Analytical Model of the Refrigeration System for a Cooling in a Hygroscopic Cycle Power Plant

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INTRODUCTION

The Hygroscopic Cycle Technology (HCT) has similar characteristics to the Rankine Cycle. However, this technology is characterized by working with hygroscopic components. The hygroscopic compounds are capable of absorbing water in the form of vapor or in a liquid state.

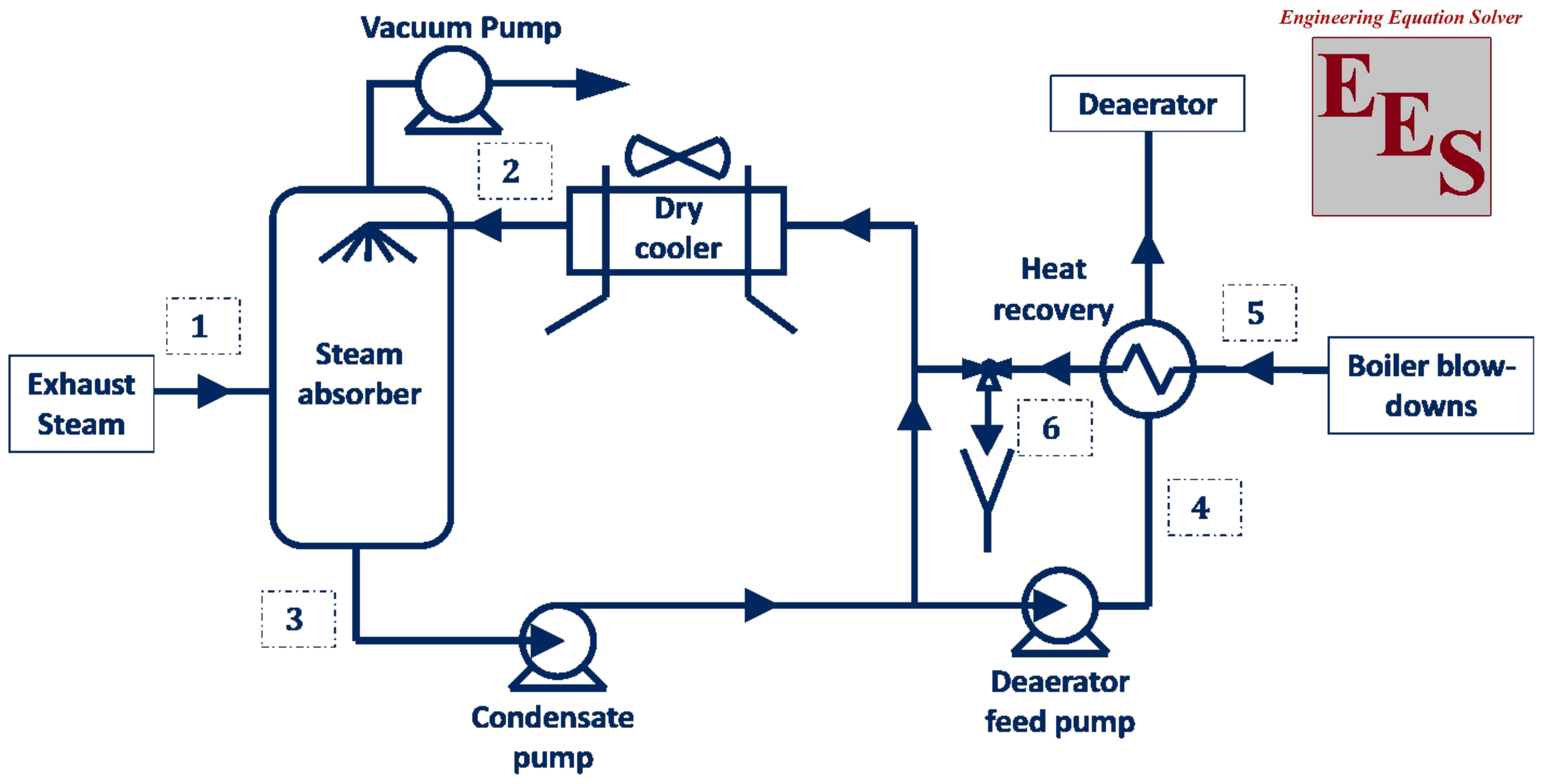
For same condensing pressure, the cooling temperature is 13 ºC higher in a real HC than in Rankine Cycle at an industrial scale. For this reason, cooling solely by air is possible in this cycle since there is enough difference between the hot fluid and the air temperatures..

