Master's thesis

NTNU
Norwegian University of Science and Technology
Faculty of Engineering
Department of Energy and Process Engineering

Adam Richard Hadley

Energy Use and Energy Efficiency Measures for a Passenger Ferry

Master's thesis in Sustainable Energy Use in Buildings
Supervisor: Natasa Nord
Co-supervisor: August Brækken
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Project Description

Cruise or passenger ships are an energy-intensive ship sector. Traditionally, ship energy needs are covered by burning fossil fuels (diesel), which leads to a high carbon footprint per passenger as well as emissions of substances that are harmful to the environment and to human health. Developments towards "greener" passenger ships have largely focused on alternative fuels and propulsion systems. This is the case for Color Line's cruise ferry Color Hybrid, which has a more environmentally friendly diesel-electric hybrid propulsion system. The system enables up to one hour of battery powered propulsion between each charging in port. However, to achieve zero emission passenger ships, it is not enough to only consider the propulsion system. The energy use for the so-called "hotel system" on board cruise ships (heating, air conditioning, cooling, appliances, etc.) accounts for an average of 40% of the ship's total energy use. It is therefore important to reduce the energy use of the hotel systems. This project is part of the innovation project LowPass (Future Low-Emission Passenger Ships), which is a collaboration between Fosen Design & Solutions, Color Line Marine, SINTEF Energy Research, and NTNU. The overall ambition of the LowPass project is to reduce the energy use of hotel systems on board passenger ships and analyze the possibility of an energy self-sufficient operation.

The aim of the master's thesis is to analyze energy use and energy efficiency possibilities for a passenger ferry using a building simulation tool. The focus should be on the hotel facilities, with some consideration of the propulsion system and other energy systems. Based on a literature study of heating, ventilation, and cooling needs on passenger ships, and the ship's specific design and operational data, the student will develop a model in IDA ICE, which will be used to evaluate different usage patterns, energy efficiency measures, and possibilities for peak load reduction.

The following tasks are to be considered:

1. Literature study on energy and fuel use in passenger ships and cruise ferries considering total annual, monthly, and daily energy use. Performance indicators for passenger ship operation should be considered. The study should specifically include heating, cooling, and ventilation systems on ships. Performance indicators for passenger ship operation should be considered.
2. Analyze energy/fuel use data for the passenger ferry available from the project partners. A literature study on energy and fuel use in passenger ships/ferries should be used as a supplement to the available data.
3. Based on the literature study and documentation available from the project partners, organize necessary data relevant for hotel systems on the analyzed cruise ferry. These should include information about heating, ventilation, and cooling systems. Further, information about the operation of the ship's HVAC systems should be organized, as well as information about internal heat gains.
4. Collect and develop relevant weather data. If necessary for the purpose of the project, develop own weather files that may be used as input.
5. Develop energy efficiency and peak load reduction scenarios that are relevant for passenger ferries. The scenarios should be discussed with the project partners.
6. Using IDA ICE, develop a model for the hotel system on the passenger ferry Color Hybrid based on points 1 and 2.
7. Calibrate the IDA ICE model based on energy use data available from the project partners.
8. Perform energy efficiency analyses for some of the suggested scenarios.
9. Prepare the master's thesis.

Preface

This thesis accounts for the final thirty study points necessary to fulfill the requirements of a two-year Master of Science degree in Sustainable Energy, with the main profile *Energy Use in Buildings*, from the Department of Energy and Process Engineering at the Norwegian University of Science and Technology in Trondheim, Norway. The primary goal for this work is to develop and verify an energy model of the Color Hybrid passenger ship's hotel system, and then assess potential efficiency improvements. The project started in 2022 as a summer job at SINTEF Energy Research, continued in the fall as a student project thesis, and was finalized in the spring of 2023 as this master's thesis. This project is part of a larger project that is a collaboration between NTNU, SINTEF Energy Research, Fosen Design & Solutions, and Color Line Marine.

I would like to thank my supervisor Natasa Nord for her guidance and willingness to take the extra time to ensure our discussions ended at a place of mutual understanding. I am also immensely grateful for the help from my co-supervisor, August Brækken. August spent seemingly countless hours giving detailed and meaningful feedback through the entire project, all while patiently tolerating my goal of improving my Norwegian. Thank you to Cecilia Gabrielii at SINTEF Energy Research for the encouragement I needed to keep the project headed in the right direction. I would also like to thank Mads Thøstesen at Fosen Design & Solutions for his help with the design documents. Thank you to Jan Bjølgerud and his hard-working team at Color Line Marine for their help with my understanding of the ship's technical systems and data availability, and their welcoming hospitality during our visit.

A thousand thank-yous go to my wife and our two daughters for their unending support and encouragement – and for moving half-way across the world with me.

Adam Hadley

Trondheim, June 2023

Abstract

The project “Future Low-Emission Passenger Ships” (LowPass) intends to find innovative solutions to reduce the energy use of passenger ship hotel systems in support of the larger effort to ultimately eliminate emissions from passenger ships. The focus of this master’s thesis, which is a part of the LowPass project, was to build an energy model of the hotel system of an efficient passenger ship, the Color Hybrid, and use the model to evaluate potential energy efficiency measures. The results are described in terms of operational impacts on the Color Hybrid and other ships.

In the first step of the model development, ship design data were reviewed and used to set model parameters within the building simulation tool IDA ICE. Many of the ship’s physical characteristics, including the ship construction and ventilation systems, were specified with high levels of detail using the design data. In the second step, model validation was performed using historical energy consumption and operational data from the ship. Detailed review and analysis of the diesel and shore power consumption data were useful to develop a better understanding of the ship’s energy use and confirm some of its operational characteristics. Historical data for the hotel system’s electricity use were compared to the modeled electrical loads in multiple ways, including total average loads, monthly load profiles, and daily load profiles. Adjustments were made to the model to achieve better alignment with the data. The model’s implied design winter heating load for each zone was compared to the actual installed heater capacity, and this comparison was used to further adjust the model. While the unfortunate lack of historical thermal energy use data causes some limitations in the model’s reliability, the use of detailed design data in the development of the model and the available electricity consumption data help support the trustworthiness of the model.

Operational data from the ship’s heating, cooling, and air conditioning (HVAC) and chilled water systems, along with discussions with the ship engineering team, helped identify six efficiency opportunities, whose energy impacts were estimated using a combination of the data and the model. Three measures related to the hotel system’s ventilation system – nighttime reduction of the galley ventilation rate, galley ventilation air heat recovery, and reduction of the ventilation rate throughout the rest of the ship – show annual hotel savings of up to 15% of the heating energy, 4.5% of the cooling energy, and 8.3% of the electrical energy. A qualitative analysis of repairing a faulty inlet air temperature sensor indicates the importance of continually recommissioning the ship’s HVAC system. Selection of windows with a 0.20 higher solar heat gain coefficient results in a 1.1% reduction in the annual hotel heating load, but a 4.4% increase in the annual cooling energy load. Lastly, a conservative savings estimate for operating the Color Hybrid’s absorption chiller was made using the modeled hotel cooling demand along with duty cycles inferred from historical temperature data from the chilled water system.

The results of the model validation process indicate further work is necessary to ensure that the model represents the ship’s hotel electricity consumption. Thermal energy consumption data for both the hot water and cooling water systems, as well as control signal data or electricity consumption of the cooling systems, will be needed to fully validate the model. However, as shown by the efficiency measure analyses, with care the model can be useful in its current form to help guide efficiency measure decisions for existing or new ships.

Sammendrag

Målet med prosjektet "Future Low-Emission Passenger Ships" (LowPass) er å finne innovative løsninger som kan brukes for å redusere energiforbruk i hotelsystemene på passasjerskip. Dette bidrar til det ultimate målet: å kutte utslipp fra hele passasjerskipet. Målet med denne masteroppgaven er å utvikle en energimodell for hotelsystemet på et effektivt passasjerskip, Color Hybrid, og deretter bruker den for å vurdere mulige energisparingsløsninger. Det blir gitt en beskrivelse av hvordan disse tiltakene påvirker både Color Hybrids drift og driften på andre skip.

Det første steget i modellutviklingen var å gjennomgå designdataene for å bestemme modellparametre i bygningssimuleringsverktøyet IDA ICE. Mange av skipets fysiske egenskaper, som konstruksjon, materialer og ventilasjonssystemer, var beskrevet i designdataene med et høyt detaljnivå. I det andre steget ble modellen validert ved hjelp av både historiske energiforbruksdata og driftsdata. Detaljert analyse av dieselforbruket og landstrømforbruket ble utnyttet for å forsterke forståelsen av skipets energiforbruk og driftsegenskaper. Historiske data for hotelsystemets elektrisitetsforbruk ble sammenlignet med modellerte laster. Dette inkluderte totale gjennomsnittslaster, månedlige lasteprofiler og daglige lasteprofiler. Modellen ble justert for å samsvare bedre med dataene. Modellberegnede vinter-designlaster i hver sone ble sammenlignet med ytelsen til skipets faktiske oppvarmingsutstyr, og dette førte til flere modelljusteringer. Til tross for begrenset tillit til modellen på grunn av manglende tilgang til historiske termiske energidata, er tilliten til modellen støttet av designdata og data for elektrisitetsforbruk som ble brukt til å bygge og justere modellen.

Seks energisparingstiltak ble identifisert med hjelp av både tilgjengelige driftsdata fra skipets oppvarmings-, kjøle- og ventilasjonsanlegg, og fra diskusjoner med skipets ingeniører. Tiltakenes påvirkning på skipets energiforbruk ble beregnet både med modellen og de tilgjengelige dataene. Tre tiltak som påvirker hotellsystemets ventilasjonssystemer viser årlig energibesparelse opp mot 15 % av oppvarmingen, 4.5 % av kjølingen og 8.3 % av elektrisitetsforbruket. En kvalitativ analyse av å reparere en lufttemperatursensor indikerer at det er viktig å kontinuerlig vurdere styring av skipets ventilasjonssystem. Vinduer med en 0.20 økning i solfaktor fører til en 1.1 % reduksjon i den årlige oppvarmingslasten, men en 4.4 % økning i den årlige kjølelasten. Til slutt ble det estimert et konservativt anslag av energibesparelsen fra å ta i bruk Color Hybrids absorpsjonskjøler. Dette ble beregnet ut ifra den modellerte kjølelasten og arbeidssyklusene som ble beregnet fra historiske temperaturdata.

Resultatene til modellvalideringsprosessen tilsier at videre arbeid er nødvendig for å sikre at modellen representerer skipets elektriske energiforbruk. For å fullstendig validere modellen trengs det termiske energiforbruksdata for oppvarming og kjøling, samt enten styringssignaldata eller elektrisitetsforbruk for kjølesystemene. Ved varsom bruk av den nåværende modellen, kan den imidlertid benyttes for å hjelpe å ta beslutninger på eksisterende skip og nye skip.

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1 Introduction

This master's thesis is a part of a larger innovation project, called LowPass, which is a collaboration between NTNU, SINTEF Energy Research, Fosen Design & Solutions, and Color Line Marine. The goal of the LowPass project is to find innovative solutions to help achieve zero-emission passenger ship operation by reducing the energy use of hotel systems on passenger ships.

Understanding the impact of hotel system energy efficiency solutions will allow for improved decision-making during the design phases of both new ship constructions and major refurbishments of old ships. LowPass intends to investigate the impact of efficiency measures on multiple ships, thereby strengthening the understanding of not only the measures themselves, but also their impact on ships with differing characteristics and needs. This knowledge will help guide the industry in taking the best path to zero emission passenger ships.

A relatively new ship called Color Hybrid is the focus of this master's thesis. The Color Hybrid makes two roundtrip journeys per day between Sandefjord, Norway and Strömstad, Sweden, carrying up to 2,000 passengers and 500 cars. It uses only batteries to sail within the Sandefjordsfjord. In addition to its large electric battery system, the ship employs various energy efficiency upgrades to ensure the hotel system needs can be met during these operating conditions without the use of the ship's diesel engines. Inclusion of the Color Hybrid in the LowPass project allows for a better understanding of the real-world effectiveness, and perhaps challenges, of many of the efficiency measures expected to be used in zero emission passenger ships in the future.

The overall objective of this master's thesis is to develop a validated and calibrated energy model to represent the hotel system on the Color Hybrid, and to use the model to test the impacts and merits of various energy efficiency measures and give insight into their use on other ships.

1.1 Background

In partnership with Color Line, NTNU, and SINTEF Energy Research, Fosen Design & Solutions has set out to research and develop approaches to eliminate emissions caused by operation of hotel facilities on passenger ships. This project is titled "LowPass – Future Low-emission Passenger Ships". Part of this research includes developing detailed energy models of various existing passenger ships. The energy models for these ships will allow investigation of efficiency improvements to the hotel systems. The ultimate goal is to provide sound advice to ship builders on the best methods to meet zero emission goals for passenger ships.

Examples of efficiency solutions being considered:

- Reduce space heating and cooling needs by using:
  - Higher efficiency windows
  - Improved deck and bulkhead[1] insulation
  - Improved ventilation system heat recovery
- Reduce electrical energy requirements through use of higher efficiency:
  - Pumps
  - Fans
  - Lighting
- Improved use of available energy resources through:
  - Improved controls to optimize use of seawater cooling

[1] "Deck" is a term used on ships to represent the floor/ceiling construction that separates two levels of the ship. "Bulkhead" is a term used on ships to represent the walls, either exterior or interior walls.

- Thermal storage to optimize the use of waste heat from the combustion engines and large electric motors
- Photovoltaic panels to generate electricity from the sun
- The use of heat pumps to gather heat from cooler sources

In previous work, an IDA ICE energy model was developed to represent the hotel system of a standard-efficiency passenger ship, the Color Fantasy [1]. The dynamic energy simulation tool IDA ICE is commonly used to estimate energy consumption in buildings. The tool accounts for the dynamic nature of the weather, and the various technical systems' energy impacts on the ship and their interaction with one another. This leads to better confidence in the results when an energy efficiency measure is applied to the building, since the measures can have unpredictable impacts on the many interconnected technical systems. While a passenger ship is not stationary and built on soil like a building, these obstacles are expected to be both overcome with minor adjustments to the modeling technique and insignificant in their overall impact to the results.

The concept of developing a model using IDA ICE to represent the hotel portion of the Color Hybrid is the same as for the work to develop the model for Color Fantasy. However, there are important distinctions in both the construction and operation of the Color Fantasy versus the Color Hybrid. There are also important differences between the types of data available during development of their respective energy models. The ability to test the robustness of efficiency measures using the models of both ships will help to improve the understanding of the measures when applied across a variety of ships. It is also important to be able to meaningfully compare the results among the different ships.

The Color Fantasy is a large, standard-efficiency passenger ship. It sails a 20-hour route between Norway and Germany over night, so its passengers make use of cabins to sleep and bathe. It has additional energy-intensive amenities like a swimming pool. The Color Hybrid is smaller, has a more efficient design, does not carry overnight passengers, does not have a swimming pool, and sails a much shorter route. Because each passenger ship has unique characteristics, energy performance metrics were used to enable comparison across ships.

1.2 Structure of the Thesis

The background (Section 1.1) describes the history and goals of the LowPass project. The findings of a previous master's thesis in the LowPass project are also summarized.

A literature review (Section 0) provides insight into passenger ship energy use metrics that other authors have suggested being used to allow simpler and more robust comparison of ship energy use. Energy efficiency upgrades for passenger ships identified through the LowPass project are also summarized.

A short description of the Color Hybrid ship is given in Section 0. A more detailed description of many of the ship's components is included in the model development section (Section 0).

The methods used for data collection are described in Section 0. This includes data related to the ship construction, and the energy and operational data. Recommendations for future data collection are found in Section 0.

The methods used to develop the building energy model are explained in Section 0. Subsections give details of how the model was developed to match the characteristics of the ship's zones (5.1), the ship's construction (5.2), the internal loads (5.3), the energy systems (5.4), and the weather and climate the ship is impacted by (5.5). These sections provide the details of model input parameters that are expected to have a large impact on the modeled energy use estimates. In addition, necessary assumptions are described. This section is useful for a reader who wants an in-depth understanding of the underlying key parameters of the model. For example, it could be a tool for future users of the model who want to adjust the model to their own needs.

Section 0 describes a detailed analysis of the delivered energy data (6.1), the hotel electricity data (6.2), the chilled water temperature data (6.3), and air handler unit data (6.4). These data come from operation of the ship, so they are useful for model calibration and to help identify efficiency improvement opportunities. For each subsection, the processes of data verification, data cleaning, and normalization are described.

The simulation model results are given in Section 0. The purpose of this section is to familiarize the reader with the model's output capabilities, develop confidence in the outputs by comparing them to expected outputs, and present model adjustments made according to this review and the results of those adjustments. Basic model outputs are presented, including overall energy use, energy use of major end-uses by month, energy use results of the air handler units, and indoor air temperatures. The model validation subsections show a more detailed comparison of the model results to known data, including space heater sizing (7.1.1) and hotel electricity consumption (7.1.2). Section 7.2 presents the adjustments made to the model based on the validation process, as well as the final results.

The sub-sections in Section 0 each describe the six different efficiency measures, their savings estimation methodology, savings results, and general operational cost implications for both the Color Hybrid and other ships.

Section 0 presents the conclusions of the master's thesis. Recommendations for future work are discussed in Section 0. This includes a summary of the recommendations for thermal energy data collection, additional model validation work, and the analysis of additional efficiency measures.

1.3 Limitations of Study

A detailed literature review is not included as a part of this report because other members of the LowPass project performed the literature review necessary for the project. The results of that review are provided in a recent journal article [2]. Efforts for this masters thesis were directed toward achieving a better understanding of the ship design documents and operational data in lieu of a more detailed literature review.

There are proprietary elements to the Color Hybrid's design. Therefore, some details are intentionally omitted from this report. Specifically, while the author had access to the ship's design data, design drawings are not shown in this report and specific operating principles are left vague to protect the proprietary elements.

Calibration of the model was not satisfactorily completed because of a lack of thermal energy use data. Only electrical energy use data were available for analysis in this thesis. While it is an important component for understanding the overall energy use of the ship, electrical consumption data alone does not provide enough information to make trustworthy adjustment to the model. Thermal energy use in the hot and chilled water systems are expected to make up a substantial portion of the hotel system's energy use. Therefore, energy consumption of the thermal energy systems is also required for model calibration. The results in this report should be used with this limitation in mind.

2 Literature Review

The literature review for this project includes different concepts for development of a passenger ship energy use metric that can be used to quickly understand how a ship's efficiency compares to other ships. In addition, a review of the energy efficiency measures suggested by the project partners is given.

2.1 Passenger Ship Energy Use Metrics

In buildings, it is common to compare the efficiency of buildings by comparing annual energy use per floor area, that is, energy density. This metric, while admittedly crude, is useful for developing a quick understanding of the nature of each building's energy efficiency. Buildings serving more or less energy-intensive purposes, however, will naturally have respectively higher or lower energy density. Therefore, it is also common to distinguish these efficiency comparisons by building purpose. For example, restaurants tend to have much higher energy density than office buildings because of their cooking and refrigeration loads. Estimating energy efficiency of passenger ships in a comparable way comes with similar, but arguably more difficult, challenges. Passenger ships is a broad term that encompasses small ships carrying passengers short distances in mild climates and without additional amenities, to large ships carrying vehicles and people long distances in cold or hot climates and with additional amenities such as hotel rooms, restaurants, and even swimming pools. Given the uniqueness of each passenger ship's route and provided services, a one-size-fits-all energy efficiency metric for passenger ships has been elusive. However, a goal of this project is to develop such a metric. Here, previous work is reviewed to give insight into possible suggestions for a fair energy use metric for passenger ships.

In previous work, the comparison metrics used for the Color Fantasy ship included total annual hotel energy use per passenger, which was finally estimated at 19.2 MWh/passenger, and propulsion energy demand per available lower berth (ALB) and per kilometer travelled, which was finally estimated to be 0.156 kWh/ALB-km [1]. While many of the important energy-intensive services are not adjusted for with these metrics, they are simple metrics that can be easily estimated. Their usefulness in comparing across ships with different amenities depends on the ability of the user to understand the impact of the caveats.

In another study, a model was developed that was specific to cruise ships sailing near Norway. The model's independent variables include information on the ship's rated speed and engine ratings. Hotel services are also reviewed in the analysis [3]. While this approach is more complex than simply estimating energy consumption per area in the example of the building metric, this model could be used to make a judgment of whether a ship is considered "efficient", "average", or "inefficient". For example, if the actual energy consumption of the ship is less than 90% of the estimated energy consumption of the ship according to this model, the ship could be labeled "efficient". Interestingly, this model was developed using ships sailing in the cold climate conditions of Norway, so the climate variable has been accounted for, but the scope of this efficiency rating metric concept would be limited.

Another study demonstrates the use of an energy and exergy analysis to better understand the efficiency of passenger ships [4]. This approach could be useful in making fair comparisons across ships and can be explored further in future work.

2.2 Energy Efficiency Upgrades

A review of energy efficiency upgrades is useful to provide insight into how the model should be built to give the model flexibility for wide-ranging variety of possible efficiency measure analysis. The project team at SINTEF Energy Research reviewed efficient technologies for passenger ships and created three memos, which were finalized in December 2022. These memos are intended to represent the types of measures to be investigated in this work. They are draft internal documents, so they are not fully referenced here, other than providing the title and author below. A summary of the contents of each memo is included here, to provide background on the types of efficiency measures that inspired the model development.

