

Energy-Efficient Free Heating and Free Cooling Integration for Offshore Substation

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Abstract

Offshore substations present unique HVAC challenges due to constrained layouts, exposure to harsh marine environments, and critical limitations on auxiliary power consumption. This study investigates the integration of free heating and free cooling strategies as a means to optimize energy efficiency in extreme cold climates. Conventional winter operation relies exclusively on electrical heating to condition incoming fresh air, resulting in significant power consumption ($\approx 136 \text{ kW}$ for -20°C ambient). In the proposed approach, chilled water is circulated through AHU cooling coils even during winter operation, effectively preheating fresh air and reducing electrical heater load. Analytical evaluation demonstrates that using chilled

water supply at 7°C reduces electrical heating demand to 38 kW , while further utilization of chilled water return from FCUs at 12°C lowers it to 19 kW , corresponding to energy savings of 72% and 86%, respectively. The study employs ϵ -NTU and LMTD methods to validate heat transfer performance and psychrometric calculations to assess indoor air conditions. Results confirm that integrated free heating and free cooling can achieve substantial auxiliary power reduction, making it a viable and practical solution for energy-efficient HVAC operation in offshore substations. The findings provide a framework for deploying low-grade thermal recovery in marine environments, with implications for reducing operational costs and improving overall system sustainability.

Index Terms- Free Cooling, Free Heating, HVAC, ϵ -NTU, LMTD, psychrometric analysis, offshore.

1. INTRODUCTION:

Cooling and heating of offshore substations constitute a complex engineering challenge due to the criticality of the infrastructure and the harsh marine environment in which these systems operate. Offshore wind turbine generator (WTG) substations are subject to significant constraints related to spatial layout, structural weight, continuous exposure to corrosive atmospheric conditions, and auxiliary power consumption. Consequently, the development of energy-efficient thermal management strategies is a key design objective. Offshore installations are typically exposed to extreme climatic conditions, characterized by high ambient temperatures and humidity during summer and very low temperatures during winter.

This study presents an energy-optimized HVAC system philosophy aimed at reducing power consumption while ensuring reliable thermal control under severe environmental conditions. The design ambient parameters considered in the analysis include a summer dry-bulb temperature of $+30^\circ \text{C}$ with 80% relative humidity and a winter dry-bulb temperature of -20°C with 99% relative humidity. Offshore HVAC systems are inherently more complex than conventional building HVAC systems due to the unique operational, environmental, and reliability requirements of offshore substations. Unlike onshore facilities, offshore substations must achieve effective thermal

control while maintaining minimal air circulation in order to limit the ingress and accumulation of salt-laden air, which can accelerate corrosion and compromise equipment reliability.

The offshore HVAC configuration typically incorporates air handling units (AHUs) to satisfy fresh air requirements and to maintain room pressurization, thereby preventing infiltration of saline ambient air. Preconditioned air is supplied to dedicated rooms at the minimum airflow rates necessary to meet ventilation and pressurization criteria. Further cooling within individual rooms is achieved using fan coil units (FCUs), which are designed to handle the full sensible heat load, including equipment losses and solar heat gains. Both AHUs and FCUs operate on a chilled water system generated by water-cooled chillers.

The principal HVAC components for the offshore substation therefore include chillers, dry coolers, AHUs, FCUs, chilled water pumps, condenser water pumps, and heat exchangers. This study examines an offshore substation characterized by the following design parameters. The facility supports a total power generation capacity of 800 MW and is located approximately 10 km offshore at a sea level elevation of 30 m. The substation comprises four decks with a total conditioned floor area of $1,400 \text{ m}^2$. The total cooling load arising from equipment losses and

solar heat gain within the rooms is estimated at 400 kW. The indoor design conditions are maintained at 25 °C during summer operation and 20 °C during winter operation. The ambient design conditions considered include a summer dry-bulb temperature of +30 °C with 80% relative humidity and a winter dry-bulb temperature of -20 °C with 100% relative humidity. A comparative assessment is conducted between a conventional

cooling approach and an optimized offshore HVAC strategy. Under standard industry practice, offshore substations are cooled and pressurized using dedicated AHUs that provide precooled fresh air to the conditioned spaces. This baseline configuration serves as a reference case for evaluating energy performance improvements achieved through alternative system design and control philosophies.

2. CALCULATION METHODOLOGY FOR THE CONVENTIONAL COOLING SYSTEM

The following calculations are performed for the conventional (standard practice) offshore HVAC cooling system. The methodology is based on fundamental psychrometric relationships, heat transfer theory, and established HVAC

design principles. These equations are used to evaluate air-side and water-side performance, cooling coil behavior, and system effectiveness under the defined offshore ambient conditions.