No research to date have directly focused on the dry-coolers used in the HC, requiring a more specific design to enhance the dissipation of large amounts of thermal power. The objective of the study was to develop an analytical model that would allow knowing in detail about the configuration and internal functioning of the HC dry-cooling system.

METHODOLOGY

PILOT PLANT (GIJÓN, SPAIN)

MODEL DIAGRAM



MODEL PARAMETERS

Pipe length (m)	5.67
Tube diameter (mm)	12
Tube material	Copper
Fins thickness (mm)	0.2
Fin pitch (mm)	2.2
Fin material	Aluminum
A_{total} (m^2)	1023.12
A_{fins} (m^2)	984.48
L (m)	1132.95
N_{fins}	539500
r_{fins} (m)	0.018
η_{vent} (%)	70

LMTD + NTU

$$Q = LMTD \cdot U \cdot A \cdot F$$

$$LMTD = \frac{(T_{hi}-T_{co})-(T_{ho}-T_{ci})}{ln\left(\frac{T_{hi}-T_{co}}{T_{ho}-T_{ci}}\right)}$$

$$NTU = -R \cdot ln\left(1+\frac{ln(1-R\cdot P)}{R}\right)$$

$$P = \frac{T_{ho}-T_{hi}}{T_{ci}-T_{hi}}$$

$$F = \frac{ln\left(\frac{1-R\cdot P}{1-P}\right)}{NTU\cdot\left(\frac{1}{R}-1\right)}$$