The first document is titled “State-of-the-art review of passive technologies used in buildings” and was authored by August Brækken. This document describes insulation, windows, window-to-wall ratio, thermal mass, and passive lighting measures that are frequently used in building applications in Norway. These measures are recommended as being useful in passenger ship applications, given the similar heating loads of hotel systems on passenger ships.

The second document is titled “Heat pumping systems for maritime applications: A state of the art review” and was authored by Håkon Selvnes. Here, the use case for heat pumps on electric passenger ships is described, along with a summary of the operational characteristics of heat pumps and refrigeration systems. Applications of heat pumps and refrigeration systems using various natural refrigerants for passenger ships are described, including cooling and freezing, air conditioning and space heating, domestic hot water heating, and using heat pumps to replace sources of high-temperature steam. Natural refrigerants come with certain safety challenges, which are also discussed in relation to the safety requirements of passenger ships.

The third and final document, titled “State of the art within thermal energy storage”, was authored by Tarjei Heggset. This document reviews various thermal energy storage technologies, including sensible, latent, and thermochemical heat storage. A literature review of various marine applications of thermal energy storage is included.

3 Ship Description

The Color Hybrid is a relatively new ship that was put into service in 2019. It has a capacity of 2000 passengers and 500 cars. The ship sails from Sandefjord in Norway to Strömstad in Sweden, and back, two times per day. The one-way transit time is about 2.5 hours [5]. It has a unique hybrid diesel-electric drive, waste heat recovery and thermal storage systems that allow it to meet a goal of zero tailpipe emissions when driving and maneuvering in the Sandefjordsfjord. Table 1 shows general size [5] and capacity [6] characteristics of the ship, and a photograph of the exterior of the ship, taken in January 2023, is shown in Figure 1.

Table 1 – Ship Characteristics

Length	160.0	meters
Width	27.1	meters
Draft	6.0	meters
Carrying Capacity	27,164	gross tonnage
Passenger Capacity	2,000	passengers
Crew Capacity	100	crew members
Vehicle Capacity	500	vehicles
Battery Capacity	4.7	MWh
Thermal Storage	4.9	MWh

Figure 1 - Exterior of Color Hybrid

According to the design documents, the ship has ten decks. Decks 1 and 2 consist of engine rooms, storage, equipment, and a few workspaces. Decks 3, 4, and 5 are the car decks. Decks 6, 7, and 8 are the primary zones occupied by the passengers and crew and consist of various shops, restaurants, and sitting areas for the passengers, as well as cabins, offices, and galleys for the crew. Deck 9 has an assortment of equipment rooms, a galley, and other crew and passenger rooms, along with a large, uncovered deck for passengers. Deck 10 is mostly the roof of deck 9, and it is inaccessible to passengers.

The ship's sailing schedule [7] is shown in Table 2.

Table 2 - Daily Sailing Schedule

Departure-Arrival Port	Depart	Arrive
Sandefjord-Strømstad	10:00	12:30
Strømstad-Sandefjord	13:40	16:10
Sandefjord-Strømstad	17:00	19:30
Strømstad-Sandefjord	20:00	22:30

The ship is scheduled to make two round-trip crossings to Strömstad every day, which is estimated to be a 66 km one-way journey, based on a distance measurement using Google Maps (Figure 2) [8]. The ship harbors overnight in Sandefjord, where it has connection to shore power.

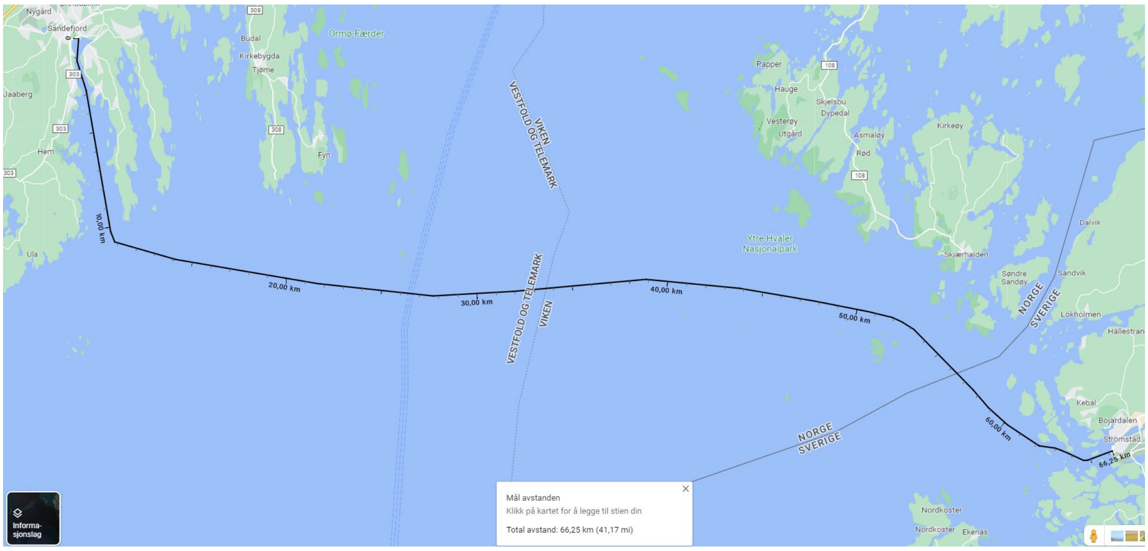

Figure 2 - Distance Estimate of the Sailing Route

4 Data Collection

Various data were made available from the project partners to support the development and validation of the model. This includes detailed design data, design electric and thermal loads, various energy consumption data, and operational data collected by the ship's energy management system.

4.1 Design Data

Detailed ship construction drawings and intended operation data for the Color Hybrid – hereafter referred to simply as "design data" – were made available and were used to assist in development of the model. This contrasts with the model of the Color Fantasy ship, where assumptions were necessary given the lack of available ship-specific data at the time of model development [1].

Access was provided to many of the ship's construction data, which are organized by SFI codes. Much of the time in this project has been to develop an understanding of the type of data available and retrieving it from the various files. For example, insulation characteristics and drawings, ship dimensions, and details of all the heating and cooling systems in each zone are known and can be accounted for in IDA ICE.

The sections below provide more details about where these files have been used in development of the model. Since the design documents are proprietary, they are not publicly available, and are therefore not referenced in Section 11 of this document. Instead, the source document names are given in italics in the sections below in order to provide a source for those with access.

While the availability of detailed design data is useful, it comes with three important caveats. The first is that judgment was required in weighing the tradeoff between using all the finely granular data available or using simplifications of the data through, for example, averages. Simplification helps with reducing both data entry time and model simulation time, but at the risk of reducing model precision. This judgement was guided based on a) an assumption of the overall impact of the decision on the model results and usability, b) what type of energy use data are expected to be available, and c) the types of efficiency upgrades that might be tested with the model. At this point in the project, however, the energy use data have not yet been provided nor have the potential efficiency upgrades been clearly identified. The model as described here may therefore need to be modified in the future, but attempts were made to minimize this need as much as possible.

The second caveat is that not all data were made immediately available at the start of the project. For example, details of the ventilation system were not available until after the initial creation of the IDA ICE zones. Given the expected importance of the ventilation system details on the energy use of the hotel system, edits were made to the model. In another example, the details of cooling systems were not known ahead of creation of the zones, and therefore some zones with active cooling systems are not included in IDA ICE. These zones are arguably not part of the hotel system, and they can likely be accounted for outside of the model. However, it may be that these should be added to the model later.

Finally, given the complexity of the ship's energy systems, there exists a lot of data. Because of the vast quantity of data, it is demanding to digest and therefore it is possible that the modeler has overlooked important details. While avoidance of this type of error is a top goal of the modeler, it is nevertheless important to highlight as a potential source of error. This can be mitigated somewhat during later stages of the project – primarily the calibration stage.

4.2 Design-day Electric and Thermal Loads

The ship designer has made available data they used to ensure adequate coverage of electric and thermal storage devices during heating and cooling design conditions. While these data are not actual operational data, and include many assumptions made by the designers, comparing them to the model outputs provides an opportunity to better understand the limitations and capabilities of the model. This type of comparison also helps ensure that the model is appropriately accounting for the main energy consuming components from the perspective of the ship designer.

In designing the size and control scheme for the electric batteries, the ship builder made estimates of the ship's electric loads on design winter and summer days for the various ship operating modes. Similarly, for the thermal storage tanks, the ship builder made estimates of the ship's heating loads on a design winter day for the various operating modes.

Analysis of the winter design electric loads is described here. These data are used to assist in model validation of the hotel electricity use (Section 7.1.2). First, data from the original file "*Electric Load Calculation for Winter Condition"* were imported into Excel. Then, each electric load for the 690 V and 230 V alternating current main switchboards were assigned to one of eight categories, based on the "Consumer" field in the original file. These loads represent all of the electrical energy consumption for the ship builder's definition of the hotel portion of the ship. Excluded loads include propulsion energy and battery charging and discharging, which are tied to the direct current switchboards.

Figure 3 shows the resulting electric loads for each of the categories and for each of the ship operating modes. The "ship systems" category represents electric loads not expected to impact the heated and cooled spaces of the ship. Examples include fin stabilizer pumps or steering gear. The "electric heating" category is mainly 150 kW of car deck heaters. This category does not include heaters used outside the hotel section of the ship, for example diesel fuel heaters, since they are not expected to impact the heated and cooled spaces of the ship. The "other hotel equipment" category represents all the leftover electric equipment that could not be included in another category, but that is expected to impact the heated and cooled spaces. Examples include the provision plant compressor motors and humidifiers.

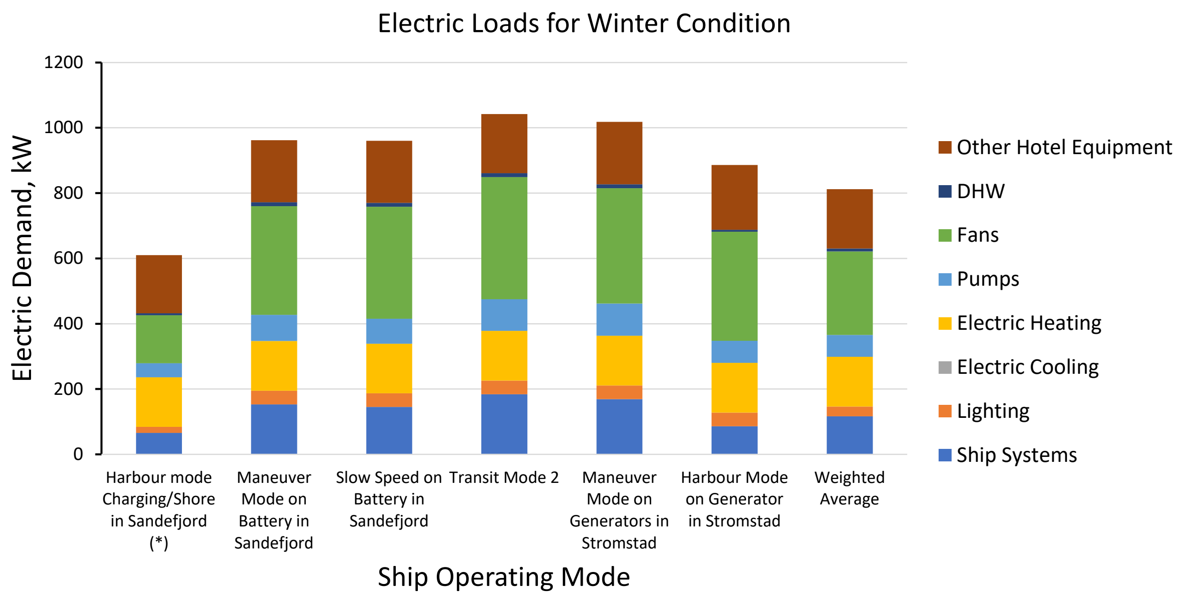

Figure 3 - Design-Day Electric Loads by Operating Mode

The first operating mode, “Harbour mode Charging/Shore in Sandefjord (*)”, is self-explanatory except that it has been adjusted to assume no use of the electric resistance heaters in the hot water systems. In the original document, it was assumed that 1290 kW is used to heat the hot water system. Instead, the thermal energy storage system will likely meet the heating load while docked, and the electric resistance heater use in the hot water system was therefore set to zero here. The “Weighted Average” operating mode represents the time-weighted average of each of the other six operating modes based on the ship’s schedule (Table 2) and an assumption that maneuvering takes 30 minutes every morning in Sandefjord and 15 minutes upon entry and departure in Strömstad and upon return to Sandefjord. The values for the weighted average are also shown in Table 3.

Table 3 - Design-Day Weighted Average Electric Loads

Electric Load Category	**Weighted Average Electric Load, kW**
Ship Systems	117
Lighting	30
Electric Cooling	0
Electric Heating	152
Pumps	103
Fans	256
DHW	9
Other Hotel Equipment	146
TOTAL	**812**

The assignment of the load categories in Figure 3 and Table 3, as described above, are subject to revision and further investigation during the model validation analysis. In total, there were 245 individual entries from the original document.

Similar analysis of the summer design electric loads can be made in future work (Section 0) to further assist in model validation. This can be especially insightful when it comes to the designers’ intentions regarding the expected loads of the electric chillers.

4.3 Weather Data

Historical average daily temperature for the Færder fyr weather station [9] was aligned with and used as the representative outdoor temperature for the historical operational and energy use data (Sections 0 and 7.1). The Færder fyr weather station was chosen due to its proximity to both Sandefjord and Strömstad, as shown in Figure 4.

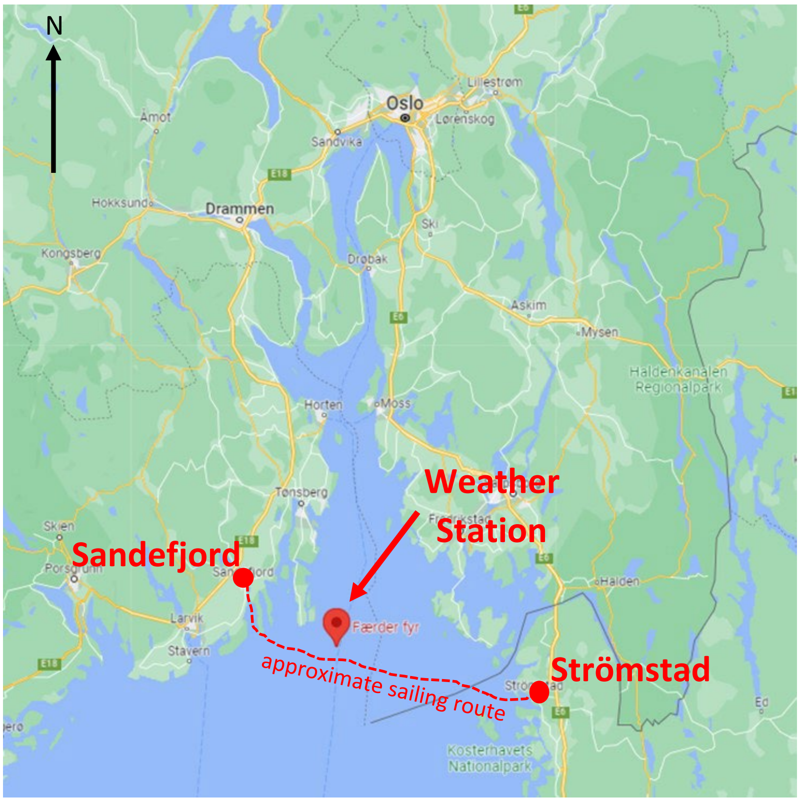

Figure 4 - Location of the Weather Station [10]

In development of the model for Color Fantasy, an analysis was performed to account for both the wind produced by the speed of the ship and the difference between on- and offshore wind. This analysis showed that these two effects did not have an appreciable impact on the energy consumption results of the IDA ICE model, so it was decided to use the more readily available onshore data [1]. For this reason, this student project uses the same approach.

Note that the climate data used in the IDA ICE model uses climate normals for this same weather station, rather than actual weather for a particular date. This is explained in more detail in Section 5.5.

Data representing the monthly average minimum, average, and maximum sea surface temperatures for the last seven years for Sandefjord and Strömstad were collected from a website [11] [12]. The values for Sandefjord are plotted by month in Figure 2Figure 5.

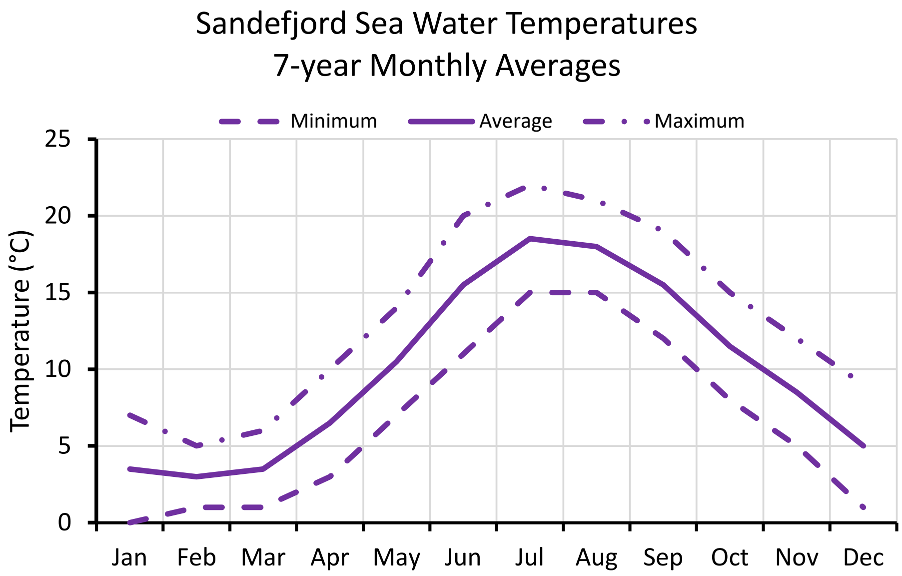

Figure 5 - Historical Sea Water Surface Temperatures, Sandefjord

Strömstad's average temperatures were within +/- 0.5 K of those shown in for Sandefjord, so they are not also included here. While the ship spends a considerable amount of time in Sandefjord, it is also out at sea. For the purposes of the analysis in this report, these harbor surface water temperatures are expected to be close enough to the open-water sea surface temperatures, but it is important to keep in mind the potential differences in surface water temperatures the ship encounters along its daily path.

4.4 Energy Use and Operational Data

Energy delivered to the ship, including diesel and electric shore power were provided by Color Line for a period of approximately January 2021 through August 2022. Under normal operating conditions, diesel consumption was provided in volumetric consumption per crossing. Electric power data from the Sandefjord shore power connection were provided as electrical energy per day. The data were analyzed to generate a method for estimating the delivered energy use during a year of normal operation. The details of the data provided and the analysis they received are described further in Section 6.1.

The electricity use of the hotel system was also provided by Color Line. These data were logged by the ship's onboard energy management system and represent the electricity delivered to the hotel system, independent of the source (batteries, generators, or shore power). The data covered the period October 13, 2021, through August 28, 2022. The data were used to generate a regression which provides an estimate of the daily average electricity use of the hotel system under normal operating conditions as a function of number of passengers and average outdoor temperature. More details of the methods used to process and analyze the data can be found in Section 6.2.

The number of passengers for each crossing was also provided by Color Line. The details of the data provided and how they are used in the analysis are shown in Sections 6.1 and 6.2.

Historical operational data logged from the ship's onboard Teknotherm energy management system, which controls the ship's 14 central ventilation systems, engine and cargo room fans, and chilled water system, were provided by Color Line. The data availability for each system varies, but can include one or more of the following:

- supply, extract, and fresh air inlet temperatures;
- extract CO_2 levels;
- supply air relative humidity;
- air pressures;
- supply air temperature, pressure, and relative humidity setpoints;
- control signals to the cooling water valve, hot water valve, humidifier, and recovery wheel;
- cooling water and hot water valve positions; and
- chilled water temperatures.

These data have been processed using MATLAB from 5-second intervals to hourly intervals. More details of the data related to the chilled water system can be found in Section 6.3, and the data related to the air handler units can be found in Section 6.4.

4.5 Ship Visit

On January 23, 2023, the author and project co-supervisor, August Brækken, visited the ship for the duration of a sailing from Sandefjord to Strömstad and back again – a journey of approximately 6 hours. The primary goal of this visit was to ascertain the difficulty of obtaining interval data representing the thermal energy consumption of the heated-water and chilled-water systems. The trip also allowed for a deeper understanding of the operation of the ship since it could be viewed in real-time. The ship's chief engineer, Jan Bjølgerud, and the ship's engineering crew assisted by showing and explaining how to operate the technical systems' monitoring and control equipment, providing tours to the main pumping and equipment rooms, and answering operational questions.

It was confirmed that the ship's onboard energy management systems have the necessary measurement points to estimate the current thermal energy loads of the heated- and chilled-water systems. The heated-water system monitoring and control system is provided by SIEMENS and the monitoring and control of the cooling water system and air handler units is provided by Teknotherm. Examples of a screenshot from the operator's control station computer monitor for the SIEMENS and Teknotherm systems are shown in Figure 6 and Figure 7, respectively.