2.1 Atmospheric Pressure at Installation Altitude

The atmospheric pressure corresponding to the offshore installation altitude is calculated using the standard barometric equation:

2.2 Saturation Vapour Pressure

The saturation vapour pressure of moist air is evaluated based on the dry-bulb temperature.

(a) For liquid water, $T \geq 0$ °C:

$$P_{ws} = 0.61094 \exp\left(\frac{17.625T}{T + 243.04}\right)$$

(b) For ice, $T < 0$ °C:

$$P_{ws} = 0.61094 \exp\left(\frac{22.587T}{T + 273.86}\right)$$

where

P_{ws} is the saturation vapour pressure (kPa), and
 T is the dry-bulb temperature (°C).

2.3 Humidity Ratio

The humidity ratio of moist air is determined as:

$$w = \frac{0.62198 P_w}{P - P_w}$$

where

w is the humidity ratio (kg water/kg dry air),

$P_w = RH \times P_{ws}$ is the partial pressure of water vapour (kPa),

and

P is the total atmospheric pressure (kPa).

2.4 Moist Air Enthalpy

The specific enthalpy of moist air is calculated using:

$$h = 1.006T + w(2501 + 1.86T)$$

where

h is the enthalpy of moist air (kJ/kg dry air), and

T is the dry-bulb temperature (°C).

2.5 Specific Heat of Moist Air

The specific heat capacity of moist air is given by:

$$P = 101.325(1 - 2.25577 \times 10^{-5}z)^{5.2559}$$

where

P is the barometric pressure (kPa), and

z is the altitude above sea level (m).

$$c_{p,ma} = 1.006 + 1.86w$$

(kJ/kg dry air·K)

2.6 Specific Volume of Moist Air

The specific volume of moist air is expressed as:

$$v = \frac{R_{da}(T + 273.15)(1 + 1.6078w)}{P}$$

where

$R_{da} = 0.287042$ kJ/kg·K.

2.7 Cooling Coil Bypass Factor

The cooling coil bypass factor (BF) is defined as:

$$BF = \frac{T_{out} - T_{ADP}}{T_{in} - T_{ADP}}$$

where

T_{ADP} is the apparatus dew point temperature (°C).

2.8 Cooling Coil Heat Transfer (Air Side)

The total and sensible cooling capacities are calculated as:

$$Q_{tot} = \dot{m}_{da}(h_{in} - h_{out})$$

$$Q_{sens} = \dot{m}_{da} c_{p,da}(T_{in} - T_{out})$$

The sensible heat ratio (SHR) is then determined by:

$$SHR = \frac{Q_{sens}}{Q_{tot}}$$

2.9 Chilled Water Flow Rate

The chilled water mass flow rate required for the cooling coil is calculated as:

$$\dot{m}_w = \frac{Q_{tot}}{c_{p,w} \Delta T_w}$$

where

$$c_{p,w} = 4.186 \text{ kJ/kg} \cdot \text{K}$$

2.10 Log Mean Temperature Difference (LMTD)

For a counterflow heat exchanger, the LMTD is defined as:

$$\begin{aligned} \Delta T_1 &= T_{h,in} - T_{c,out} \\ \Delta T_2 &= T_{h,out} - T_{c,in} \\ \text{LMTD} &= \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \end{aligned}$$

2.11 Overall Heat Transfer Coefficient (UA)

The overall heat transfer coefficient is obtained from:

$$UA = \frac{Q}{F \cdot \text{LMTD}}$$

where

F is the correction factor (typically 0.9–1.0 for crossflow wet coils).

2.12 Capacity Rates

The heat capacity rates of air and water are calculated as:

$$\begin{aligned} C_{air} &= \dot{m}_{da} c_{p,ma} \\ C_{water} &= \dot{m}_w c_{p,w} \\ C_{min} &= \min(C_{air}, C_{water}), C_{max} = \max(C_{air}, C_{water}) \\ C_r &= \frac{C_{min}}{C_{max}} \end{aligned}$$

2.13 Number of Transfer Units (NTU)

$$\text{NTU} = \frac{UA}{C_{min}}$$

3. CALCULATION PROCEDURE AND RESULTS: CONVENTIONAL COOLING SYSTEM

Given Design Data

The calculations are performed for the summer and winter design conditions of the offshore substation HVAC system using the standard cooling configuration. The key input parameters are summarized below:

