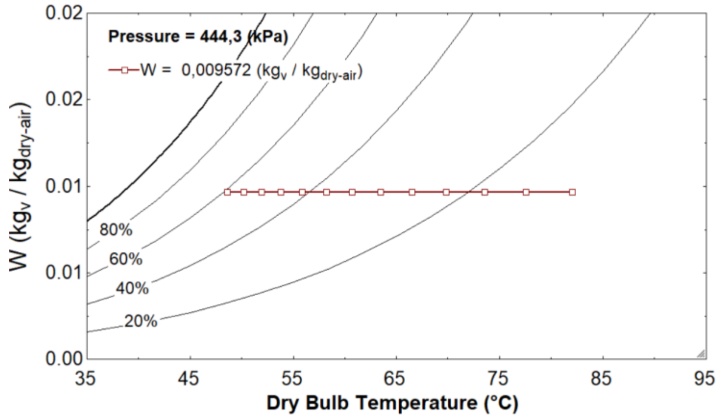

Figure 8. Thermodynamic states of the air in CAC considering temperature of the inlet cooling engine water at CAC LT of 35,55ºC and the maximum humidity at exit of the cooling coil.

Analyzing Figure 8, it can be seen that there will be no condensation of vapor from the air in the CAC. However, it is necessary to understand at which point would start to occur condensation. Thereby, keeping all other parameters constant, the cooling water temperature is lowered by steps of 1°C, until condensation occurs. The result for the higher temperature condition in which water vapor condensed from the air occurred in the CAC is shown in Figure 9.

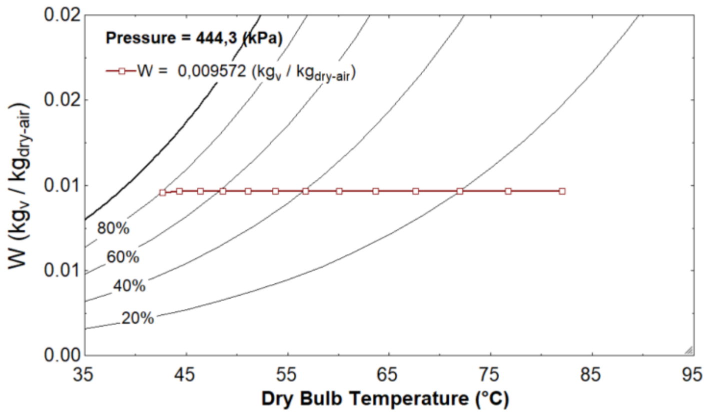

Figure 9. Thermodynamic states of the air in CAC considering temperature of the inlet cooling engine water at CAC LT of 27ºC and the maximum humidity at exit of the cooling coil.

Considering inlet water temperature equal to 27°C, the CAC outlet air has lower humidity ratio than the inlet air. Consequently, there is formation of condensate in the heat exchanger, thus indicating a limitation of the CAC cooling process. Therefore, the engine cooling water temperature must be greater than 27°C to prevent condensate forming in the CAC.

5. FINAL COMMENTS AND OUTLOOK

The purpose of this work is accomplished successfully, presenting off-design results for distinct heat exchanger and discounting the impact of each result for the engine and chiller.

It is concluded that, due to its sizing being made to attend 99% of demands in the period analyzed, the cooling coil outlet air in critical ambient conditions will have higher absolute humidity than initially expected. However, the maximum saturated air temperature at the cooling coil outlet, calculated in the off-design modeling, is still less than 13,8°C, indicating that the engine operates safely for thermal analysis even for the worst air conditions.

The Charge Air Cooler analysis demonstrates that, even for the highest absolute humidity calculated at the outlet of the cooling coil, there is no condensation of vapor contained in the air in the CAC. However, when evaluating the critical point at which condensation would occur in the CAC rows, it is observed that engine cooling water temperature, at the CAC LT inlet, can be reduced from 33.5°C to 27°C. Therefore, additional cooling of engine air in CAC would reduce peak pressure in cylinders, consequently the engine would operate at safer conditions without reaching the maximum limit pressure of 186 bar inside cylinders.

The study carried out at the auxiliary heat exchanger showed that the radiator should deliver engine cooling water to the auxiliary HX in a range of 35 to 55ºC, depending on the ambient condition, thus making it possible for the engine cooling water to come out at 33.55ºC at the auxiliary HX outlet. However, if the radiator does not provide sufficient cooling, the engine cooling water temperature would rise above 33.55ºC at the auxiliary HX outlet, and therefore, it would impair the cooling capacity of the CAC. Consequently, combustion air would enter the cylinders at higher temperature, increasing peak pressure. Therefore, engine power output could be reduced to prevent the peak pressure inside cylinders of reaching its maximum limit.

Analogous WHR HX analysis shows working temperatures, as a function of gas flow, so that the WHR HX outlet temperature remains at 98ºC. Results demonstrate the need for a damper, to control gas flow, in order to maintain outlet water temperature fixed.

For future works, it is suggested:

- Comparations of the results with performance data obtained after the implementation of each HX at the power plant.
- An integrated modelling with all the thermal system components communicating with each other.

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8. RESPONSIBILITY NOTE

The authors are solely responsible for the printed material included in this paper.