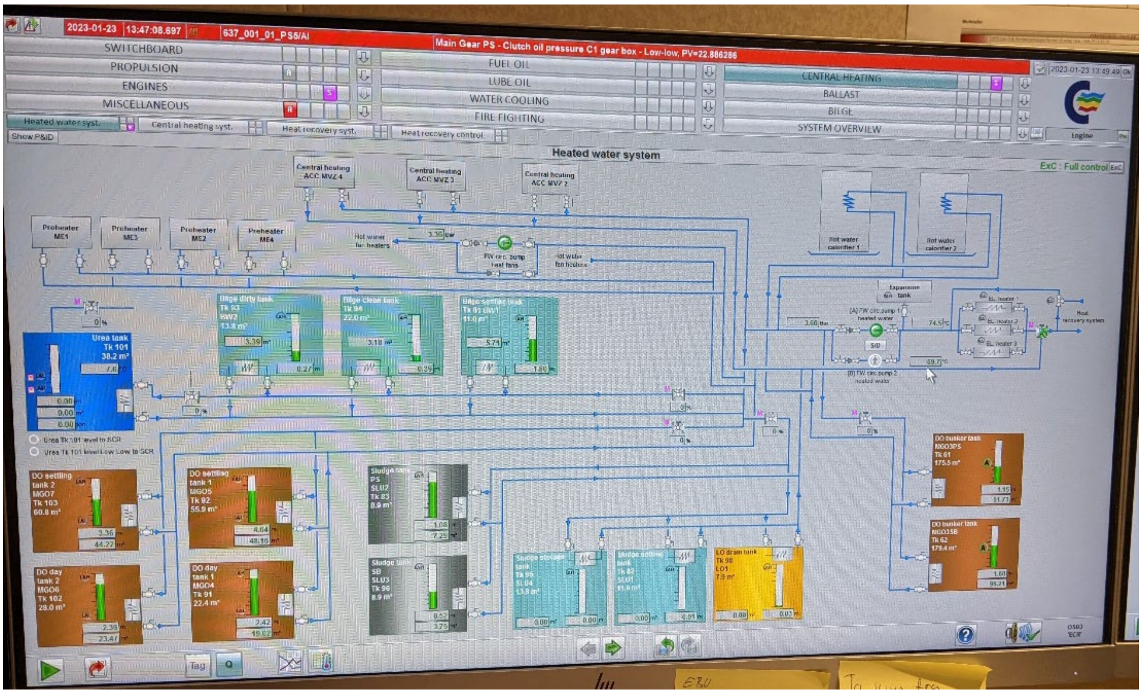

Figure 6 - SIEMENS Monitoring and Control Screen, Heated Water

The heated water system, as shown in Figure 6, receives heated water from the heat recovery system (from off the screen to the right) and sends it to the various heating loads on the ship, including the central heating systems (upper left portion of the screen) and the hot water heaters (upper right portion of the screen). Important data, which could be used to calibrate the IDA ICE model, include supply and return water temperatures, valve positions, electric heater status, and circulation pump status. Similar control screens and data are available for the central heating system and heat recovery system. The Time-series historical data for these data points were only available in graphical form but are not currently available for download from the SIEMENS system. However, it is possible that export of these data could be made available in the future with some further software development [13]. Recommendations for data collection are discussed in more detail in Section 10.1.

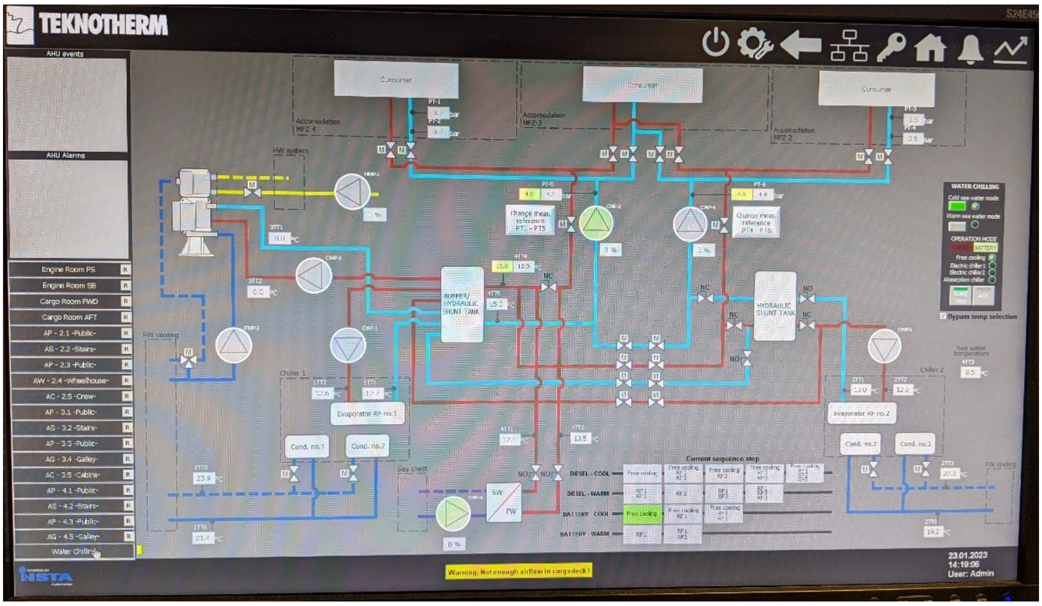

Figure 7 - Teknotherm Monitoring and Control Screen, Chilled Water

Logging of historical temperature data from the chilled water system was made available, as discussed in Section 4.4. The operator's monitoring and control screen, shown in Figure 7, was used to verify understanding of the data provided. Similar monitoring and control screens are available for the ventilation systems. While much of the data in the system is being logged, the system has additional measurement points that are not currently being logged. These would be helpful in providing a clear estimate of chilled water energy consumption. Examples of useful data not currently being logged include the speed of the circulation pumps and the operation modes of the chillers, absorption chiller, and seawater heat exchanger. These recommendations are discussed in more detail in Section 10.1.

While the simplest path to data collection is likely logging more of the data available from the onboard monitoring and control systems, the ship visit also confirmed that installation of portable data logging equipment should be possible as an alternative. For example, during the ship visit, installation of an ultrasonic flow meter seemed plausible for the heater water system in the "water mist safety room" on deck 5 and for the chilled water system in the pump room on deck 1. Alternatively, since the circulation pumps are controlled by constant pressure, logging of the power to the circulation pump motors, along with a pump curve and an adjustment for motor and pump efficiency, could be used to estimate the flow of the chilled water system.

The ship's engineering team confirmed that the flow rates in the heated water system to both the accommodation heating and hot water heaters is hand-set by control valves and that the flow rates to these systems should be constant and relatively trustworthy [13]. During the visit, data from the valves' labels were confirmed to match the drawings.

The engineering team identified the following three difficulties with operation of the ship:

1. The absorption chiller has been difficult to commission and operate, and it is not currently operational. The system was not commissioned with the original commissioning of the ship and the crew did not originally receive training on operation of the sysem. There have been difficulties with finding qualified service personnel for the absorption chiller, but it has since been properly commissioned and the crew has now received proper training. However, a hermetically sealed pump has failed. This part is on order and should be replaced before the summer season [14].
2. Automatic control of the heat recovery system does not function as necessary. This means the crew must make manual control decisions to adjust the valve positions and make use of the two storage tanks. For example, when one tank has been fully heated, the crew changes the valve settings to redirect heat from the charged tank, to the other tank. In addition, during cold periods requiring use of the heat from the storage tanks, the crew must again change the valve settings to make use of the heat in both tanks. During cold periods, when the heat recovery system has been discharged of its heat, meaning the return temperature from the heated water system is less than 55°C, the crew must manually decide and send a control signal to energize the electric heaters [13].
3. Solar heat coming through the windows can cause difficulty with overheating of the passenger areas. The window surfaces have been measured by the crew to be as high as 50°C. They have asked the window supplier to help provide insight into and recommendations to possibly improve this issue. The public fan coils and chilled air from the ventilation system do not have enough capacity to always keep up with the cooling load in areas close to the windows receiving sunlight [13] [14].

5 Model Development

The following sections describe the use of the available ship data and any important modeling decisions to develop the IDA ICE model. The goal of the modeling is to represent as accurately as possible the actual construction and operation of the ship, while at the same time being frugal with both the time spent developing the model and the model's simulation time requirement. The model developer faces many complicated and interrelated tradeoffs. For example, matching the floor areas of the zones exactly takes development time but improves accuracy, and minimizing the number of zones helps keep computational time low but raises accuracy concerns. In addition, using the zone boundaries to automatically build thermal mass into the building envelope could help with accuracy and efficiency of development. The most important of these modeling decisions are discussed here.

5.1 Zones

Development of the zones used within IDA ICE involves several tradeoffs. The number of zones, their geometric complexity, and their dimensional accuracy play a major role in the simulation time of the model, as well as its accuracy.

One way to speed up model development time is to import building information models (BIM) or computer-aided design drawing (DWG) files into IDA ICE. A BIM was not available, and the import of the DWG file was attempted, but unsuccessful after a few attempts. Rather than dimension the zones "by hand", however, a bitmap image file (BMP) file for each of the upper decks was created, imported into the IDA ICE model, and scaled to align with the size of the ship. The picture was used to speed up creation of the zones by allowing tracing of the floorplan in IDA ICE. It should be noted that the zones are approximately the correct area, volume and shape, but not exact. Dimensions were spot-checked with the dimensions in the DWG files.

While there were no specific rules used in creating the zones and the philosophy used evolved as more was learned about the ship and its systems, the following general guidelines were used:

- Zones should not extend to more than one deck;
- Areas of the ship without separating doors should be in the same zone; and
- Where the ventilation system serves separated but adjacent areas of the ship, "merging" of these areas into the same zone should be strongly considered.
    - Exception: Where these areas have significantly different ventilation rates, setpoints, occupancy rates, schedules, internal loads, or exterior exposure, these areas should be split into separate zones.

Most of the attention in developing the zones was given to the passenger decks 6, 7, and 8, since these represent the "hotel" portion of the ship.

IDA ICE has a zone multiplier feature that allows for streamlined simulations where only one of a particular type of zone is modeled and then any of the similar zones are simply multiplied using the simulation results. This tool was considered for the cabins on deck 8, of which many are of the same floor area. However, these cabins have multiple variations in exterior exposure through the ceiling and floor. A single cabin with a multiplier would not capture this variability, so this approach was decided against. The ship's design is complicated enough that there were no other areas identified that could accurately benefit from this IDA ICE feature.

Given that, and to keep simulation times low, an averaging approach was selected as a compromise. All the cabins' floor area was simulated, but the cabins were merged within each fire safety zone based on whether they were in one of three categories: have exterior wall on starboard side, have exterior wall on port side, or have no exterior wall. If there are efficiency measures whose energy modeling would benefit from a more detailed analysis of the cabins in deck 8, the cabins could be re-built as separate zones. One downside with this approach is that under actual operation, cabins near each other may occasionally require cooling while others require heating because of, for example, different internal gains or occupant control settings. By combining them into the same zone, the cabin-specific internal gains and control settings are shared, so the model's results can only convey the average need. This means the model would show both less heating and less cooling during these conditions. Overall, this is expected to be a minor contributing factor to the overall energy use of the ship, so combining the cabins to reduce runtime is likely an example of a good modeling decision. In addition, the model is usually run with the heating and cooling setpoints set a few degrees apart from one another, which will further reduce the overall amount of heating and cooling use.

Figure 8 shows the floorplan in IDA ICE of the forward section and a portion of the mid-section of deck 8, i.e., the two sections of the ship that contain all the crew cabins. The long rectangular zone at the top of the figure, which has a yellow pop-up label, represents the merging of nine cabins on the port-side of the ship. Similar merging occurred to represent the nine cabins on the starboard side, as well as in the center of the ship, where 34 cabins are merged into one nearly square zone. This is an example of how multiple rooms within the ship have been merged to create the zones within IDA ICE.

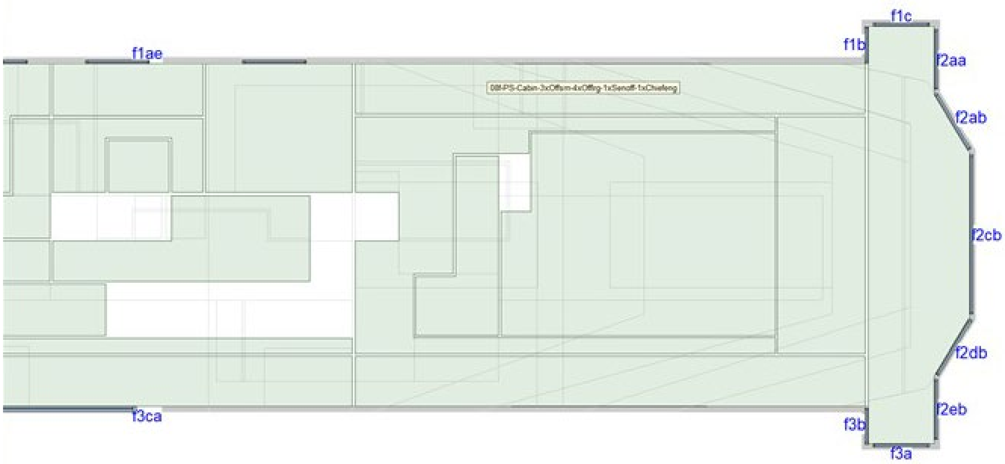

Figure 8 - IDA ICE Floorplan Deck 8 Forward

Some areas of the ship are connected by more than one deck level. These are notoriously difficult zones to model in IDA ICE given the complexity of the bi-directional airflow. Because of numerical instability issues, the IDA ICE user manual recommends avoiding the use of large openings to represent the geometry [15]. One area of the ship with this issue is the large open atrium in the aft section of decks 6 and 7. Here, after unsuccessfully exploring many options of connecting the zones in the model, the decision was to model the zones as if they are split by a floor. This simple solution keeps the model stable with low runtimes, while allowing for a clear understanding of the uncertainty it causes. Overall, the expected impact is low, however, ventilation and heat provided to these zones will be more acutely distinct from each other in the model than in the actual case, so attention to these zones' results will be necessary. While these impacts are expected to represent a very small fraction of the overall energy use of the hotel system, this atrium area represents a sizeable fraction of the area in the passenger-occupied portion of the ship and may have undue influence on certain efficiency measures.

The stairs are another example of areas of the ship that are connected over multiple decks. Separate zones were not created for all stair areas of the ship, especially in the beginning stages of defining the zones in the model. For example, the stairways in the car deck sections have not been accounted for as separate zones. While their inclusion would improve accuracy, they were left out to save development and simulation time since they are not expected to have an appreciable impact on the overall results. Upon reflection, and now with a good understanding of the operation of the systems of the ship, it would have been ideal to have built separate zones for the stairs since they are served by their own air-handling unit in each fire zone, they do not have cooling or heating equipment, and they are likely to have very different internal gains, occupancies, and schedules than the other zones. As with the atrium, however, the best approach would still likely be to keep each stair zone separate on its own deck level. The likelihood of this stair zoning strategy constituting a large part of the energy model error is quite low.

Lack of a complete set of stair zones in IDA ICE may be an issue in the future. For now, stair airflows are assigned to as nearby zones as possible. The airflow rates and energy content of that air for the ventilation system supplying air to the stairwells will be somewhat off. This occurs on decks 3 through 5 (car decks), which have been assigned to the combined stair/other zone on deck 6. This is not expected to have a large effect on the overall results, but a detailed analysis of the ventilation system or air quality in these zones may justify revision.

The zones representing the car decks on decks 3, 4, and 5 are sized according to the areas, ceiling heights, and volumes shown in the Ventilation System for Cargo Hold Cargo Decks 3, 4 & 5 drawing. This will allow correct air change rates, which will likely be important given the expected large energy use by the fans to produce high flows.

At the time of creating the zones for rooms in decks 1 and 2, limited data were available for heating systems, but there were no data for the cooling systems. Therefore, the guidance for zone creation was the same as that used to develop zones for the passenger decks, but with the added guideline of only creating zones for areas that appeared to have occupants – that is, those areas with space heating. Other areas that receive cooling water were not included in the model. However, these are mostly areas that appear to receive cooling for the purpose of keeping battery rooms, engine rooms, or computer equipment adequately cooled and therefore would be considered separate from the hotel system, which caters more to the passengers' comfort needs. However, the cooling loads in these equipment areas are met with the same cooling water system serving the passenger comfort systems, so depending on the energy use data received and efficiency measures tested, these areas may need special attention or revisions to the zone definitions.

There are large areas of the ship on decks 1 and 2 that are not included in the model. They are empty building bodies. Another approach would be to create large unconditioned zones to represent the steel structure that is exposed to ambient air and seawater. This may provide a more realistic representation of the actual conditions in those decks, rather than empty building bodies, which are not simulated with any heat transfer. These big zones would likely need to have proper accounting of the heat output from the engines and the exhaust ventilation that exists in the engine rooms and associated rooms. The exhaust ventilation in these zones affect the temperatures and use electricity - it is difficult to separate the temperature effects (heat from the engine rooms) and electrical energy (required to vent fuel/exhaust) from the diesel engine systems. This is an example of where there is overlap between the ship's drive system and hotel system that is difficult to disaggregate.

Table 4 gives an overview of the zones built into the IDA ICE model, by deck. There are a total of 58 zones, making up 18,735 m^2 of the floor area of the ship.

Table 4 – Model Inputs, Zones

Deck	Floor Area, m^2		Zone Count		Contents
10	-	464	-	12	Roof
09	464		12		Misc.
08	3,020	8,742	19	33	Passengers
07	2,850		7		
06	2,872		7		
05	2,624	8,705	1	7	Cars
04	3,012		1		
03	3,069		5		
02	823	823	6	6	Misc.
01	-		-		
Total		**18,735**		**58**	

A three-dimensional rendering of the final model is shown in Figure 9.

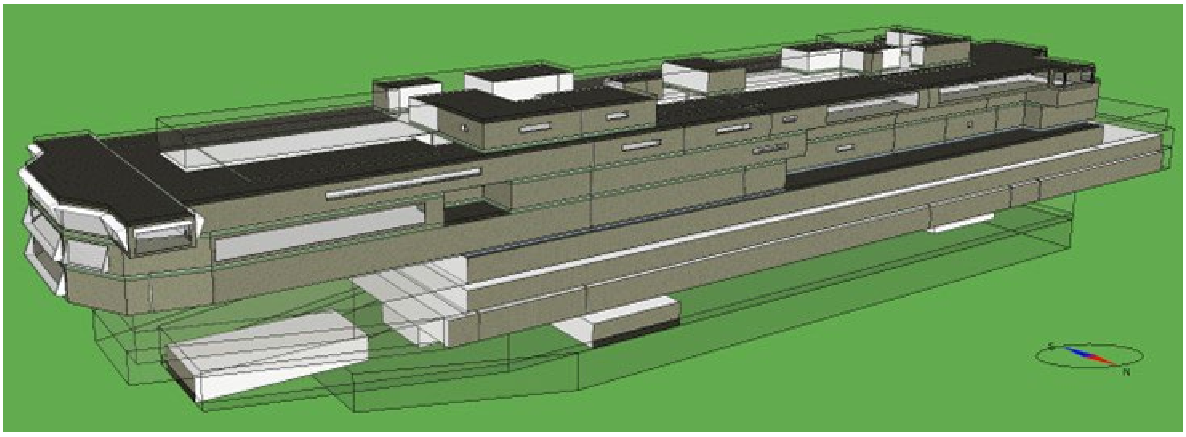

Figure 9 - 3D IDA ICE Model

5.2 Construction

The following sections describe in detail the insulation levels, windows, and air leakage of the ship and how they are represented in the model.

5.2.1 Insulation

Insulation characteristics for bulkheads and decks were developed in IDA ICE using information contained in the "Thermal/Sound and Insulation Plan" and "Insulation Details" drawings. The "Thermal/Sound and Insulation Plan" drawings show insulation locations for both the deck heads ("ceiling" of that deck level) and the bulkheads ("walls"). The "Insulation Details" drawings show the details for the insulated deck and the insulated bulkhead.

There are many different insulation details spread throughout the ship. Rather than enter all these details into IDA ICE and linking them to the associated components, which would have been unnecessarily time-consuming, the most common insulation details were used, and similarly insulated components were assigned to these.

Figure 10 shows an example of the construction definition data entry in IDA ICE. In this case, it represents an exterior insulated bulkhead. The finishing details were not provided in the drawings, so a thin aluminum plate was assumed as the interior surface and an air gap was used as a placeholder to provide additional thickness to the wall to align with the most common thickness that appears in the floorplan drawings.