- **Summer inlet air:**
Dry-bulb temperature $T_{db} = 30.0^\circ\text{C}$, relative humidity $\text{RH} = 80\%$
- **Summer outlet air (assumed):**
 $T_{out} = 12.5^\circ\text{C}$, $\text{RH} \approx 95\%$
Apparatus dew point (ADP) = 11.5°C , bypass factor (BF) = 0.10
- **Chilled water (summer):**
 $7.0 / 12.0^\circ\text{C}$ (supply / return)

2.14 Heat Exchanger Effectiveness (ϵ -NTU Method)

For a crossflow heat exchanger with both unmixed fluids, the effectiveness is expressed as:

$$\epsilon = 1 - \exp \left[-\frac{1}{C_r} (1 - \exp(-C_r \cdot \text{NTU})) \right]$$

2.15 Heat Transfer Rate

The heat transfer rate using the ϵ -NTU method is calculated as:

$$Q = \epsilon C_{min} (T_{h,in} - T_{c,in})$$

2.16 Outlet Temperatures

The outlet temperatures of the cold and hot fluids are given by:

Cold fluid (air):

$$T_{c,out} = T_{c,in} + \frac{Q}{C_c}$$

Hot fluid (water):

$$T_{h,out} = T_{h,in} - \frac{Q}{C_h}$$

2.17 Winter Relative Humidity Calculation

Under winter operation, where no condensation occurs, the humidity ratio remains constant. The outlet relative humidity is therefore calculated as:

$$\text{RH}_{out} = \frac{P_w}{P_{ws}(T_{out})}$$

where

$$P_w = \frac{wP}{0.62198 + w}$$

- **Winter inlet air:**

$$T_{db} = -20.0^\circ\text{C}, \text{RH} = 100\%$$

- **Winter water inlet:**

7.0°C (identical UA and water flow rate as summer)

- **Airflow rate:**

$$12,000 \text{ m}^3/\text{h} (3.333 \text{ m}^3/\text{s})$$

- **Altitude:**

30 m above sea level

Symbols and Notation

P : atmospheric pressure (kPa)

z : altitude (m)

P_{ws} : saturation vapour pressure (kPa)
RH: relative humidity (–)
 w : humidity ratio (kg/kg_{da})
 h : specific enthalpy (kJ/kg_{da})
 $c_{p,ma}$: specific heat of moist air (kJ/kg_{da}·K)
 v : specific volume (m³/kg_{da})
 $\dot{m}_{da}$: dry-air mass flow rate (kg/s)

$\dot{m}_w$: water mass flow rate (kg/s)
 Q : heat transfer rate (kW)
 UA : overall heat transfer conductance (kW/K)
 F : LMTD correction factor (–)
 C : capacity rate (kW/K)
NTU: number of transfer units (–)
 ε : heat exchanger effectiveness (–)

4. SUMMER OPERATING CONDITION (COOLING MODE)

Atmospheric and Psychrometric Properties

The atmospheric pressure at the offshore installation altitude is calculated as:

$$P = 100.97 \text{ kPa}$$

At an inlet air temperature of 30 °C, the saturation vapour pressure is:

$$P_{ws} = 4.2367 \text{ kPa}$$

The partial vapour pressure is therefore:

$$P_w = 0.80 \cdot 4.2367 = 3.3893 \text{ kPa}$$

The corresponding inlet humidity ratio is:

$$w_{in} = 0.02160 \text{ kg/kg}_{da}$$

The inlet air enthalpy is calculated as:

$$h_{in} = 85.42 \text{ kJ/kg}_{da}$$

The specific volume of inlet air is:

$$v_{in} = \frac{0.287042(30 + 273.15)(1 + 1.6078 \cdot 0.02160)}{100.97} = 0.892 \text{ m}^3/\text{kg}_{da}$$

Air and Water Mass Flow Rates

The dry-air mass flow rate corresponding to the specified airflow is:

$$\dot{m}_{da} = \frac{3.333}{0.892} = 3.738 \text{ kg/s}$$

At the assumed outlet air temperature of 12.5 °C:

$$\begin{aligned} P_{ws} &= 1.4468 \text{ kPa}, P_w = 0.95 \cdot P_{ws} = 1.3745 \text{ kPa} \\ w_{out} &= 0.00858 \text{ kg/kg}_{da} \\ h_{out} &= 34.24 \text{ kJ/kg}_{da} \end{aligned}$$

Cooling Coil Load

The total cooling capacity of the AHU cooling coil is:

$$Q_{tot} = \dot{m}_{da}(h_{in} - h_{out}) = 3.738(85.42 - 34.24) = 191.3 \text{ kW}$$