$$R = \frac{T_{ci}-T_{co}}{T_{ho}-T_{hi}}$$

Thermal Power Absorbed by Air

$$\dot{Q} = \dot{m}_c \cdot cp_c \cdot (T_{co} - T_{ci})$$

Thermal Power Transfered by Liquid

$$\dot{Q} = \dot{m}_c \cdot cp_c \cdot (T_{co} - T_{ci})$$

Pressure Drop in Air Flow

$$\Delta P_c = Eu \cdot \rho_c \cdot v_c^{\,2}$$

Power Consumed by Fans

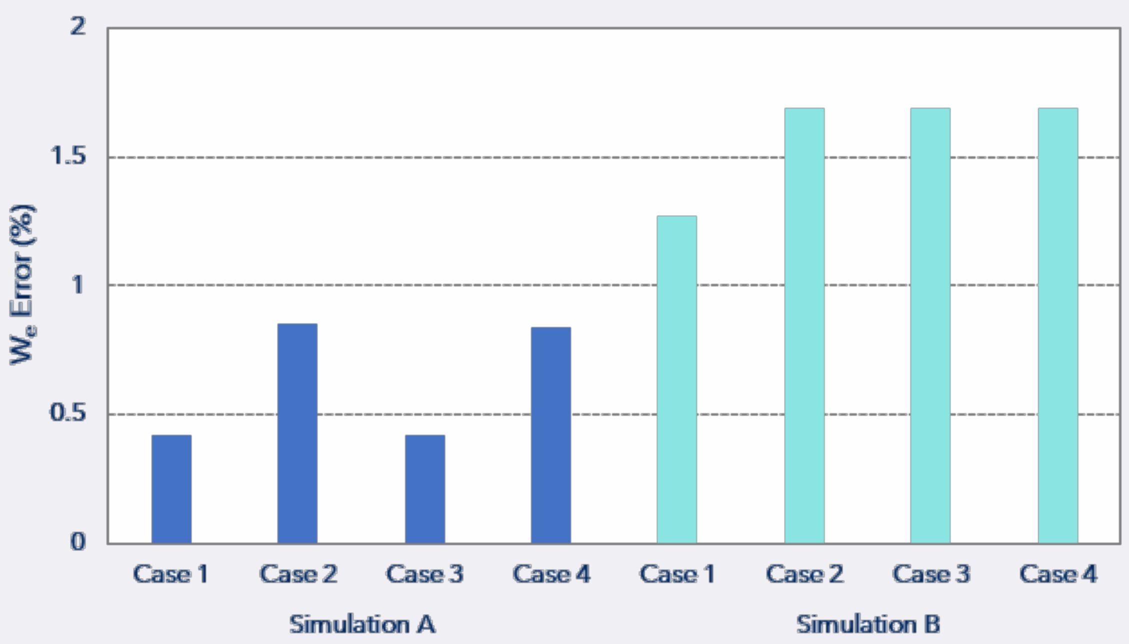
$$W_m = \Delta P_c \cdot \dot{V}_c$$

$$W_e = \frac{W_m}{\eta_{fan}}$$

RESULTS

Experimental data from pilot plant

	Case 1	Case 2	Case 3	Case 4
T_a (°C)	10.02	17.84	24.15	35.10
T_{hi} (°C)	25.15	33.62	33.91	50.90
T_{ho} (°C)	18.10	26.31	32.60	43.91
$\dot{V}_h$ (m^3/h)	18.00	18.05	18.06	18.06
$\dot{V}_c$ (m^3/h)	51,239.8	51,240.0	51,240.2	51,240.3
W_e (kW)	2.36	2.36	2.37	2.37



Simulation A

INPUT SET
- Inlet T of hot and cold fluids
- Outlet T of hot fluid

	Case 1	Case 2	Case 3	Case 4
T_{hi} (°C)	25.15	3°3.62	39.91	50.90
T_{ho} (°C)	18.10	26.31	32.60	43.91
T_{ci} (°C)	10.02	17.84	24.15	35.10
T_{co} (°C)	18.00	26.95	34.06	44.22
$\dot{V}_h$ (m^3/h)	18.00	18.10	18.70	18.80
$\dot{V}_c$ (m^3/h)	51,238.3	51,238.9	51,239.2	51,239.9
U (W/ m^2K)	25.10	25.31	25.66	26.13
F	0.81	0.81	0.8	0.8
η_{fins}	86.10	86.13	86.21	86.36
$\dot{Q}$ (kW)	139.00	139.08	139.35	139.90
W_e (kW)	2.37	2.38	2.38	2.39

Simulation B

INPUT SET
- Inlet T of hot and cold fluids
- Mass flow of hot fluid

	Case 1	Case 2	Case 3	Case 4
T_{hi} (°C)	25.15	33.62	39.91	50.90
T_{ho} (°C)	18.50	26.87	32.89	44.10
T_{ci} (°C)	10.02	17.84	24.15	35.10
T_{co} (°C)	18.90	26.69	33.83	44.56
$\dot{V}_h$ (m^3/h)	18.00	18.05	18.06	18.06
$\dot{V}_c$ (m^3/h)	51,238.1	51,239.0	51,240.0	51,241.2
U (W/ m^2K)	25.30	25.31	25.66	26.13
F	0.81	0.81	0.81	0.80
η_{fins}	86.10	86.13	86.21	86.36
$\dot{Q}$ (kW)	140.07	141.32	141.88	141.91
W_e (kW)	2.39	2.40	2.41	2.41

CONCLUSIONS

The model developed for the simulation of the dry-coolers has been validated with the experimental results obtained in the pilot plant. These results allowed us to contrast the design to later extrapolate the results to other working conditions.

The model of the dry system for the refrigeration of the HCT will be very useful for the development of a global model of the cycle and for the simulation under different working conditions. That will, be done in future works in order to improve the efficiency and the general performance of the technology.

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