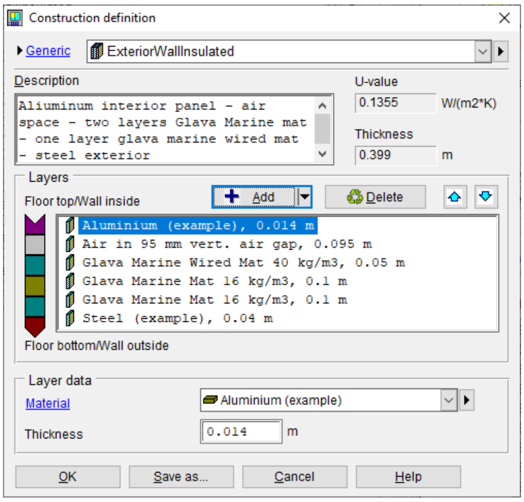

Figure 10 – IDA ICE Construction Definition, ExteriorWallInsulated

Table 4 shows the overall U-value of each construction type used in the IDA ICE model, along with its name, deck location, number reference from the "*Insulation Details*" drawing, and total thickness. The steel components are assumed to be 4 cm thick, with a thermal conductivity of 0.193 W/m·K. The thermal conductivity of the insulation is assumed to be 0.036 W/m·K, based on manufacturer data sheets. Since there are only seven construction types used, they should be simple to adjust as necessary, either to better align with the as-built structure, or to test alternative insulating strategies.

Table 5 – Model Inputs, Deck and Bulkhead Constructions

Type	IDA-ICE Name	Decks	Insulation Detail Number	Total Thickness, m	Overall U-value, W/m2K
Deck	InternalDeck	6-7 and 7-8	2.5 and 6.5	0.397	0.45
	InternalInsulatedDeck	5-6	9.4	0.318	0.13
	RoofInsulated	roof of 8 and 9	20.2	0.397	0.11
Bulkhead	ExteriorWallInsulated	6, 7, and 8	20.2	0.399	0.14
	ExteriorWallUninsulated	5 and lower	n/a	0.040	5.86
	InteriorSteelWallInsulated	6, 7, and 8	1.2, 5.2, 3.2	0.200	0.43
Interior partition	InteriorPartitionUninsulated	6, 7, and 8	n/a	0.050	2.89

Thermal bridges are expected to be significant since the entire structure is made of steel, a building material with relatively high thermal conductivity. IDA ICE lends itself to easily adjusting the thermal bridge values. Although it appears in the "*Insulation Details*" drawings that an effort has been made to counteract the thermal bridges by extending insulation along the bulkhead 0.45 meters into the conditioned space, the thermal bridging through the steel is still expected to be significant. As shown in Figure 11, the thermal conductivity values in IDA ICE have been set to "very poor". The proper setting of these values to match the as-built ship is an area that could benefit from further study. In addition, optimization of the method used to overcome thermal bridging could be a fruitful passive efficiency measure worthy of further study.

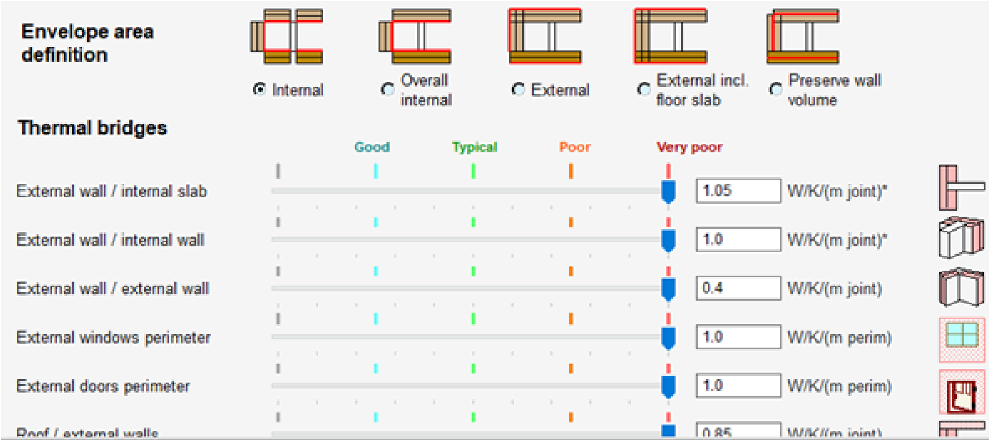

Figure 11 – Model Inputs, IDA ICE Thermal Bridges

Interestingly, many of the interior bulkheads are insulated to provide fire protection, and perhaps sound and thermal insulation. These insulated steel walls were included in the IDA ICE model, where they occurred, to separate the zones. This had the added benefit of accounting for some of the thermal mass of the steel in the ship's construction.

5.2.2 Windows

Drawings "*700-515-001-5*" and "*700-515-002-2*" provide details related to the windows on the ship, including window construction, recess depth, location, and size of each window. The IDA ICE "standard" (versus "detailed") window data entry was used to enter the efficiency metrics for the window assembly. Here, the primary efficiency metrics of interest are the solar heat gain coefficient and U-value of the glazing component of the window assembly, along with the frame fraction and U-value of the window frame.

Glazing U-value and recess depth were entered as given in the drawings. There, the guidance U-value for the glazing is stated as 1.5 W/m^2K. The one exception is that the glazing U-value is 5.8 W/m^2K for the heated wheelhouse windows[2]. Recess depth varies from 0.22 meters to 0 meters, depending on window type and location.

There are three locations in an installed window where thermal bridging could occur:

1. Between the panes of a multi-pane window assembly;
2. Through the frame of the window; and
3. Between the frame and building connection.

Since IDA ICE includes an overall glazing U-value in its "standard" glass construction data entry, the first case is already included in the U-values mentioned above. It is noted in the window drawings that the windows panes are separated from each other with a rubber gasket to minimize thermal bridging. The second case, thermal bridging through the window frame, is covered by the frame U-value and frame fraction entries in the window data entry screen. The U-value is assumed to be very

[2] The drawing for the wheelhouse windows (*700-515-006*) is not given in the files we have. However, it appears that the windows have similar efficiency as the others, since they have energy efficient glass indicated. This is contrasted by the fact that they are listed as having heaters for defrost in the general discussion in the windows document *700-515-001-5-Notes*, as well as a "guidance" U-value of 5.8 instead of 1.5 for the other windows. In the end, the wheelhouse windows are modeled in IDA ICE with a glass U-value of 5.5 to account for these factors. Modeling of the heating of the windows is likely not easily added to IDA ICE, so a higher U-value would help account for this in the overall energy use results.

poor, 5.5 W/m^{2}K, because of the required use of steel for the ship's window frame. The frame fraction entry is described below since it is connected to the issue of window merging. The third case, thermal bridging through the frame and building connection, is covered in the thermal bridge setting in IDA ICE, which is set to "very poor" (1.0 W/m·K). As mentioned previously, this value is easy to change in IDA ICE.

Determining the as-built thermal bridge values would require better information than that found in the window drawings mentioned. While detailed, the drawings do not clearly state the materials used in the window frame components. Knowledge of the materials' thermal conductivities is required to estimate properly the thermal bridge values. However, if needed, further detailed modeling of the as-built window performance could be made.

To keep simulation time low, the windows in each zone were merged so that each exterior surface has no more than one window. A calculation was made to find the final merged window dimensions, ensuring the total glass area and total exposed frame area are both correctly accounted for. At the same time, the frame perimeter of the merged window was matched as closely as possible with the sum of the frame perimeters of the actual windows. The frame perimeter of the merged window will most likely be lower than the sum of the frame perimeters of the actual windows. By reducing the height of the merged window, the merged window mimics the longer frame perimeter and reduced light and solar gain caused by the vertical frame components of the individual windows. Given the uncertainty in the thermal characteristics of the window frames, this admittedly crude adjustment should not be of concern regarding the overall model results. However, if efficiency measures related to window size or shape are explored, the impact of this window merging should be considered carefully.

The windows on the front of the ship have either an inward or outward tilt, which affects their solar heat gain, among other things. This was accounted for in IDA ICE by use of the tilt angle in the standard window dialog box. Figure 12 shows this in the 3-D view in IDA ICE, where the windows in the wheelhouse, with a 24-degree outward tilt, are highlighted in red.

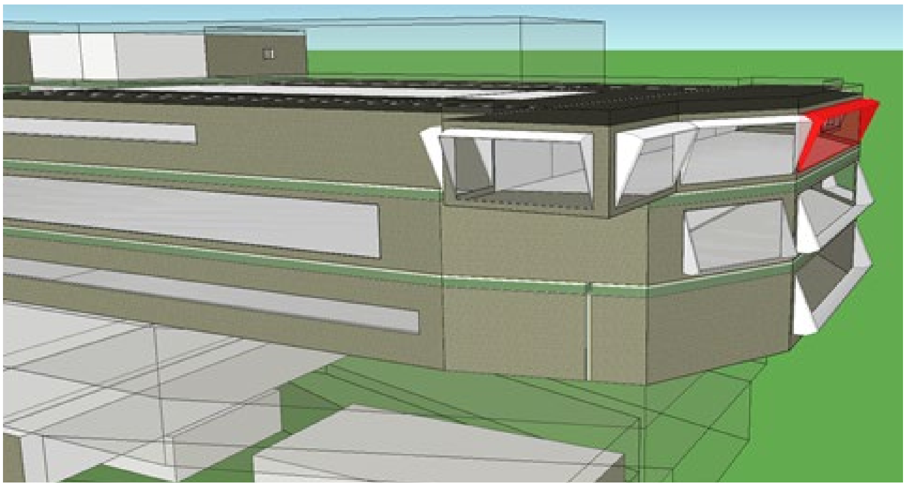

Figure 12 - Window Tilt on Forward Decks

5.2.3 Air Leakage

The infiltration is effectively set to zero in the IDA ICE model because:

- Pressure coefficients are set to zero. Non-zero values are required in order to consider wind-driven infiltration. This applies when the “Wind driven flow” option is selected, as opposed to the “Fixed infiltration” option, in the Infiltration inputs in IDA ICE.
- Internal air passageways have not yet been included in the model. The magnitude of wind driven infiltration is impacted by the size of internal leaks and openings, but these passageways would also allow for the buoyancy effect to cause pressure differences between zones on different decks, and thus infiltration, to be modeled between zones.

The ship is expected to be very air-tight, given its steel construction and sea-going capability, so this is a reasonable starting point for the model. However, more information on the actual leakage of the structure would be helpful since large leaks, should they exist, could lead to large uncontrolled heat loss or heat gain that should be accounted for in the model.

5.3 Internal Gains

Internal gains are defined as heat-producing devices and people that give off heat to the spaces within the ship. These are described in the following sections and include provisional cooling equipment, occupants, lighting, and other equipment.

5.3.1 Provision Cooling Equipment

Provision cooling systems on the ship consist of refrigerant piping circuits that connect refrigeration system components (compressors, etc.) to multiple provision cooling requirements (for example evaporators located in cold rooms and walk-in freezers). Currently, these systems are not accounted for, and the cold and frozen rooms served by these systems are not represented in the IDA ICE model. Depending on future energy use data and modeling needs, these could be added to the IDA ICE model or accounted for in a post-processing step.

5.3.2 Occupancy

Assumptions were made to establish the number of people and their timing within the zones in the IDA ICE model. On average, the ship is assumed to have a passenger and vehicle occupancy rate of 80%, and an employee occupancy rate of 90% during the day. Half of the employees present during the day are assumed to occupy the ship overnight, and 20% of the passengers are assumed to make a round-trip journey without going off the ship.[3]

The ship’s sailing schedule was used, along with the assumptions above, to create average daily occupancy levels for the ship. Figure 13 shows the resultant daily occupancy levels for the three occupant types: passengers, employees, and cars.

[3] Occupancy data (Section 4.3) will be used to adjust these original assumptions in the next version of the model.

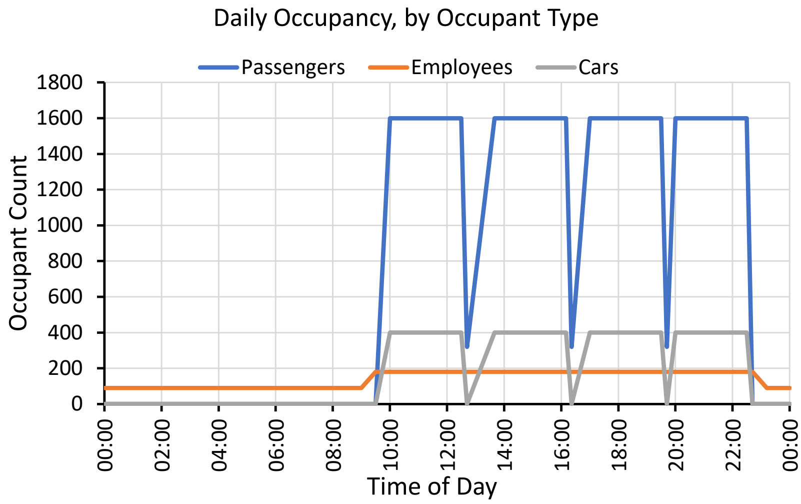

Figure 13 - Model Inputs, Daily Occupancy by Occupant Type

The passenger and employee occupants were divvied into each of the ship's IDA ICE zones based on the expected use of the zone. The following criteria were used to assign the zone to one of five different occupancy profiles:

- Presence and number of seats available to passengers,
- Expected availability of the zone to passengers,
- Whether the zone was unlikely to have occupants (e.g. equipment rooms), and
- Whether the zone was a sleeping cabin.

The five occupancy profiles are named and described as follows:

- PassengerSeatingAreas – Zones with designated passenger seating receive that number of onboard passengers, adjusted by an assumed passenger seated rate of 85%. These zones are assumed to have zero employees.
- RoamingAreasDay – Zones without passenger seating, but that are likely available to passengers, receive the remaining passengers at a rate proportional to the fraction of that zone's floor area to the total floor area of zones with this occupancy profile.
- EmployeeOnlyAreasDay – Zones likely to be unavailable to passengers and only occupied by employees. This includes daytime occupancy in the cabins, in addition to other employee-specific areas like galleys and offices.
- EmployeeNightAreas – Zones identified as cabins. This occupancy profile only includes cabin occupancy during the night.
- Unoccupied Areas – These include areas such as equipment rooms, car decks, or other areas not expected to be regularly occupied, but that are represented as zones in the IDA ICE model because they have space heating, ventilation, or cooling equipment.

The average daily occupancy levels of the first four occupancy profiles are shown in Figure 14. These are the profiles used in the IDA ICE model.

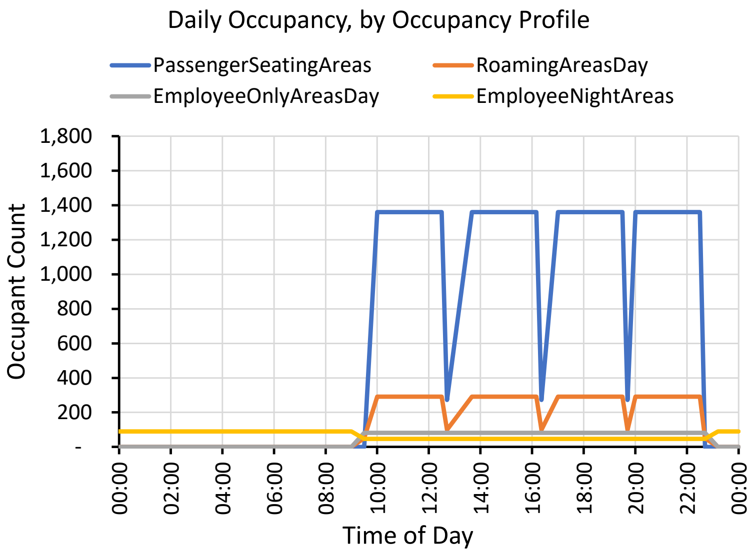

Figure 14 - Model Inputs, Daily Occupancy by Occupant Profile

Any heat gain from cars or people on the car decks is expected to have inconsequential impacts on the energy use of the hotel system, since the car decks have high ventilation rates and no ventilation heat recovery, since the car decks are isolated from adjacent zones with insulation, and since they are minimally heated compared to their heating load. Therefore, the car decks are not assigned occupancy of people or cars in the IDA ICE model.

5.3.3 Lighting

Details of lighting equipment in the ship were not available, so generic LED lighting was assumed, given the recency of the ship's construction. The installed lighting power density levels are based on the maximum lighting power levels given in the Norwegian Standard for "Energy Performance of Buildings: Calculation of Energy Needs and Energy Supply", SN-NSPEK 3031:2021, for building types related to the expected ship zone use type [16]. These values, along with the associated floor areas, for the three different zone use types are shown in Table 6.

Table 6 - Model Inputs, Lighting Power by Zone Use Type

Zone Use Type	Floor Area, m^2	SN-NSPEK 3031:2021	
		Building Type	Lighting Power Density, W/m^2
Other Areas	5,770	Office Building	3.7
Hotel Areas	968	Hotel	2.4
Retail Areas, except Restaurants	3,465	Business Building	7.8

The car decks are not included in Table 6. Instead, a lighting power density of 1.4 W/m^2 is assumed, which comes from ASHRAE 90.1 lighting power density allowance for parking garages [17].

Two load shapes are used to represent the power levels of the lighting systems over the course of the day. These follow somewhat the shape of the occupancy schedules. The employee night areas, however, have reduced power levels during expected sleeping hours. The two zone types for scheduling lighting are called LightingFrequent, which represents areas of the ship expected to have frequent occupancy (or example kitchens and offices), and LightingInfrequent, which represents areas of the ship expected to have infrequent occupancy (for example storage and equipment rooms). The lighting schedules for these two zone types are shown in Figure 15 in terms of the fraction of installed lighting power throughout the day.

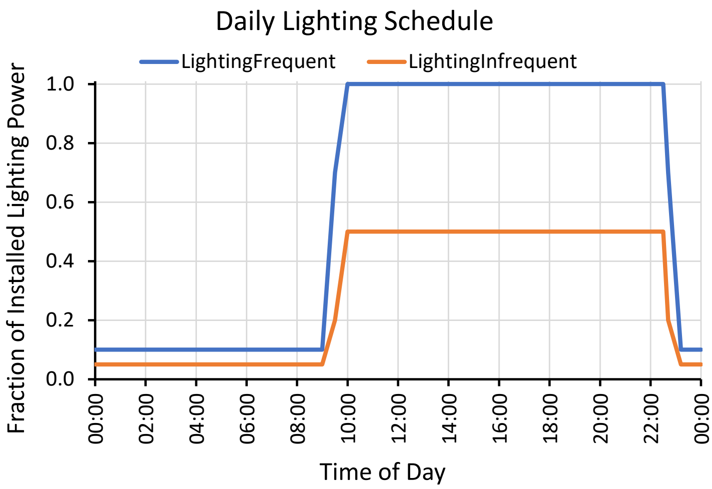

Figure 15 - Model Inputs, Daily Lighting Schedules

5.3.4 Other Equipment

To account for heat from equipment such as computers, audio-visual equipment, and other heat-producing devices, the equipment power densities from the Norwegian Standard for "Energy Performance of Buildings: Calculation of Energy Needs and Energy Supply", SN-NSPEK 3031:2021 [16], were used for building types related to the ship zone use type. This is similar to the methodology used for lighting, however consideration of whether a zone was expected to have heat-producing equipment led to the zones being aligned with different building types in this case. The values used and associated floor areas for the three different zone use types are shown in Table 7.

Table 7 - Model Inputs, Equipment Power by Zone Use Type

Zone Use Type	Floor Area, m^2	SN-NSPEK 3031:2021	
		Building Type	Equipment Power Density, W/m^2
Offices and Equipment Rooms	1,028	Office Building	8.6
Hotel Areas	1,143	Hotel	1.3
(not used)	0	Business Building	17.2
Other Areas	8,032	Cultural Building	1.5

There are important differences between Table 6 and Table 7. The business building, which is described as a retail store or a gas station, was used to represent the lighting power density in the retail areas of the ship because of the expected high levels of lighting used to market the merchandise. However, the cultural building, described as a movie theater, museum, library, or a building for religious activities, was used to represent the equipment power density. This is because the use of equipment in the passenger areas of the ship is expected to be better represented by a library, for example, because of libraries' much lower use of heat-producing equipment.

Given the high level of uncertainty in the actual load levels and their operating schedule, the load shape used to represent the power levels of the heat-producing equipment is the same as the LightingFrequent shape shown in Figure 15.

Rooms that use hydronic space cooling to cool equipment rather than people, are found in the lower and upper decks. These rooms house heat-generating devices such as batteries, IT/computer equipment, and engines. Depending on whether these rooms end up being added as zones to IDA

ICE, the rooms' equipment heat output rates should be aligned with expectations of the specific equipment and scheduled accordingly. This model modification could be made after receiving and analyzing more energy use data and after further studying the equipment design details. Alternatively, a post-processing adjustment could be made outside of IDA ICE.

5.4 Energy Systems

The HVAC system consists of ventilation plants, zonal heating, zonal cooling, and domestic hot water systems that are served by hot-water and cold-water plants. The hot and cold-water plants are powered by a combination of waste heat from the ship's diesel engines and electricity from the ship's generators. The drive system and how it supplies hot and cold water is not covered in detail in this report, but a high-level understanding of the hot-water and cold-water plants is helpful for proper understanding of the systems they serve. Attention to these systems should be included in later versions of the model, or handled in models outside IDA ICE. The focus of this project thesis is on the energy demands of the zonal heating and cooling systems and the ventilation plants.