The sensible cooling load is calculated as:

$$Q_{sens} = \dot{m}_{da} \cdot 1.006(30 - 12.5) = 65.8 \text{ kW}$$

The resulting sensible heat ratio (SHR) is:

$$\text{SHR} = 0.344$$

The required chilled water mass flow rate is:

$$\dot{m}_w = 9.139 \text{ kg/s} \approx 32.9 \text{ m}^3/\text{h}$$

Heat Transfer Parameters

The log mean temperature difference (LMTD) is calculated as:

$$\begin{aligned} \Delta T_1 &= 30 - 12 = 18.0 \text{ K} \\ \Delta T_2 &= 12.5 - 7 = 5.5 \text{ K} \\ \text{LMTD} &= 10.54 \text{ K} \end{aligned}$$

The overall heat transfer conductance is therefore:

$$UA = \frac{191.3}{0.95 \cdot 10.54} = 19.10 \text{ kW/K}$$

The outlet temperature estimated using the bypass factor relationship is:

$$T_{out} = T_{ADP} + \text{BF}(T_{in} - T_{ADP}) = 13.35 \text{ °C}$$

which is in close agreement with the assumed outlet temperature of 12.5 °C.

Total Cooling Demand

The total cooling demand for the offshore substation under the conventional cooling strategy is:

$$Q_{total} = Q_{AHU} + Q_{FCUs} = 591.3 \text{ kW}$$

5. FREE COOLING STRATEGY:

Feasibility and System Configuration for a Water-Cooled Chiller with Dry Cooler and Plate Heat Exchanger:

Free cooling is employed in offshore substations to significantly reduce HVAC energy consumption by utilizing low ambient air temperatures as an alternative cooling medium during extreme cold weather conditions. In this mode of operation, the chiller compressors are fully disabled, thereby eliminating compressor power consumption while maintaining the required chilled water supply for space cooling.

During free cooling operation, the entire chilled water return flow is diverted through a dry cooler, where heat is rejected directly to the ambient air. The activation of free cooling is governed by an outdoor air thermostat that continuously

monitors ambient conditions and enables the mode when the outdoor temperature falls below a predefined threshold. The cooling demand of the fan coil units (FCUs) is met by the dry cooler through an intermediate plate heat exchanger, which hydraulically separates the building chilled water loop from the dry cooler loop while ensuring efficient heat transfer.

The plate heat exchanger facilitates thermal energy exchange between the FCU chilled water circuit and the dry cooler circuit without direct fluid mixing, thereby preserving water quality and system reliability. This configuration allows the FCUs to maintain the required indoor thermal conditions using ambient-driven heat rejection, with auxiliary components such as pumps and control valves remaining in operation. The free cooling chilled water circuit adopted in this study is illustrated schematically in below Fig.1

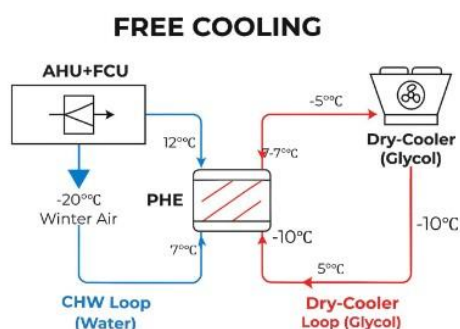


Fig.1

Free Cooling Feasibility and Sizing Analysis

The total cooling demand of the offshore substation, including air handling units (AHUs) and fan coil units (FCUs), is estimated as:

$$Q_{total} = 591.3 \text{ kW} (Q_{AHU} = 191.3 \text{ kW}, Q_{FCU} = 400.0 \text{ kW})$$

A dry cooler with a rated nameplate capacity of 780 kW is provided. Under winter design conditions of -20°C ambient temperature and 100% relative humidity, the dry cooler capacity exceeds the total system cooling requirement. Consequently, full free cooling operation is achievable during winter using a plate heat exchanger (PHE) to thermally couple the dry-cooler loop and the chilled water (CHW) loop.