5.4.1 Heated Water Systems

There are two heated water loops serving different loads in the ship: a main loop and an accommodation loop. The lower decks and the domestic hot water system are served by the main heated water system loop, which delivers 75°C water. Other loads on the main heated water system include the heat exchangers that serve the 70°C accommodation heated water systems, which provide heat to hotel decks 6 through 9. The main loop receives its heat from a combination of diesel engine heat recovery and electric heaters[4]. A thermal energy storage system consisting of two as well as heat exchangers to recover waste heat from the engine exhaust and engine cooling systems. In addition to the electric storage with the batteries, the ship has two 140 m^3 water storage tanks are integrated into the main heated water loop. designed to store heated water from the waste heat system at up to 90°C. The storage system, which can provide useful heat to the ship's systems until the water is cooled down to 75°C, amounts to a total capacity of nearly 4.9 MWh.

The operation of the heat sources depends on the operating mode of the ship and the load levels, which is described in "*Heat recovery and hot water system – function description*" document. The workbook "*Heat Balance Calculation*" gives an indication of the overall design winter heat load of the ship and an explanation of how the heat recovery and heated water storage system works over the course of the different operating modes. The "*Heated Water System*" and "*Central Heating System*" drawings describe the heated water system loops and provide information about the sizes of the heating loads served by the loops. These data were used to help inform sizing of some of the zonal heating equipment in the IDA ICE model, particularly those located in the lower decks.

IDA ICE has the functionality to model the various systems that provide heat to the heated water loops, along with their detailed interaction with the loads. These details are not included in the current version of the model. Instead, a heating COP of 1.0 is assumed. Therefore, the model results provide the amount of heat energy used by the hotel system's heated water loads. Adding heated water plant characteristics to the model could allow a clearer understanding of the heat sources used under the varying operating conditions and the amount of delivered energy required by those sources. This model enhancement could be done in future work (Section 0). In this report, the hotel load outputs from the IDA ICE model are used in separate analyses outside the IDA ICE model.

[4] The main loop has a 450 kW and a 378 kW electric heater, while the three accommodation hot water loops also each have 270 kW electric heaters. Together, these are sized to cover the design heating load of 1200 to 1300 kW.

5.4.2 Chilled Water System

Based on the *Chilled Water Diagram* drawing, the chilled water plant, which is designed to provide 14°C water, consists of the following three types of equipment:

- A heat exchanger connected to seawater, to allow for direct "free" cooling from cold seawater;
- An absorption chiller with 400 kW cooling capacity, fed by the 90°C heat recovery system; and
- Two vapor compression chillers with 919 kW cooling capacity each, using electricity.

The operation of the various modes of the chilled water plant is described in the *Chiller Sequence* document (*45117911 chiller sequence R09 30.04.2019*). The summary of priority given to the cooling equipment is:

1. free cooling (if cold seawater),
2. absorption chiller (if waste heat available),
3. electric chiller 1, and
4. electric chiller 2.

Like that described above for the hot water plant, the IDA ICE functionality is not used to model the cold-water plant. The COP is currently assumed to be 1.0, so the model results provide the amount of cooling energy used by the hotel system's cold-water loads. In order to estimate the delivered energy requirements of the cooling water system, more detailed knowledge would be needed regarding the operation of each of the four cooling systems. Improving the functionality of the model to account for the consumption and operating sequence of the various cooling systems is described as a part of possible future work (see Section 0).

5.4.3 Space Heating and Cooling

The hydronic zonal heating systems on the ship consist of a combination of convectors, public fan coil units (PFCU), cabin units, fan heaters, and duct heaters. The hydronic zonal cooling systems are a combination of public fan coil units (PFCU), chilled beams (CHB), fan coils, and rack coolers. The *HVAC Ventilation Ducts Layout*, *Principle Duct Arrangement,* and *CW Diagram Sheets* drawings were used to collect locations, type, and capacity data on every unit in the ship. Table 8 and Table 9 show the total capacity of the different types of hydronic zonal heaters and cooling equipment, respectively. The data for each individual heating and cooling unit on the ship were assigned to the associated IDA ICE zone. The heating and cooling capacity is concentrated to passenger decks 6, 7, and 8, as shown in Figure 16. The 15 kW of heaters and 409 kW of cooling equipment identified as "na" represents the units not assigned to a zone in the IDA ICE model.

Table 8 – Hydronic Zonal Heater Types and Capacities

Type	Capacity, kW	
Convector	253	
PFCU	104	
Cabin Unit	182	727
Fan Heater	89	
Duct Heater	99	

Table 9 - Hydronic Zonal Cooling Equipment Type and Capacities

Type	Capacity, kW	
PFCU	105	575
CHB	8	
Fan Coil	235	
Rack	226	

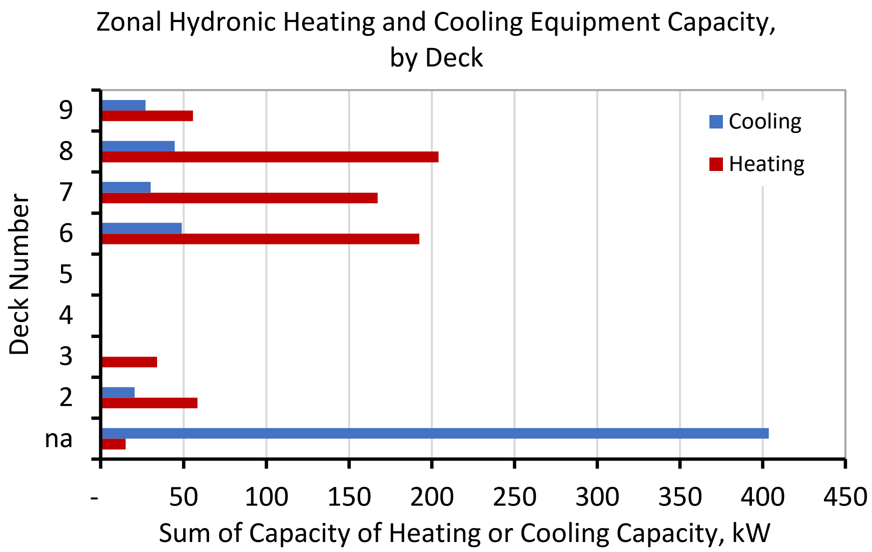

Figure 16 - Location and Capacities of Hydronic Zonal Heating and Cooling Equipment

Because the IDA ICE zones were built prior to having access to the cooling equipment design data, many of the rooms in the ship that receive only zonal cooling, that is without zonal heating, are not included as zones. These rooms are located on decks 1, 2, and 9, and contain some type of heat generating equipment that needs cooling. In other words, these do not appear to be regularly occupied areas and could be considered outside the hotel system of the ship. Their inclusion, as either zones or as post-processing adjustment outside IDA ICE, could be considered after receiving the ship's energy use data.

The sum of the heating and cooling capacities of the hydronic zonal units for each zone was added to the IDA ICE model as a simple fan coil unit with hydronic heating and cooling. In zones with no heating, a small heating capacity of 0.1 W was entered. In zones with no cooling, the cooling was set to 0 watts[5]. The other inputs, such as fan power (3%), are currently left at the defaults. Regarding fan power, the IDA ICE user manual states this can be zeroed out to improve simulation speed [15].

The car deck zones have been set to use electric resistance space heaters. The design power is set to 50 kW for each of the three car decks (deck 3, 4, and 5) based on data found in the "*Electric Load calculation – winter condition"* design document. The other zones use hydronic heating and cooling in the fan coils.

The technical specification document for the ship (*181127_Technical Specification (Teknotherm) - R01*) indicates that COMF-C(2) is used throughout the ship. The term COMF-C(2) aligns with the term

[5] Data entry using the IDA ICE table would not allow a zero for the heating capacity. It appears at the time of writing, however, that a zero is, in fact, allowed as a valid data entry. Regardless, this will not have a meaningful impact on the results whether it is corrected or not.

"crn 2", which refers to a medium level of indoor climate comfort per the ship classification rules [18]. This provides the allowed air temperature range for the ship and insight into how the ship is likely operated. To simplify the interpretation of the results, common setpoints are used for all zones in the IDA ICE model. These heating and cooling setpoints are shown in Figure 17, along with how they compare to the rules for comfort class 2 and the different zone types. Zone types A and B share common settings, and they include cabins and hospital rooms. Zone types C and D have the same settings, and they include wheelhouse, control rooms, office areas, and public spaces.

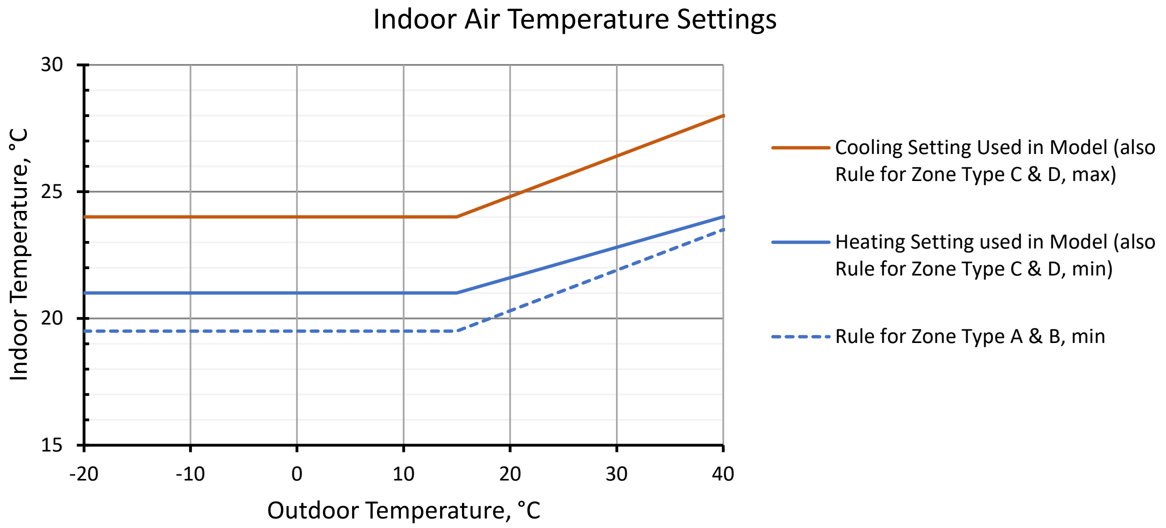

Figure 17 – Model Inputs, Temperature Settings

Heating and cooling temperature settings can have a significant impact on the energy use of the ship's hotel system. Incorporating the actual space heating and cooling setpoint temperatures in each zone could improve the model's representation of the ship's actual energy use.

5.4.4 Ventilation

Each of the three fire zones on the ship has its own set of ventilation air handler units (AHU). There are separate plants with heat recovery ventilation that provide variable air volume (VAV) and constant air volume (CAV), and non-heat recovery plants that provide ventilation air to stairs, galleys, and wheelhouse. In total, there are 14 ventilation plants, all of which are physically located on deck 9, with ductwork connecting them to their respective zones. The ventilation system is separate from the hydronic zonal heating system, except for the case of the cabin units, which act as reheaters by providing both heat and ventilation fresh air.

Detailed data and drawings for each ventilation system are available in the "*Air Treatment Equipment*" and "*Automation*" folders within the "*571 Ventilation & AC Plant*" folder. Table 10 shows some of the important characteristics of each system, including the design amount of supply and return air, percentage of return air, temperature effectiveness of the heat exchanger, the capacity of the hydronic heating and cooling coils, the electric power draw of the humidifiers, the sum of the measured supply air, and the assigned name of the air handler unit included in the IDA ICE model. According to the design data, air handlers used for stairs do not have heat recovery. Instead, they have return air recirculation. The fresh and recirculated air ratio is adjusted by manual dampers. The public and crew area AHUs have no recirculation and have heat recovery instead. Galley AHUs have 100% fresh air with no heat recovery. Airflow in galleys is 40 air changes per hour (ACH), but manually adjusted variable speed control allows demand-based ventilation. In addition to the heating coils in the air handlers, the galleys have duct heaters downstream of the air handler for further heating of the air delivered to the galleys. The galley on deck 7, which is served by AG-3.4, has an air heater with a 27 kW capacity. The galleys on decks 8 and 9, which are served by AG-3.4, have deck heaters with capacities of 11.8 kW and 9.7 kW, respectively. The AHU serving the wheelhouse has 100% recirculated air. Table 10 also shows the specific fan power (SFP) for both the supply and, if available, the return fan. These data come from the individual air handler specification sheets.

Table 10 - Summary Characteristics of All Ventilation Plants

System Number	Design Supply Air (m3/hr)	Design Exhaust Air (m3/hr)	AHU Type / Return Air %	Temperature effectiveness of EE	Heating Capacity (kW)	Cooling Capacity (kW)	Humidifier Electrical Power (kW)	Sum of Measured Supply Air, m3/hr	IDA-ICE AHU Assigned Name	SFPv.supply, kW/(m3/s)	SFPv.exhaust, kW/(m3/s)
AP-2.1	14,058	11,837	EE	74.3	27.4	142.0	11.4	14,101	**VAV**	2.75	2.00
AS-2.2	3,500	1,400	40 %	n/a	50.6	27.5		3,597	**Stairs**	1.95	
AP-2.3	17,690	15,921	EE	70.1	46.9	165.0	19	17,750	**VAV**	2.67	1.96
AW-2.4	14,057	-	100 %	n/a	29.1	53.9		13,920	**Wheelhouse**	2.05	
AC-2.5	12,302	10,236	EE	75.7	21.3	126.0	11.4	12,345	**CAV**	3.05	2.74
AP-3.1	11,744	11,036	EE	74.2	21.9	109.0	11.4	12,037	**CAV**	2.94	2.11
AS-3.2	3,559	1,424	40 %	n/a	50.5	27.2		3,582	**Stairs**	1.95	
AP-3.3	14,416	13,010	EE	74.1	29.7	141.0	11.4	14,520	**VAV**	2.40	1.99
AG-3.4	14,100	-	0 %	n/a	209.0	101.0		13,268	**Galley**	2.08	
AC-3.5	12,760	10,966	EE	76.3	23.0	110.0	11.4	12,356	**CAV**	3.04	2.52
AP-4.1	15,000	13,908	EE	73.5	30.3	149.0	15.3	15,001	**VAV**	2.78	2.00
AS-4.2	5,284	2,114	40 %	n/a	60.2	36.0		5,268	**Stairs**	2.12	
AP-4.3	14,600	13,140	EE	70.4	43.9	143.0	15.3	14,491	**VAV**	2.94	2.13
AG-4.5	5,244	-	0 %	n/a	70.5	37.0		5,291	**Galley**	2.14	

Distinction of the equipment across fire zones is not expected to be important to the results, so a simplification was made by combining ventilation systems, according to their primary characteristics (heat recovery/% return air), across fire zones prior to their inclusion into the IDA ICE model. This was done to help keep simulation runtime low. The distinction between the AHUs with VAV and CAV is retained since the VAV system has the capability to reduce energy use with lower airflows. Combining them into one air handler unit in the IDA ICE model, while being computationally more efficient, would dilute the savings achieved by the speed reduction of the VAV system. The AHU serving the wheelhouse (100% return air recirculation) is entered into the IDA ICE model as an AHU, rather than a zonal system, because this is likely how it is accounted for in the actual energy data. Table 11 shows the results of this simplification. Supply airflow rates were summed, temperature effectiveness values of the heat exchangers were weighted according to supply airflow rates, heating and cooling capacities were summed, and SFPs were weighted according to fan power.

Table 11 - Summary Characteristics of Simplified IDA ICE Ventilation Plants

Assigned Name	Measured Supply Air Max (m3/hr)	Temperature effectiveness of HX (%)	Return Air Recirculation	Heating Capacity (kW)	Cooling Capacity (kW)	Humidifier Electrical Power (kW)	average SFPsupply (kW/(m3s))	average SFPexhaust (kW/(m3s))
VAVmainAHU	75,863	72	na	178	740	72	2.706	2.013
CAVmainAHU	36,738	75	na	66	345	34	3.014	2.474
WheelhouseAHU	13,920	na	100 %	29	54	-	2.050	
StairsAHU	12,447	na	40 %	161	91	-	2.022	
GalleyAHU	18,559	na	0 %	280	138	-	2.096	
TOTAL	**157,527**			**714**	**1,368**			

The "standard air handling unit" in IDA ICE was customized by adding a steam humidifier for both the VAVmainAHU and CAVmainAHU units. The humidifier is controlled to turn on when the relative humidity in the supply airstream goes below 20%. The ship uses electric-powered steam humidifiers, so the energy source in IDA ICE for the humidifier is electricity. There is no capacity limit built into the control of the humidifier in IDA ICE, so the electric power requirements of the humidifier in the IDA ICE results should be verified to not exceed the values in Table 11. A screenshot of the air handler schematic in IDA ICE is shown in Figure 18.

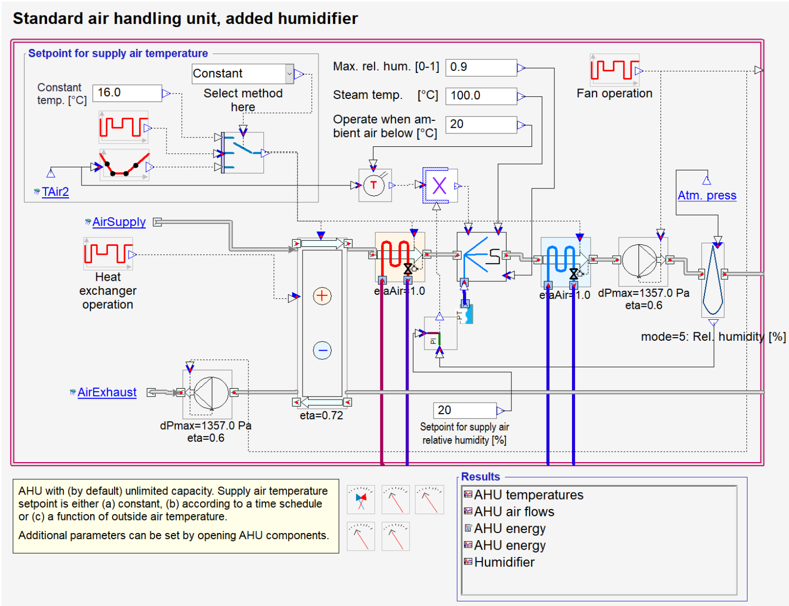

Figure 18 - IDA ICE Air Handling Unit with Humidifier, VAV and CAV

The air handler servicing the wheelhouse, WheelhouseAHU, uses 100% recirculated air. A "mixing box (recirculation)" type air handler was used in IDA ICE for this application, where the mixing box is set to use 0% outdoor air. The AHU servicing the stairs, StairsAHU, uses the same type of air handler in IDA ICE, except the mixing box is set to a minimum 50% supply of fresh air[6]. The air handler servicing the galleys, GalleyAHU, is a "standard air handling unit", but with the heat exchanger effectiveness set to zero.

Detailed design airflow rate data for each room of the ship are available in the "*Cabin Unit List"* and the "*Principle Duct Arrangement"* drawings. These were used in combination with the "*Test sheets AHU"*, which provide measured supply register airflow rates, to assign a measured supply airflow rate to each zone in the IDA ICE model and associate it with the appropriate IDA ICE ventilation plant.

IDA ICE allows the use of more than one AHU to supply ventilation air to a single zone. This feature was used in the model for this ship because of the merging of some of the zones. Regarding data entry, multiple AHUs serving the same zone requires careful manual data entry since the table data entry screen in IDA ICE has only one AHU listing for each zone.

[6] The return air recirculation for the stairs is given in the specifications as 40%, however the unit is also listed as having exhaust air that matches this value. Given this discrepancy, and its likely minimal impact on the results, it is entered into the IDA ICE model as having 50% return air recirculation.

The design supply airflow rates are usually higher than the design exhaust airflow rates because of the combination of using a cascade ventilation strategy, where fresh air is delivered to some zones and extracted from others, and the use of separate exhaust-only fans in certain areas of the ship. An example of cascade ventilation occurs in the cabins, where the drawings show that approximately half of the supply air to the cabins is exhausted from within the cabin's bathroom, while the remaining half is exhausted from the adjacent corridor.

In the model, a balanced ventilation approach was used where the exhaust airflow for each zone is assumed to be identical to the supply airflow for each zone. While this does not match exactly the design of the ship, it allows for simpler accounting of the ventilation air. This approach should suffice in giving accurate enough energy accounting of the ventilation system, but it can be refined later as needed.

For the cabins, this means all the air is both supplied and exhausted in the cabins, and the corridor is not used. For the large open space in the aft sections of decks 6 and 7, each deck gets its own balanced supply and exhaust, rather than supply air entering on deck 6 and exhausting from deck 7. These simplifications may need to be explored further in future work. Correcting the cabin-to-corridor or deck 6-to-deck 7 airflow distribution is unlikely to play a major role in the total energy use. However, other examples such as the air handler units serving the galleys, which cause significant negative pressures, have large-capacity heating coils, and have no heat recovery, may have more important consequences.

The numerous exhaust fans in the ship have not been directly included in the IDA ICE model. Many of these are used to exhaust air from various equipment rooms, so they are not expected to be important to the hotel system. In addition, given the approach described above to set the ventilation system exhaust flow rates equal to the supply flow rates, many of these exhaust fans are expected to be accounted for. However, this is an area that will likely need further development in the model:

- One reason is that unrealistic temperature or air quality results can be expected in areas receiving no ventilation in the model. This could be solved by implementing a cascade ventilation system.
- Another reason is that in reality the ventilation systems with heat recovery do not have all the exhaust air available to exchange heat with. As currently modeled, this will result in over-estimating heat recovery rates. This can be adjusted either through addition of exhaust-only systems and associated re-accounting of exhaust airflow rates, or through downward adjustment of the heat recovery effectiveness based on the limited availability of exhaust airflow.

The technical specification document for the ship, "*181127_Technical Specification (Teknotherm) - R01*", gives supply air temperatures for the galleys for design summer and winter conditions. In summer, the galley supply air temperature is to be 10°C lower than the outside maximum temperature, while in winter the supply air temperature is to be 20°C. Given that the outside temperature rarely exceeds 30°C, if ever, the galley supply air temperature is set to a constant 20°C in the IDA ICE model.

This assumption was checked versus the operational data (Section 4.3). However, the two galley air handler units' operational data are only available for a four-month summer and fall period in 2022. They show supply temperatures of 18°C and 12°C during these periods, as shown in Figure 48, but an additional duct heater heating coil installed downstream of each of the galley air handlers adds uncertainty in the usefulness of the supply air temperature data for the purpose of the model's supply air setting. The heating of the supply air to the galley, presumably through the use of the duct heaters, appears to be controlled based on a temperature sensor in the space, which the data show average 19.8°C for the galley on deck 7 and 24.6°C for the galley on deck 8. Given these values and the uncertainty surrounding the specifics of the controls, the original assumption of a 20°C galley supply air temperature remains in the model. However, this remains an area that could be trued up in the model to align with actual conditions, with a better understanding of the control and operation of the galley ventilation air and a longer-term data collection effort covering more months of the year.

For other room types, temperature of the supply air is assumed constant at 16°C. While operational data show a changing supply air setpoint over the course of the year for one air handler, in Figure 47, and the four-month average air handler supply air temperature for other air handlers are available as shown in Figure 48, there is not enough information to establish a reliable logic to represent the supply air temperature settings for the Color Hybrid in the model. Therefore, the model uses the simple assumption of a constant supply air temperature of 16°C. With this, it is important to remember this setting as an important caveat when reviewing the results. The supply air temperature setting will affect the balance between the winter-time energy use of the zonal heating systems and the ventilation system, and the balance between the summer-time energy use of the zonal cooling systems and the ventilation system. Therefore, adjustment of this setting will be a useful calibration tool during a verification exercise where both actual energy consumption of the heating, cooling, and ventilation systems and the actual ventilation system temperature settings are available for the same period.

The galley ventilation systems are assumed to be operated at 50% capacity during employee occupied hours (see 5.3.2 Occupancy) and 10% capacity otherwise. This is an important assumption in the model because the air handler units serving the galleys have no heat recovery. The operational data (Section 4.3) suggest the galley fans run at higher levels than this, so this setting is discussed in more detail in terms of the baseline and efficient-cases of the various efficiency measures in Section 0, and it is specifically relevant in the galley ventilation rate measure in Section 8.1, where the assumption here is used as the efficient-case. When calibrating the model to thermal energy use data, as suggested as future work in Section 0, the actual operation of the galley ventilation system should be closely understood since it can account for over half of the heating energy demand of the air handlers, even with the more conservative 50% day/10% night operation assumed in the adjusted model (See Table 19).

The CAV air handler units are assumed in the model to operate all the time. The VAV air handler units are identified as using CO_2 occupancy sensors to adjust the flow rate. This is the control approach selected in the model. The occupancy schedules therefore drive the flow rates to the zones using VAV. Reduction of the ventilation flow rates for both types of air handler is discussed in more detail in relation to the proposed efficiency measure in Section 8.3.

The car decks have large reversible supply and exhaust fans that are used for ventilation of the vehicle exhaust. The document "*Ventilation system for Cargo Hold: Cargo Decks 3, 4, & 5"* explains the operation scheme of the fans as providing 20 air changes per hour during loading and unloading and 10 air changes per hour while the ship is sailing. The documents *"Technical Specifications"* and "*Car deck fans specification*" provide the volumetric flow rate, static pressure, and power consumption rating of the 8 fans. Each of the four larger fans have an airflow rating of 141,050 m^3/hr and a power consumption rating of 39 kW, while the remaining four smaller fans have an airflow of 64,000 m^3/hr and a power consumption of 19 kW, both at a static pressure of 450 Pa. These fans are oversized by a factor of 2, meaning that during the full demand of 20 air changes per hour, the fan speed is reduced to provide only half of the rated airflow shown here. Although the ventilation system in the car decks does not resemble a typical air handler unit, it was added to the IDA ICE model as a standard air handler unit, with half the total rated flow shown here, i.e., 411,000 m^3/hr. The as-operated pressure rise is not known, so it is assumed to be 450 Pa and the efficiency is assumed to be 60%, with a VAV part-load performance using the "ASHRAE standard 90.1 (2007) appendix G" default in IDA ICE. The schedule for the car deck ventilation assumes 10% rated airflow at night, 100% airflow during expected loading times, and 50% airflow during sailing.

5.4.5 Domestic Hot Water

The ship's domestic hot water (DHW) system consists of two 2,000-liter calorifiers that together receive up to 280 kW of heat through a heat exchanger tied to the main heated water system. The primary side of the heat exchanger is designed for 75°C supply and 60°C return water, while the secondary side is designed to heat the domestic hot water from 10°C to 70°C.

The ship is assumed to use 30.1 kWh/m^2yr for heating domestic hot water[7]. This value aligns with the given energy use of hot water for hotels in the Norwegian Standard for "Energy Performance of Buildings: Calculation of Energy Needs and Energy Supply", SN-NSPEK 3031:2021 [16]. Hotels are the building type in the standard with the highest domestic hot water energy use per day. While the majority of the ship does not operate similar to a hotel, the assumption used here is a starting point. The highest value in the standard was chosen to represent the estimated energy use in the Color Hybrid because it is expected that many of the occupants will eat a meal at one of the ship's restaurants during each of the four 2.5-hour journeys and the hot water associated with the restaurants is expected to be high.

This assumption was originally used in modeling of the Color Fantasy, but later increased to 58.5 kWh/m^2yr after calibration of the model [1]. The Color Fantasy has both overnight guests with shower facilities and a swimming pool, so its domestic hot water energy demand could be expected to be higher than the Color Hybrid's.

The load duration curve of the domestic hot water system is assumed to have full power between 9:30 am and 10:30 pm, zero power from 11:00 pm to 9:00 am, and a linear rise and fall period of a half hour each. This, along with the annual energy use of 30.1 kWh/m^2yr and the ship's occupied floor area of 18,735 m^2, results in an implied required water heating capacity of 114.4 kW. Given the simplified assumptions used here, this corresponds well with the as-installed 280-kW heat exchanger used as the ship's DHW heater. While these assumptions are not adequate to represent the water heater's instantaneous power requirements, it should suffice in providing a realistic energy use down to a daily time interval level.

5.5 Climate

The weather file used is historical normal weather, in the form of ASHRAE IWEC2, from a lighthouse off the coast of Sandefjord, called Færder fyr. Its location is about 1/3 of the way from Sandefjord to Strömstad, making it a good fit for the weather the ship is likely to encounter on its journey. However, the ship spends much of its time in port, and the orientation of the ship changes in each direction of travel and in each port. In the current model, the ship faces toward the east. These

[7] Note that the value for hot water consumption in the "Extra energy and losses" form in IDA ICE is input as 23.15 kWh/m^2yr. This is because that form assumes a temperature rise of 50 K, whereas the ship's domestic hot water delivery temperature is set to 70°C, resulting in a temperature rise of 65 K with an assumed incoming temperature of 5°C. This adjustment results in the model providing the correct amount of annual energy use for domestic hot water – 30.1 kWh/m^2yr – while still accounting for the higher hot water temperature.

issues should be accounted for in the development of a more accurate weather file and a solution for ensuring the correct ship orientation during the various operating modes.

During the model validation and calibration step, actual weather for the period aligning with the energy use was used instead of historical normal weather. See Section 4.3 for more details on the weather data used to align with operational and energy use data, and Sections 0 and 7.1 for use of the data in the data analysis and model validation, respectively.

6 Data Analysis Results

In this section, data collected from the ship are compared to their modeled counterparts. The comparison data are used to represent the actual conditions with which the model should be judged and adjusted if necessary. These comparison data are either in the form of design data, or in the form of measured energy use or temperature data during ship operation. The comparison data and model output data are analyzed. This is followed by a discussion of how well the model results align with the comparison data and whether model adjustments are advised.

6.1 Delivered Energy

In this section, measured electrical shore power data and diesel energy use data are explored in-depth to allow calculation of the expected annual delivered energy consumption under "normal" operating conditions. Together, these two energy sources serve both the hotel system and the propulsion system.

Volumetric diesel consumption data were made available for the period between January 1, 2021, and August 28, 2022. The raw data were most often provided as liters of diesel consumed per crossing, but sometimes the data had been aggregated into longer time periods, for example liters of diesel consumed per day. A visual representation of the raw diesel consumption data is provided in Figure 19.

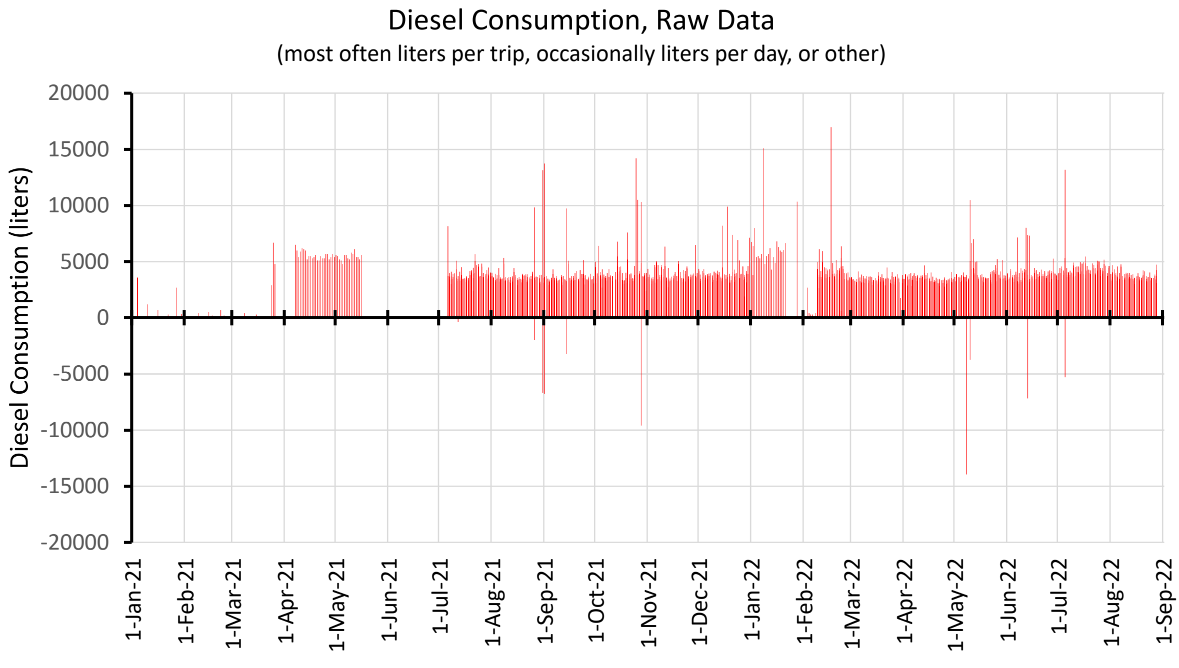

Figure 19 – Diesel Consumption, Raw Data

The diesel consumption data were further investigated to understand the suspected irregularities, such as the large negative values in Figure 19, for example at the end of August and November 2021 and the beginning of May 2022. These large negative values are often preceded or followed by a large positive value. While the exact reason is unknown, the data appear to receive occasional manual accounting adjustments, which, along with the occasional shift in granularity, suggests the data are likely manually entered. For the purposes of their use as a "truth-set", or a training data set for which to calibrate the IDA ICE model, these anomalies appear to be mostly rectified when summing all values to the daily levels, and completely rectified when summing to the weekly levels. Further comparison with the electric consumption data in periods of regular ship operation, as discussed below, suggests the diesel consumption data are sufficiently reliable to be used for model calibration.

Electrical energy consumption data from the shore power connection at the dock in Sandefjord were also provided for the same period as the diesel consumption data – between January 1, 2021, and August 28, 2022. The raw data were provided in daily intervals. Given their regularity, the data appear to be machine generated. A downward adjustment of just under 600 kWh/day was made to the raw daily electrical energy consumption values, because the Color Viking, which sails the same route as the Color Hybrid, also uses the shore power during its two daily stops in Sandefjord. The exact distribution of electrical energy to the two ships is unknown, but Color Line estimates under normal operation periods, the Color Viking uses approximately 18,000 kWh/month of shore power [19]. The estimated daily shore power consumption (purple line, secondary y-axis) and daily diesel consumption (red line, primary y-axis) for the Color Hybrid are shown in Figure 20. Here, the diesel is assumed to have an energy content of 10.7 kWh/liter. Also shown are the estimated periods of irregular operation caused by COVID-19 pandemic restrictions (grey shading) and regular ship operation (green shading).

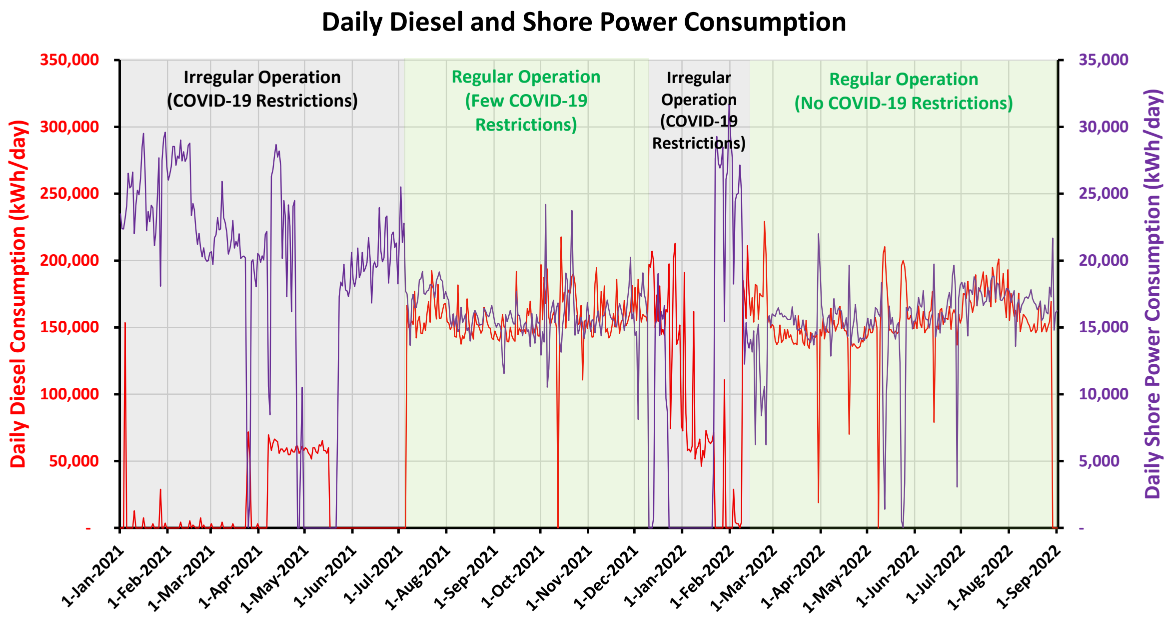

Figure 20 – Daily Diesel and Shore Power Consumption

For the first period of irregular operation, from January 1 to early July 2021, the ship often uses between 18,000 and 30,000 kWh/day of shore-power, with the exception of the early and middle parts of May and a few days of low use in March and April. The diesel use during this period is mostly zero, with a noticeable period of daily consumption just over 50,000 kWh/day in April and May. Since this period occurred during heavy travel restrictions of the corona pandemic, the ship was likely docked in Sandefjord and unused for most, if not all, of the period. Therefore, this period was ignored in the calibration analysis.

For the second period identified as having irregular operation, from early December 2021 to mid-February 2022, the irregular use patterns of both diesel and electricity clearly confirm limited operation of the ship. However, the pattern also suggests some operation occurred during this period. Corona pandemic restrictions were increased in early December, but the extent to which this impacted the operation of the ship is not clear. Given the uncertainty of the ship's operational schedule during this time, and the likelihood of irregular operation, this period was also ignored in the calibration analysis.

The two periods identified in Figure 20 as having regular operation show a consistent pattern, with the daily diesel and electricity consumptions remaining fairly consistent over time and most often aligning well with one another. While the first regular operation period between early July and early December 2021 had several known corona restrictions, the data suggest similar operation to the second regular operation period, which is known to have very limited, if any, pandemic-related restrictions. Therefore, both these periods will be investigated further for use in the model calibration.

Figure 21 shows the same data as shown in Figure 20, but with just the first period of usable data.

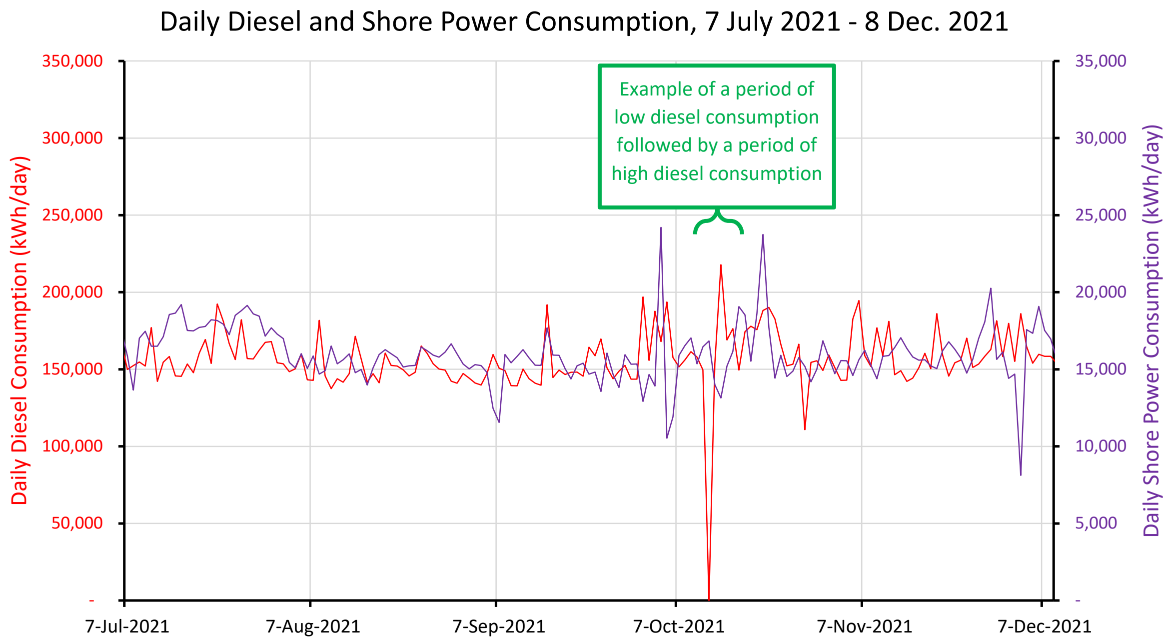

Figure 21 – Daily Diesel and Shore Power Consumption, First Usable Period

Similarly, Figure 22 shows a closer look at just the second period of usable data.

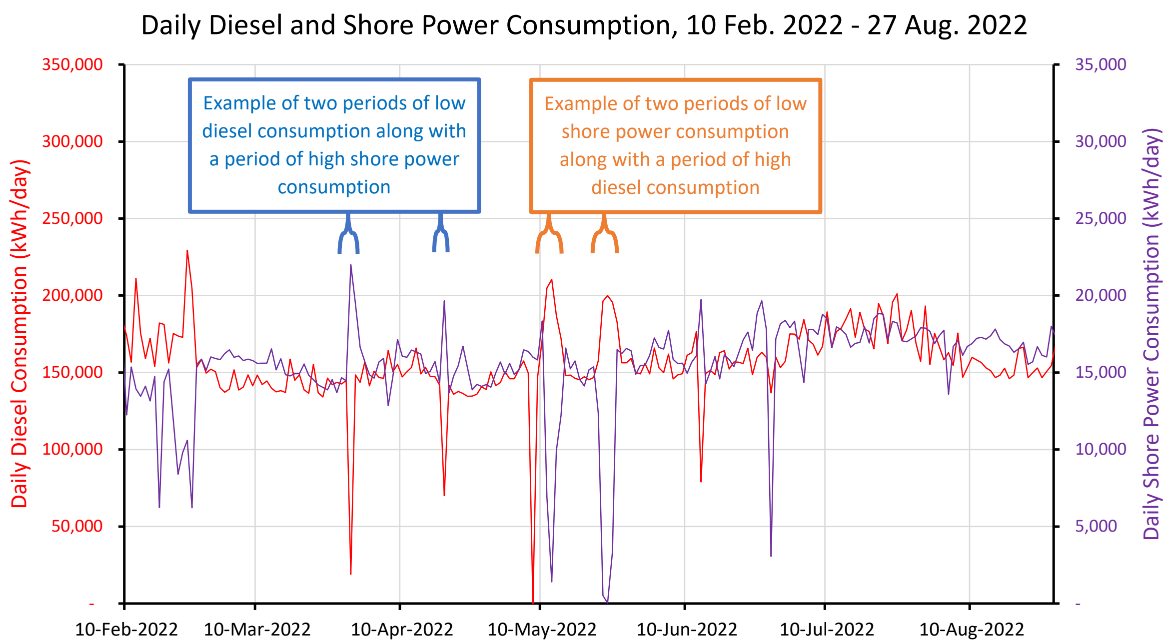

Figure 22 – Daily Diesel and Shore Power Consumption, Second Usable Period

There are three types of noticeable anomalies in these time periods that deserve discussion. The first type of anomaly is a period of low diesel consumption without a noticeable corresponding deviation in the electricity consumption, but with a subsequent upward deviation in diesel consumption shortly afterward. An example of this is pointed out by the green box in Figure 21. This type of anomaly is likely due to the apparent accounting adjustment described above, in relation to Figure 19. Therefore, they are not expected to reflect the actual fuel usage of the ship and efforts to ensure they do not impact the calibration will need to be made.