Inputs and Assumptions

The free cooling analysis is based on the following inputs and design assumptions:

Cooling loads:

$$Q_{AHU} = 191.3 \text{ kW},$$

$$Q_{FCU} = 400.0 \text{ kW},$$

$$Q_{total} = 591.3 \text{ kW}$$

Nominal chilled water temperatures:

$7/12^\circ\text{C}$ ($\Delta T = 5 \text{ K}$), consistent with coil selections

Dry cooler nameplate capacity:

780 kW

Dry-cooler loop temperature difference:

$\Delta T = 5 \text{ K}$ (typical control band)

Heat exchanger configuration:

Counterflow plate heat exchanger

Freeze protection:

Glycol-based fluid on the dry-cooler loop during winter operation

Governing Equations

The free cooling performance is evaluated using standard heat transfer relationships:

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(A) Heat balance

$$Q = \dot{m} c_p \Delta T$$

(B) Capacity rate

$$C = \dot{m} c_p (\text{kW/K})$$

(C) Counterflow heat exchanger LMTD

$$\begin{aligned} \Delta T_1 &= T_{h,in} - T_{c,out} \\ \Delta T_2 &= T_{h,out} - T_{c,in} \\ \text{LMTD} &= \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \end{aligned}$$

(D) Heat exchanger duty

$$Q = UA \cdot \text{LMTD} \Rightarrow UA = \frac{Q}{\text{LMTD}}$$

Circuit Flow Rate Determination

Assuming a temperature difference of 5 K on both circuits, the required mass flow rates are calculated as follows:

Chilled Water Loop

$$\dot{m}_{chw} = \frac{Q_{total}}{c_p \Delta T} = \frac{591.3}{4.186 \cdot 5} = 28.25 \text{ kg/s}$$

$$C_{chw} = \dot{m}_{chw} c_p = 118.3 \text{ kW/K}$$

$$LMTD = 17.0 \text{ K}$$

Dry-Cooler Loop

$$\dot{m}_{dc} = \frac{Q_{total}}{c_p \Delta T} = 28.25 \text{ kg/s}$$

$$C_{dc} = 118.3 \text{ kW/K}$$

The equal capacity rates on both sides simplify heat exchanger sizing and control.

Plate Heat Exchanger Sizing Requirement

To deliver chilled water at 7/12 °C under winter free cooling conditions, a conservative cold-side temperature profile is assumed.

Case 1: Cold Glycol Circuit

$$T_{c,in} = -10 \text{ °C}, T_{c,out} = -5 \text{ °C}$$

The corresponding terminal temperature differences are:

$$\Delta T_1 = 12 - (-5) = 17.0 \text{ K}$$

$$\Delta T_2 = 7 - (-10) = 17.0 \text{ K}$$

6. FREE HEATING CONCEPT USING CHILLED WATER DURING WINTER OPERATION

Under conventional offshore HVAC design philosophy, chilled water circulation through the AHU cooling coil is discontinued during winter and extreme cold weather conditions, as precooling of fresh air is not required. Instead,

The required overall heat transfer conductance of the plate heat exchanger is therefore:

$$UA = \frac{591.3}{17.0} = 34.8 \text{ kW/K}$$

Dry Cooler Capacity Verification

The available dry cooler capacity under winter ambient conditions is:

$$Q_{available} = 780 \text{ kW}$$

which exceeds the required system cooling load:

$$Q_{available} > Q_{total} = 591.3 \text{ kW}$$

The resulting free cooling fraction is therefore:

$$\text{Free cooling fraction} = 100\%$$

No thermal shortfall is observed, confirming that full free cooling operation is achievable at -20 °C ambient temperature.

electric heaters are employed to raise the fresh air temperature to the required supply condition. While operationally simple, this approach results in substantial electrical power consumption during prolonged cold periods.

This study introduces and validates a novel operating concept, herein referred to as **free heating**, in which chilled water is intentionally circulated through the AHU cooling coil during winter operation. Despite its nominally low temperature, chilled water at 7 °C is significantly warmer than the extreme ambient air conditions typically encountered at offshore sites. Heat transfer from the chilled water to the incoming fresh air therefore results in a measurable increase in supply air temperature, thereby reducing the downstream electrical heater duty.

The proposed concept is analytically evaluated and verified for extreme winter conditions, demonstrating that the use of chilled water through the AHU cooling coil can effectively preheat fresh air and achieve meaningful reductions in electrical heater power consumption.