The second and third type of anomaly during the regular operation periods is a low daily consumption of either diesel or electricity, along with a simultaneous high consumption of the opposite energy source. Examples of both of these are highlighted by the blue and orange boxes in Figure 22. These two types of anomalies appear to represent the actual overall fuel usage of the ship, where a certain change in ship operation is affecting the otherwise-expected consumption in a way that causes a shift from using one type of energy source to using the other type of energy source. These times of tradeoff of energy sources can be used to develop a better understanding of the data and the ship operational characteristics.

A reasonable explanation for the case of lower diesel consumption and higher electricity consumption (blue box in Figure 22) is that the ship did not travel, so it did not use diesel for propulsion. In addition, the ship could not take advantage of waste heat or electricity generation from the diesel engines. Extra shore power was therefore used to cover heating and electrical needs while in port. The issue with using these data to develop a better understanding of the ship operation, however, is that, in essence, the ship did not operate. Therefore, these data will not be used. In addition, given that there is storage of energy on the ship in many forms (electric batteries, hot water storage tanks, and diesel), it is difficult to be sure of the underlying cause of the reduction in diesel energy use, and whether these causes actually occurred simultaneously with the increase in electricity consumption.

An explanation for the case of higher diesel consumption and lower electricity consumption (orange box in Figure 22) is that the ship used the diesel generators to replace shore power. While the same caveats regarding energy storage apply here, it does appear the ship was in operation. Evidence of the merits of this explanation can be shown by a rough calculation of the thermal energy efficiency of the diesel generators. Here, thermal energy efficiency is calculated by dividing the deviation in average electrical energy use per day (that is, the amount of shore power replaced with power produced by the diesel generators) by the deviation in average diesel energy use per day (that is, the extra diesel used to generate the electricity). These deviations are estimated by subtracting the consumption during the event from the average consumption for the three days before and after the event. The resulting estimated thermal efficiency is 19% for the first “orange” event and 29% for the second “orange” event. Given the uncertainty in this calculation method, these values can be considered roughly the same as the approximate 30% efficiency that would be expected for a diesel generator. Thus, this analysis provides evidence that the data and ship operation are relatively well-understood.

The electrical energy consumption data discussed above applies only to the shore power connection in Sandefjord, where burning of diesel fuel is not normally allowed. It would seem that a more detailed look at these data and the hotel needs during docking hours can perhaps provide a method to understand the energy use of the electric components of the hotel system. The following complicate this analysis, however:

- use of the batteries to supply propulsion energy for sailing and positioning to and from the dock in Sandefjord and the use of shore power to charge these batteries;
- storage of thermal energy on the trip to Sandefjord and subsequent use of this thermal energy to supply heat while docked in Sandefjord; and
- unknown use of electric heaters while docked to supplement the hotel system heating needs.

For these reasons, additional analysis using the shore power data will not be undertaken to understand the hotel electricity consumption. Luckily, additional electrical energy use data from the ship’s own monitoring system have been made available to the project. This is discussed in detail in Section 6.2.

Knowledge of the total delivered energy use over the course of a year allows the consumption of this ship to be compared to other ships. The data from normal operation, shown in Figure 21 and Figure 22, can be used to estimate the total delivered energy consumption for a year. Figure 23 shows the available delivered energy consumption data from both periods averaged by month. Average daily shore power energy use is shown in purple, and average daily diesel energy use is shown in red. Also shown, in green and on the secondary axis, is the number of days of usable data represented by each month.

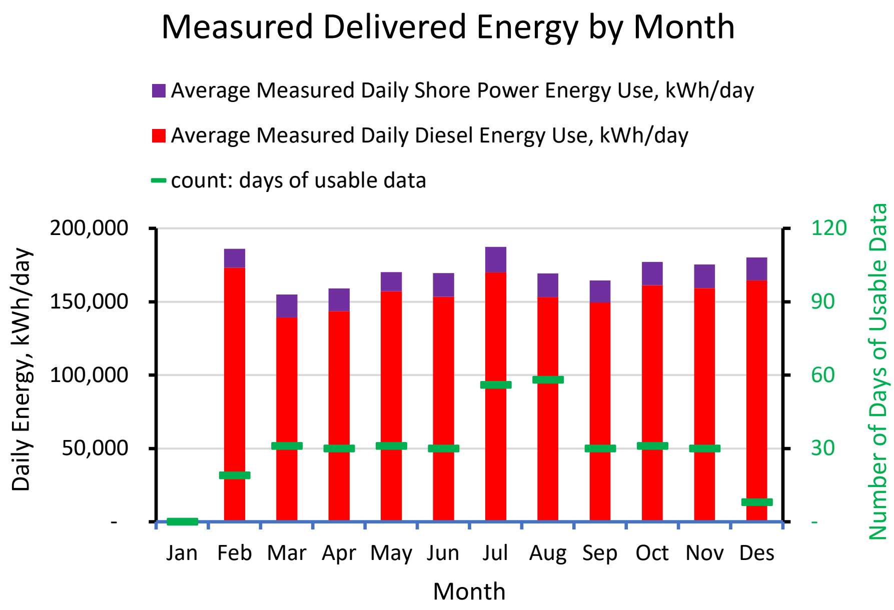

Figure 23 – Measured Daily Delivered Energy by Month, Available Data

It is clear that no usable data are available in January and there are limited data available in February and December. It appears that there is a slight increase in delivered energy in the summer months, and perhaps the winter months, suggesting the delivered energy use could be dependent on the weather. However, additional investigation should be made to confirm or deny weather dependency.

A scatter plot of the daily diesel consumption versus daily average outdoor temperature (ODT) is shown in Figure 24. See Section 4.3 for more details on the weather data. Each red point represents a day of usable data. A blue dashed line and associated equation and coefficient of determination show the results of a linear regression of the data.

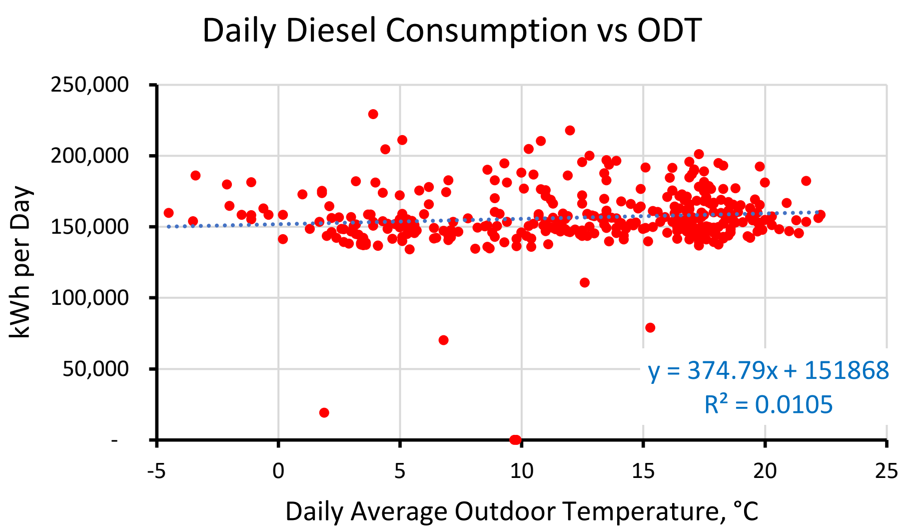

Figure 24 – Measured Daily Diesel Consumption versus Daily Average Outdoor Temperature

The scatter plot in Figure 24 does not appear to show a relation between daily average outdoor temperature and daily diesel consumption. The low coefficient of determination, or R^2 value, of 0.01 confirms this lack of relationship. Since the ship is designed to take advantage of waste heat to cover both winter heating and summer cooling loads during normal operation, it is logical that the ship's diesel consumption is independent of outdoor temperature.

A similar scatter plot of the daily shore power consumption versus daily average outdoor temperature is shown in Figure 25. Each purple point represents a day of usable data. The result of a linear regression is shown as a blue dashed line and associated equation and coefficient of determination.

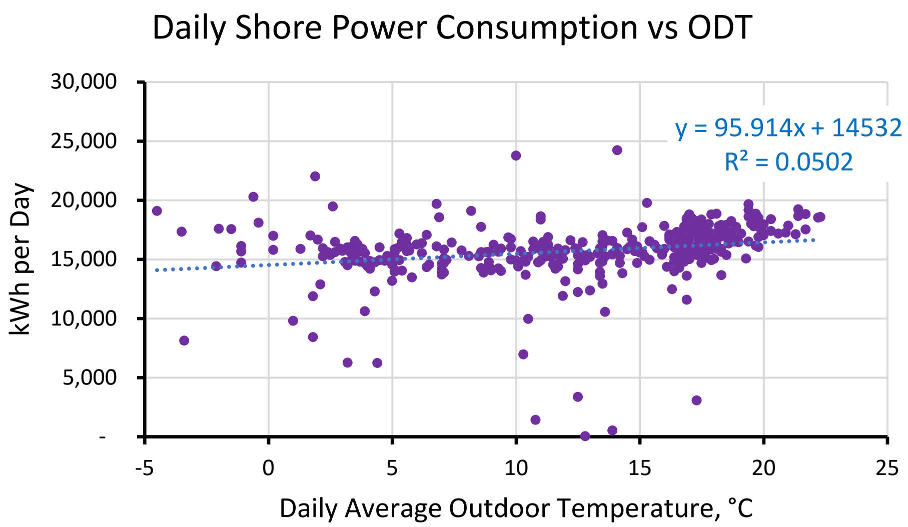

Figure 25 – Measured Daily Shore Power Consumption versus Daily Average Outdoor Temperature

A low coefficient of determination of 0.05 indicates that the linear regression calculated here is not well-aligned with these data. In the case of shore power, it is reasonable to expect the ship could use more electricity to provide weather-dependent cooling while docked during the summer and perhaps to provide some weather-dependent electric heating while docked in the winter. In fact, if the regression is ignored, the scatter plot in Figure 25 appears to show a weak relationship between daily average outdoor temperature and daily shore power consumption, with an apparent rise in daily shore power with a rise in temperature, starting at outdoor temperatures above about 10°C to 15°C. In addition, there appears to be a slight increase in shore power consumption with a fall in temperature, starting at outdoor temperatures below about 5°C. These relationships could be explored further with two separate regressions for the lower and higher outdoor temperatures. However, the charging of the batteries using shore power and discharging of the thermal storage tanks while docked in Sandefjord complicates the analysis. In addition, shore power makes up only about 10% of the total delivered energy. Therefore, a search for a more detailed relationship between shore power and outdoor temperature will not be undertaken here.

As discussed above, the daily diesel consumption outweighs the daily shore power consumption by a factor of ten, so the sum of the two show no relationship to daily outdoor temperature.

Historical passenger count data were provided by Color Line. The data consist of the number of passengers for each sailing between February 9 and October 16, 2022. Figure 26 shows the daily diesel consumption per day versus the average number of passengers per trip. The metric "average passengers per trip" was used instead of simply passengers per day to allow easier comparison to the ship's passenger carrying capacity. There are normally four trips per day, but occasionally there are only two trips in a day. The data underwent a minor data cleaning to remove data that were not useful to this analysis. In total, 199 days of data were originally available. The following days were removed:

- Four days with only two trips,
- One day with no recorded diesel energy consumption, and
- One day with no recorded passengers.

Each red triangle represents one day of data where four trips were made. A blue dashed line and associated equation and coefficient of determination show the results of a linear regression.

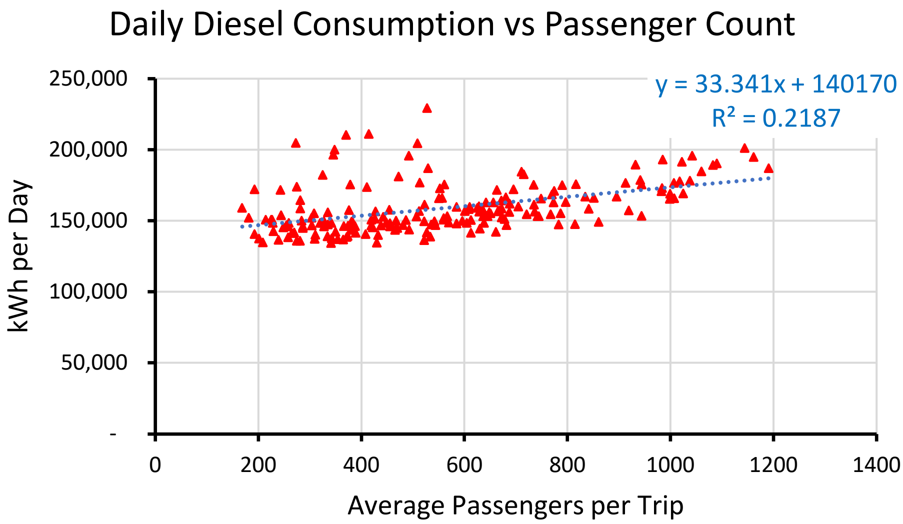

Figure 26 – Measured Daily Diesel Consumption versus Passenger Count

In contrast to the outdoor temperature, the passenger count shows a clear relationship with daily diesel consumption. The slope of the regression in Figure 26 indicates that the daily diesel consumption increases by 33.3 kWh for each increase in average passenger count per trip. Since four trips were made each day in this dataset, this equates to approximately 8.3 kWh for each additional passenger making one trip.

Figure 27 shows the daily shore power consumption versus average passengers per trip. The same data cleaning used on the data in Figure 26 was made to the data in Figure 27.

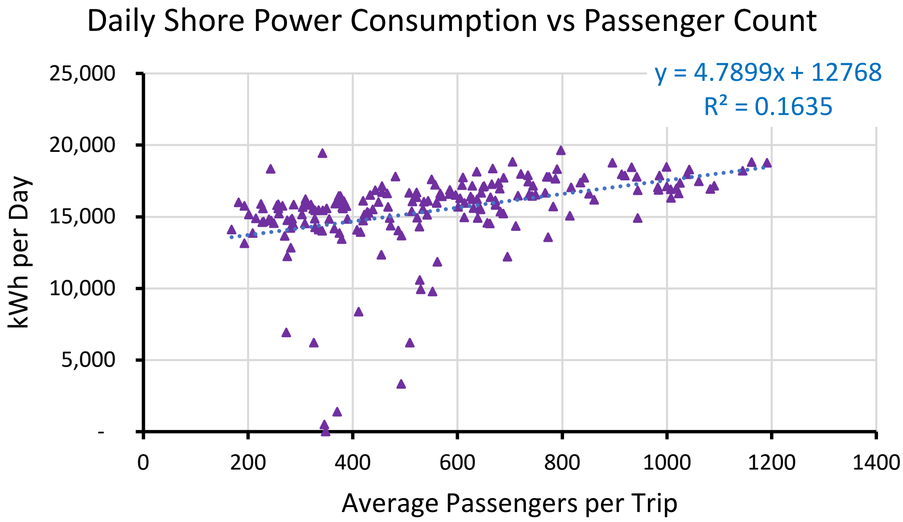

Figure 27 – Measured Daily Shore Power Consumption versus Passenger Count

A weak, but apparent relationship between daily shore power consumption and passenger count exists. The slope of the regression in Figure 27 indicates that the daily shore power consumption increases by approximately 4.8 kWh for each increase in average passenger count per trip, or 2.2 kWh for each additional passenger making one trip.

A scatter plot of the measured total delivered energy versus passenger count is shown in Figure 28. Each black triangle is simply the sum of the daily diesel consumption (Figure 26) and daily shore power consumption (Figure 27).

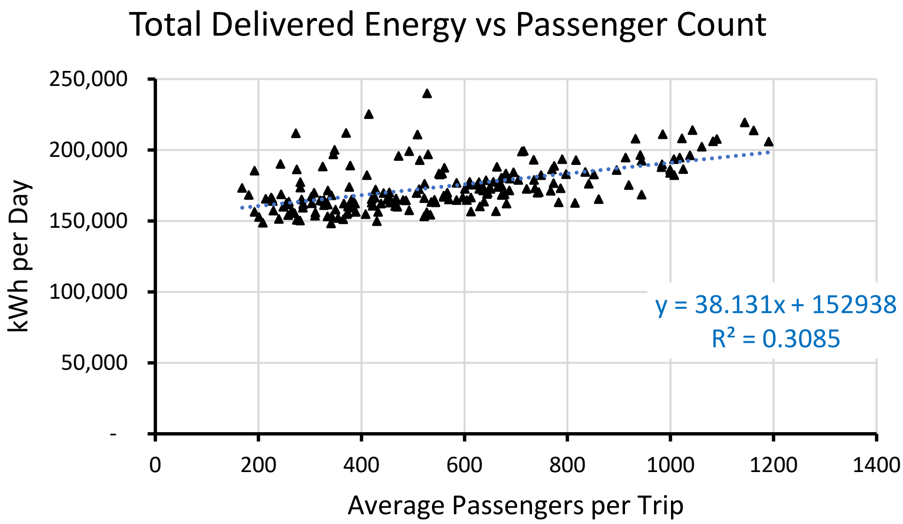

Figure 28 – Measured Daily Total Delivered Energy versus Passenger Count

The total delivered energy has the strongest linear relationship to passenger count. The slope of the regression in Figure 28 indicates that the daily total delivered energy increases by approximately 38.1 kWh for each increase in average passenger count per trip, or 9.5 kWh for each additional passenger making one trip.

From these data, it is not clear what loads are causing passenger count to influence delivered energy. Possible explanations include:

- Higher ventilation rates for the demand control ventilation areas of the ship;
- Higher restaurant energy requirements to prepare more food and wash more dishes;
- Higher plug loads, hot water, and other miscellaneous passenger-specific loads; and
- Higher propulsion loads caused by the weight of the additional passengers and their vehicles.

This analysis confirms which factors should be considered when estimating the ship's delivered energy for a year. Outdoor temperature does not need to be taken into consideration in an annual extrapolation, while passenger count should be.

The average number of passengers per month is shown in Figure 29. The solid blue columns represent the average number of passengers per trip based on the data provided. The green horizontal lines represent the number of days of data available for each month, shown on the secondary y-axis. The columns with blue stripes represent the estimated average number of passengers per trip. These estimates, which apply to the months of January, November, and December, are based on the average number of passengers counted in February and October.

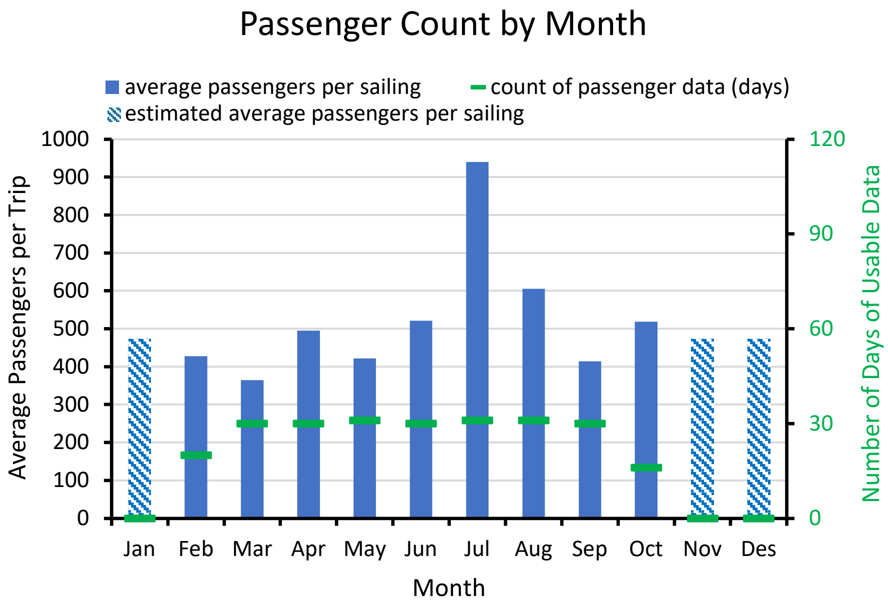

Figure 29 – Passenger Count Data

The passenger count data shown in Figure 29 were used with the regression equations in Figure 26 and Figure 27 to estimate the daily diesel and shore power consumption, respectively, for each month. This represents a method to estimate the energy use over the course of an average year, assuming the passenger counts by month shown in Figure 29 are representative of the average. The results of this analysis are shown in Figure 30. The red and purple columns present the average measured daily diesel and shore power consumption, respectively – note that this is the same data as shown in Figure 23. The striped-red and striped-purple columns represent the estimated daily diesel and shore power consumption, respectively.