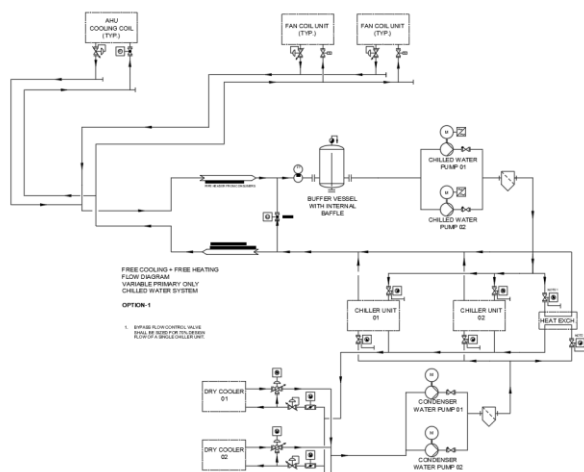


Fig.2

7. WINTER OPERATING CONDITION: FREE HEATING WITH 7 °C CHILLED WATER

Psychrometric Properties and Airflow

At an ambient dry-bulb temperature of -20 °C, saturation vapour pressure over ice is calculated as:

$$P_{ws} = 0.1031 \text{ kPa}$$

Given a relative humidity of 100%, the partial vapour pressure is:

$$P_w = 1.00 \cdot P_{ws} = 0.1031 \text{ kPa}$$

The corresponding inlet humidity ratio is:

$$w_{in} = 0.000636 \text{ kg/kg}_{da}$$

The specific volume of the inlet air is:

$$v_{in} = 0.720 \text{ m}^3/\text{kg}_{da}$$

The resulting dry-air mass flow rate is:

$$\dot{m}_{da} = \frac{3.333}{0.720} = 4.627 \text{ kg/s}$$

Heat Capacity Rates

The heat capacity rates of air and water are determined as:

$$C_{air} = \dot{m}_{da} \cdot c_{p,ma} = 4.627 \cdot 1.007 = 4.66 \text{ kW/K}$$

$$C_{water} = \dot{m}_w \cdot c_{p,w} = 9.139 \cdot 4.186 = 38.26 \text{ kW/K}$$

Thus,

$$C_{min} = 4.66 \text{ kW/K}, C_{max} = 38.26 \text{ kW/K}, C_r = 0.122$$

ϵ -NTU Analysis and Heat Transfer

Using the previously estimated cooling coil conductance

$UA = 19.10 \text{ kW/K}$, the number of transfer units is calculated as:

$$NTU = \frac{UA}{C_{min}} = \frac{19.10}{4.66} = 4.10$$

For a crossflow heat exchanger with both unmixed fluids, the effectiveness is:

$$\epsilon = 0.960$$

The maximum temperature difference between the fluids is:

$$\Delta T_{max} = T_{h,in} - T_{c,in} = 7 - (-20) = 27.0 \text{ K}$$

The heat transfer rate achieved through free heating is therefore:

$$Q = \epsilon \cdot C_{min} \cdot \Delta T_{max} = 0.960 \cdot 4.66 \cdot 27.0 = 120.8 \text{ kW}$$

Outlet Conditions

8. FREE HEATING CONCEPT USING RETURN CHILLED WATER DURING WINTER OPERATION

This study introduces and validates a novel operating concept, herein referred to as **free heating**, in which return chilled water is intentionally circulated through the AHU cooling coil during winter operation. chilled water at 12 °C is significantly warmer than the extreme ambient air conditions

The outlet air and water temperatures are calculated as:

$$T_{air,out} = T_{air,in} + \frac{Q}{C_{air}} = 5.93 \text{ °C}$$

$$T_{water,out} = T_{water,in} - \frac{Q}{C_{water}} = 3.84 \text{ °C}$$

The outlet relative humidity of the air stream is:

$$RH_{out} = \frac{P_w}{P_{ws}(T_{out})} = \frac{0.1031}{0.9295} \approx 11.1\%$$

As the humidity ratio remains constant and no condensation occurs, the free heating process is entirely sensible.

LMTD Cross-Verification

To validate the ϵ -NTU results, an LMTD-based cross-check is performed using:

$$T_{h,in} = 7 \text{ °C}, T_{h,out} = 3.84 \text{ °C}, T_{c,in} = -20 \text{ °C}, T_{c,out} = 5.93 \text{ °C}$$

The resulting LMTD is:

$$LMTD = 7.34 \text{ K}$$

The corresponding heat transfer rate is:

$$Q = UA \cdot F \cdot LMTD = 19.10 \cdot 0.95 \cdot 7.34 = 133.2 \text{ kW}$$

The close agreement between the ϵ -NTU and LMTD results confirms the robustness of the analysis.

typically encountered at offshore sites. Heat transfer from the chilled water to the incoming fresh air therefore results in a measurable increase in supply air temperature, thereby reducing the downstream electrical heater duty more efficiently comparison to chilled water supply passing.

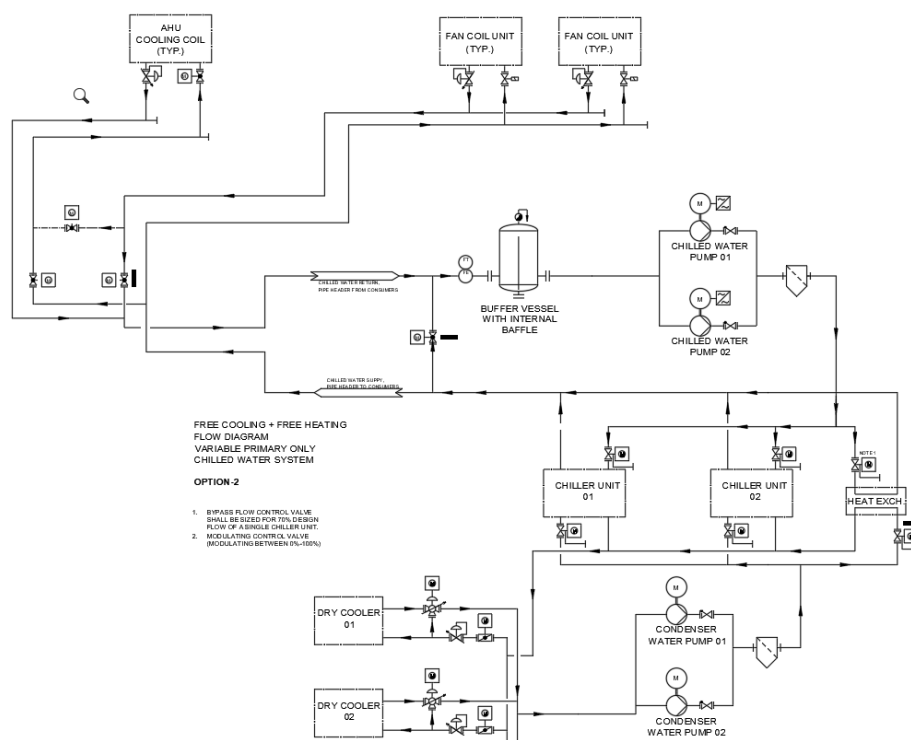


Fig.3

9. RESULTS AND DISCUSSION

A comparative evaluation of three operating scenarios was conducted under extreme winter ambient conditions to assess the effectiveness of the proposed **free heating** concept for offshore substations. Given the stringent constraints on auxiliary power consumption in offshore installations, the reduction of electrical heating demand represents a critical design objective. The results clearly demonstrate that free heating, when integrated with the free cooling configuration, offers a highly effective strategy for minimizing electrical power consumption during extreme cold weather.

Case 1: Standard Heating Configuration (Baseline Practice)

Under conventional operating philosophy, chilled water circulation through the AHU cooling coil is discontinued during winter operation. Fresh air heating is achieved solely through electric heating coils. For an ambient temperature of -20°C and a target supply air temperature of $+15^{\circ}\text{C}$, and assuming purely sensible heating without humidification, the required heating capacity is calculated as:

$$Q_{\text{heat}} = \dot{m} \cdot c_p \cdot (T_2 - T_1)$$

$$Q_{\text{heat}} = 136.2 \text{ kW}$$

Thus, the electrical heating capacity required under standard practice is approximately **136 kW**. This value represents a significant auxiliary power demand, particularly when sustained over extended winter operating periods.

Case 2: Free Heating Using Chilled Water Supply (7°C) with Free Cooling

In the second scenario, free heating is achieved by circulating chilled water at 7°C through the AHU cooling coil during winter operation. As demonstrated in the analytical section, heat transfer from the chilled water to the incoming fresh air raises the air temperature to 5.93°C without any electrical heating input.

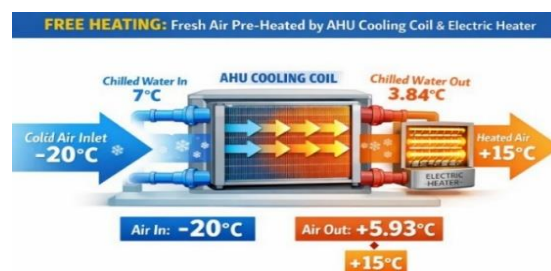


Fig.4

The resulting electrical heating requirement is therefore reduced to:

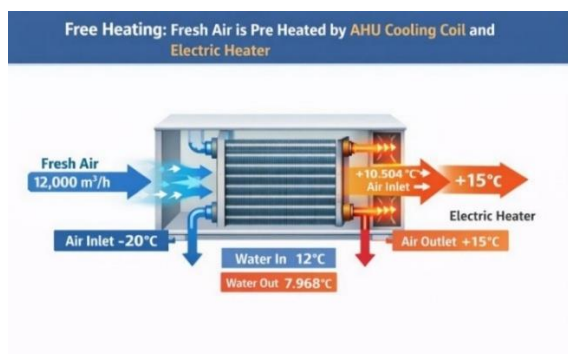
- -20°C to $+5.93^{\circ}\text{C}$:
0 kW (achieved entirely through free heating using chilled water)
- $+5.93^{\circ}\text{C}$ to $+15^{\circ}\text{C}$:
38 kW (electric heater)

The total electrical heating power required in this case is **38 kW**, representing a reduction of approximately **72%** relative to the standard heating configuration.

Case 3: Free Heating Using Chilled Water Return (12°C) with Free Cooling

In the third scenario, free heating is further enhanced by utilizing the chilled water return line at a higher

temperature of 12 °C. Based on the corresponding free heating calculations, the incoming fresh air temperature is increased to approximately 10.5 °C before electric heating.



10. DISCUSSION AND PRACTICAL IMPLICATIONS

The analysis demonstrates that chilled water, even at relatively low temperatures, constitutes a valuable thermal resource under extreme cold ambient conditions. By exploiting the large temperature differential between ambient air and chilled water, the AHU cooling coil effectively operates as a heat exchanger, substantially reducing reliance on electrical heaters.

11. CONCLUSION

Based on the comprehensive analytical evaluation of multiple operating scenarios, it is concluded that the implementation of free heating has a substantial and measurable impact on reducing auxiliary power consumption in offshore substation HVAC systems under extreme winter conditions.

The results demonstrate that, under conventional operating philosophy without free heating, the electrical power required for fresh air heating at an ambient temperature of -20 °C is approximately 136 kW. When free heating is introduced by circulating chilled water at 7 °C through the AHU cooling coil, the electrical heating demand is significantly reduced to 38 kW, corresponding to a power reduction of nearly 72%.

Furthermore, enhanced performance is achieved by utilizing the chilled water return from FCUs, with an inlet temperature of approximately 12 °C, as the AHU coil supply. Assuming constant FCU sensible load throughout the year, this configuration further lowers the electrical heating requirement to 19 kW,

Fig.5

The resulting electrical heating requirement is:

- **-20 °C to +10.5 °C:**
0 kW (free heating using chilled water return)
- **+10.5 °C to +15 °C:**
19 kW (electric heater)

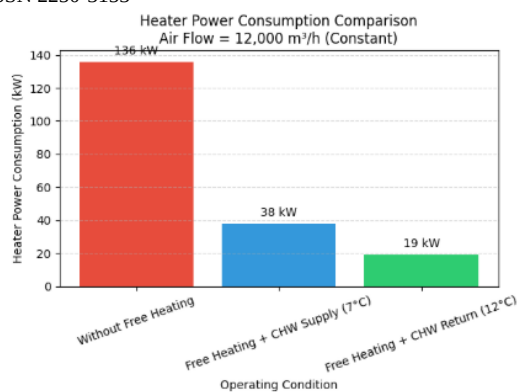
The total electrical heating demand is therefore reduced to **19 kW**, corresponding to an **86% reduction** compared to the baseline case.

For offshore substations—where power availability, reliability, and operational cost are critical constraints, the proposed free heating strategy offers a robust and energy-efficient alternative to conventional electric heating. When combined with free cooling operation, the approach enables year-round optimization of HVAC energy consumption without additional major equipment, relying instead on intelligent use of existing system components.

representing an overall reduction of approximately 86% relative to the baseline condition.

These findings confirm that chilled water, even during winter operation, represents an effective and underutilized thermal resource that can be leveraged for air preheating. The proposed free heating concept enables substantial recovery of low-grade thermal energy without the need for additional major equipment, relying instead on optimized utilization of existing chilled water infrastructure.

The comparative analysis and graphical representation clearly illustrate the progressive reduction in electrical heating demand across the evaluated scenarios, highlighting free heating as a highly efficient and practical solution for offshore substations. Given the critical importance of power efficiency, reliability, and operational cost in offshore environments, the adoption of free heating in conjunction with free cooling provides a robust strategy for year-round HVAC energy optimization.



- Without Free Heating → 136 kW (very high heater energy)
- Free Heating + CHW @ 7°C → 38 kW (≈72% saving)
- Free Heating + CHW @ 12°C → 19 kW (≈86% saving)

REFERENCES

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