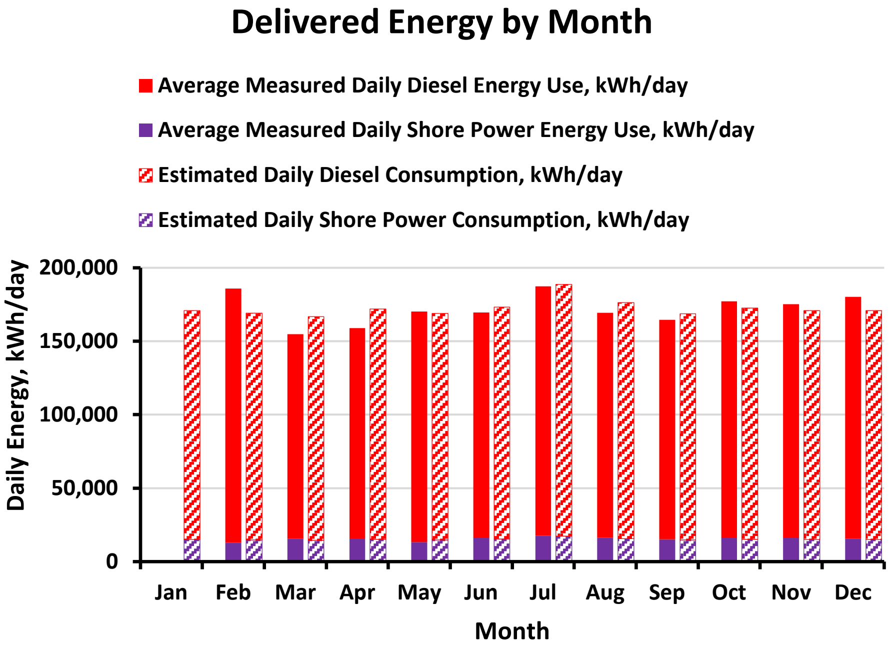

Figure 30 – Estimated and Measured Delivered Energy by Month

As seen in Figure 30, the largest difference between the measured and estimated daily delivered energy occurs in the month of February. A closer look at the data in February in Figure 22 reveals less shore power energy use in favor of higher diesel energy use. This is expected to cause overall higher delivered energy because of the lower efficiency associated with diesel than electricity. Since there is no reason to expect this type of operation as "normal", the proposed method of estimating shore power and diesel consumption by using the regressions in Figure 26 and Figure 27, along with expected monthly passenger numbers, is considered reasonable.

The annual expected delivered energy, including the breakdown between shore power and diesel consumption, is shown in Table 12. This is a "normal" value, which means it is intended to represent the ship delivered energy use under normal operating conditions.

Table 12 – Estimated Normal Annual Delivered Energy

Delivered Energy Source	Annual Energy Use, MWh/yr
Diesel	57,399
Shore Power	5,556
Total	62,956

The values in Table 12 are adjusted based on the expected "normal" passenger loading presented in Figure 29, so they represent the sum of the estimated values in Figure 30. The analysis above shows that outdoor temperature did not have an impact on delivered energy, but that passenger counts had an effect. There is an important caveat to this analysis: It is unknown whether the proposed passenger loading will apply to future operation, or whether outdoor temperature will continue to not impact delivered energy consumption. Future data collection efforts for the ship should check for these and the possibility of other factors' influence on delivered energy. This type of data normalization process should be made prior to comparing new data sources to the values shown in Table 12.

The model currently attempts to account only for the hotel energy use and does not account for propulsion energy. Therefore, comparison of the model results to those in Table 12 are of limited value since the propulsion energy is expected to make up more than half of the overall energy use. Instead, the values shown here are useful for comparing the delivered energy requirements of various ships.

6.2 Hotel Electricity

In this section, the measured electricity consumption of the hotel system is explored in detail. The electricity consumption of the hotel system is explicitly included in the model, so detailed analysis and understanding of the measured consumption lends itself well to calibration of the model.

Raw data representing the electrical energy use of the hotel system, as supplied through the ship's two main electric switchboards, were provided by Color Line in irregular intervals of approximately 2 seconds. In addition, electric power flows to and from the battery systems were provided in the same format. The data cover the time period between October 13, 2021, and August 28, 2022. MATLAB was used to process and aggregate the data to regular 1-minute intervals.

Figure 31 shows the data for April 4, 2022. This day was chosen mostly at random. Viewing the data over the course of a day allows for better understanding than attempting to visualize all the data at once. The electric power to the hotel is shown by the red line. The electric power for the batteries is shown by the green line. Net electric power flowing to the batteries is represented by negative electric power values, while net electric power flowing from the batteries is represented by positive electric power values. The ship schedule is indicated in shaded boxes. These times align with those described in Table 2.

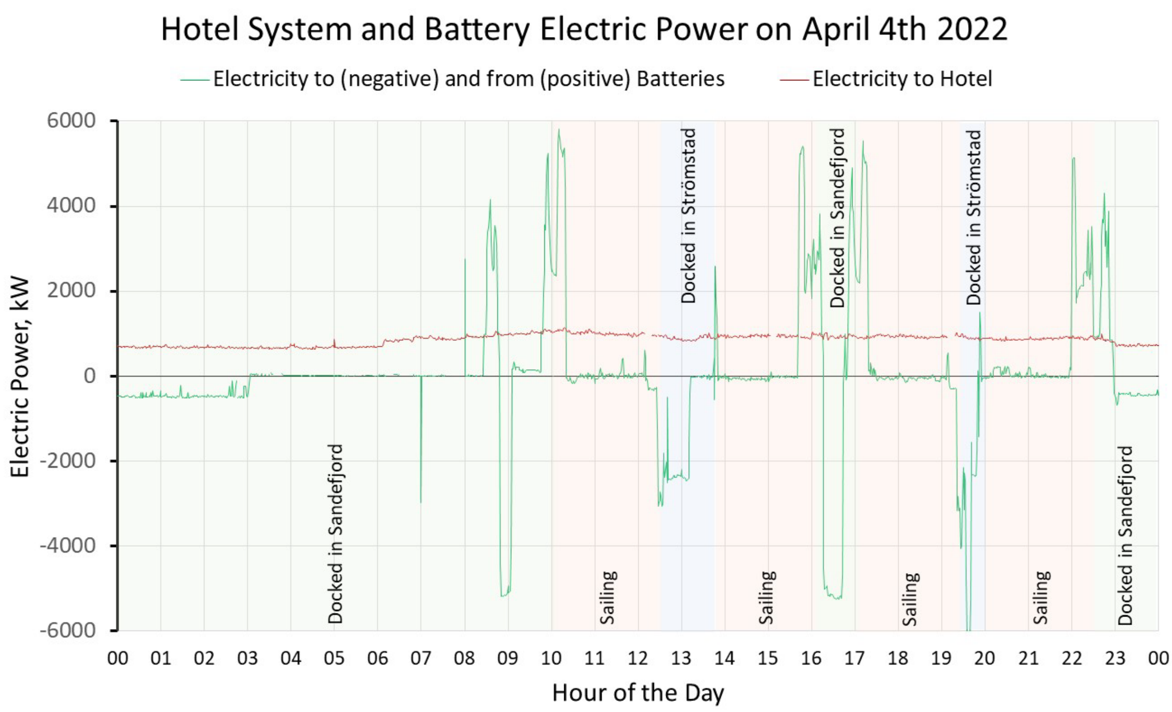

Figure 31 – Hotel System and Battery Power Data for One Day

Depending on the operating conditions of the ship, the electricity provided by the batteries can be used to power the hotel, the ship's propulsion, or both. The battery storage appears to operate as expected. The following description of the battery data shown in Figure 31 is based on an understanding of the available design data and operation philosophy documents provided by Color Hybrid:

1. Overnight slow charging occurs while the ship is docked in Sandefjord;
2. High electricity use from the batteries at around 8:30 appears to be providing electrical propulsion energy to re-position the ship to the car-loading dock;[8]
3. High-speed charging occurs shortly after the ship is re-docked and connected to shore power at just before 9:00;
4. After the ship is loaded, it uses battery energy to sail through Sandefjord at around 10:00;
5. While docked in Strömstad, the diesel generators provide battery charging at around 12:30 to 13:00;
6. When the ship arrives in Sandefjord, it uses battery energy for propulsion, receives charging while docked, and uses battery energy for propulsion again upon departure – all occurring between about 15:30 through 17:30;
7. Another charging through diesel generators occurs while docked in Strömstad around 19:30; and
8. One final round of electrical propulsion energy upon arrival in Sandefjord occurs at around 22:00 before starting the cycle again for the night and the next day.

Since the focus of this study is on the hotel system's energy use and efficiency opportunities, the battery storage system and associated electricity flow data will not be covered in more detail. The battery consumption data are shown above in the event that future analysis of the electricity storage system is desired.

Investigation into the hotel electricity use data allows for a better understanding of the real operation of the ship, thereby providing useful information for model calibration. The electricity to the hotel is independent of the electricity flowing to and from the batteries. Figure 31 shows a fairly constant hotel electricity demand of about 700 to 1000 kW on April 4, 2022. This will be explored further for all days with usable data. Figure 31 also shows an issue that requires an additional data cleaning step: There is a discontinuity in the electric power demand (red line) at just after 12:00 and just after 19:00.

In order to allow manipulation of the data in Excel, and to fix the data continuity issue, the data were aggregated to 15-minute intervals. Since the hotel electricity demand is fairly stable, if the MATLAB-processed minutely data provided at least one useable value within a 15-minute period, then the average of the minutely data was used. If there were no usable minutely data within a 15-minute period, the aggregated value for that period was left blank. In total, within the period of available data, October 13, 2021, through August 28, 2022, 2.7% of the minutely data are blank, and this results in 0.4% of the 15-minute data having blanks. This low level of missing data is very unlikely to have a meaningful impact on future use of these data. The missing data in future analysis is overcome by either ignoring or assuming average values for the blanks.

A time-series graph of all of the available 15-minute average hotel electricity demand data is shown in Figure 32. The start of each month is labeled on the x-axis. The shaded area represents irregular operation during the COVID-19 pandemic (Labeled "Corona Restrictions" in the graph).

[8] It should be noted that at the time of writing this report, there is a limited understanding of the charging and discharging of the battery at the dock in Sandefjord, so these scenarios are assumed.

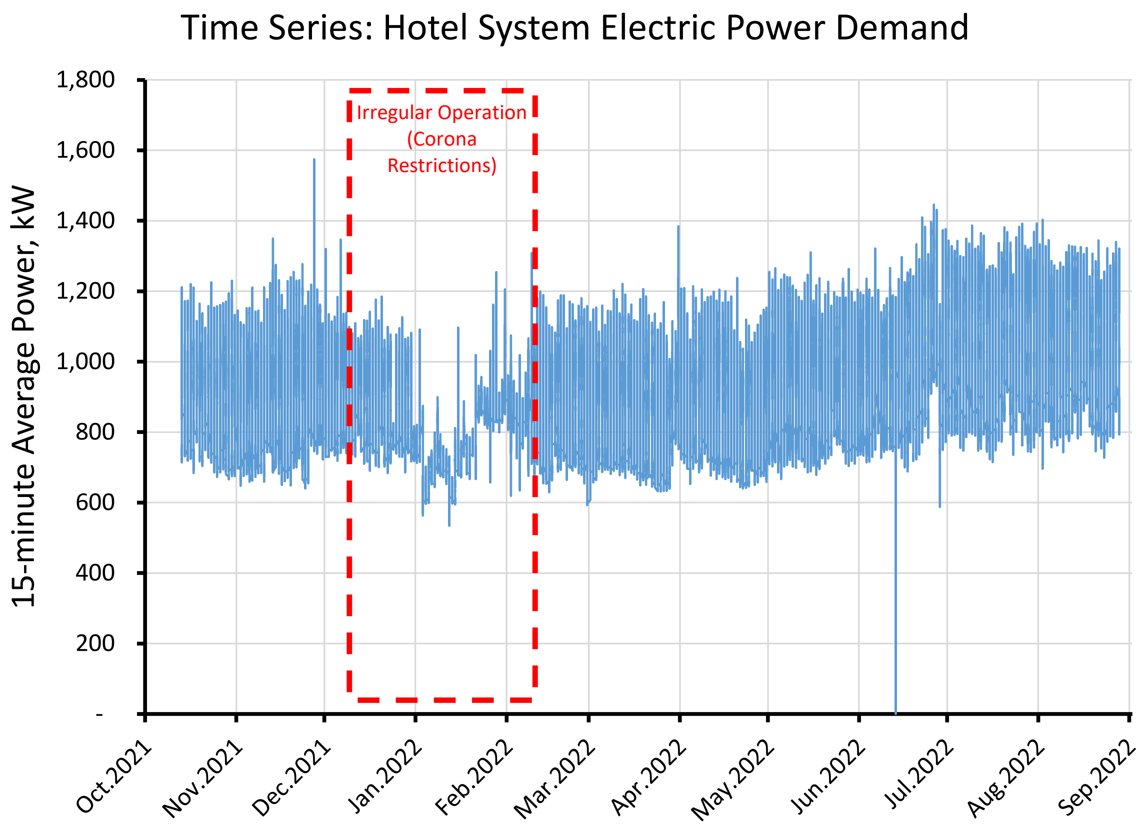

Figure 32 – 15-minute Average Hotel System Electric Power Demand, All Available Data

Figure 32 shows a fluctuating range of electricity consumption over the time shown, with the high end of the range occurring most often at, or slightly less than 1200 kW. The exception is that during the end of the period, from around mid- to late-June 2022, the high end of the range rose by about 100 kW. The minimum end of the power draw range is between 600 and 800 kW. Actual instantaneous power demand peaks and troughs are higher and lower than shown by these 15-minute averages and can be further explored through analysis of the 1-minute or raw data, if desired.

To allow a better understanding of the data shown in Figure 32, one week of data is shown in Figure 33. The start of each day of the week is shown on the x-axis. The date range selected here begins with April 4, 2022, to allow comparison to the data shown in Figure 31.

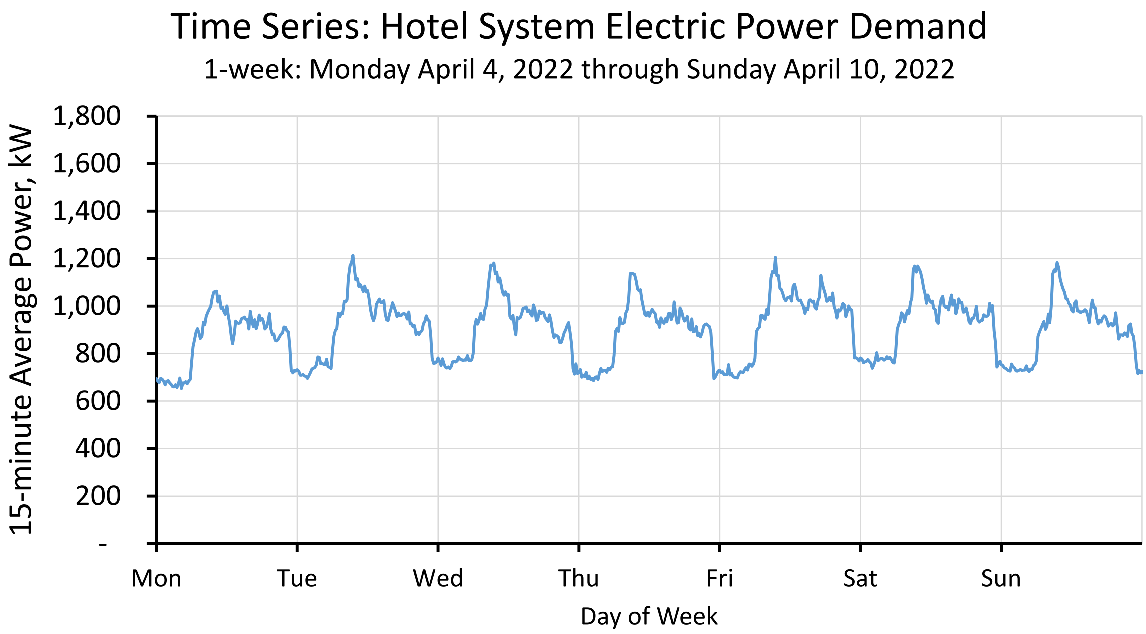

Figure 33 – 15-minute Average Hotel System Electric Power Demand, 1 week of Data

This closer look at the data provides insight into the electric power demand of the hotel system. Figure 33 confirms an expected diurnal pattern, where electric power demand increases during daily operation and falls at night when there are no passengers. Interestingly, there does not appear to be a difference between weekend and weekday – Saturday and Sunday's electric demand patterns appear similar to Monday through Friday's pattern. This is in spite of higher passenger loading on these two weekend days (average of 593 passengers per trip) versus these five weekdays (average of 320 passengers per trip). Further analysis of other impacts would need to be made in order to make any conclusions along these lines, however.

Aggregating the data further into daily intervals allows for further analysis of impacts of passenger counts and outdoor temperature on the hotel electricity demand. The red solid line in Figure 34 shows the daily average electric power demand of the hotel system over the period of available data. The blue dashed line shows the average number of passengers per trip, given on the primary y-axis. Note the availability for these data is limited to a shorter time period than the hotel power data. The black dashed line shows average daily outdoor temperature, on the secondary y-axis, for the Færder fyr weather station.

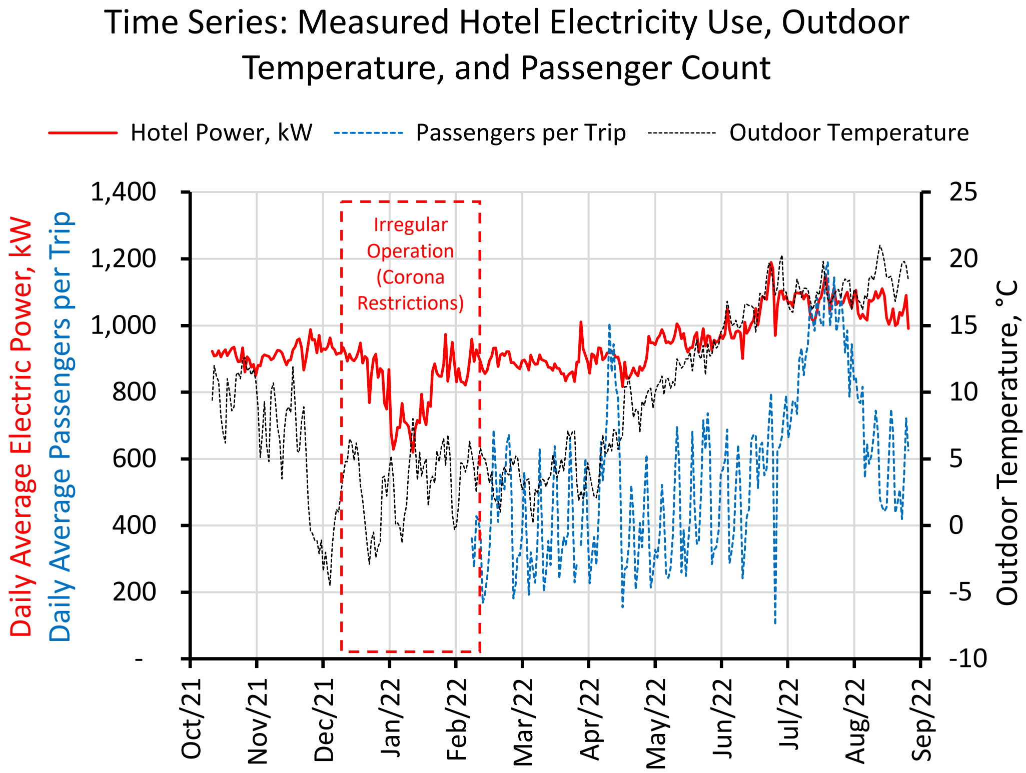

Figure 34 - Time Series of Daily Hotel Electricity Demand, Outdoor Temperature, and Passenger Count

Daily average electric power tends to be around 900 kW, with approximately 100 kW swings up and down, for the majority of the usable period. The exception is a noticeable increase in daily average electric power, to approximately 1100 kW ± 100 kW, beginning around mid-June and lasting through the end of the period. Daily average passengers per trip usually range between 200 and 700, with the exception of the Easter holiday period in mid-April, where the average peaks at about 1000 passengers per trip, and the summer period in July, where the average peaks at about 1200 passengers per trip. Daily average outdoor temperatures in the usable period are as low as nearly -5°C and as high as about 20°C. There appear to be possible relationships between daily average electric power and outdoor temperature.

Figure 35 shows the daily average electric power demand versus the daily average outdoor temperature at the Færder fyr weather station. There are three distinct types of data shown here. The first, represented in dark red points, is data for the first period of usable data (October 13 to December 8, 2021). The green points represent data from the period with unusable data (December 9, 2021, through February 9, 2022). Finally, the red points represent the data from the second usable period (February 10 through August 28, 2022).

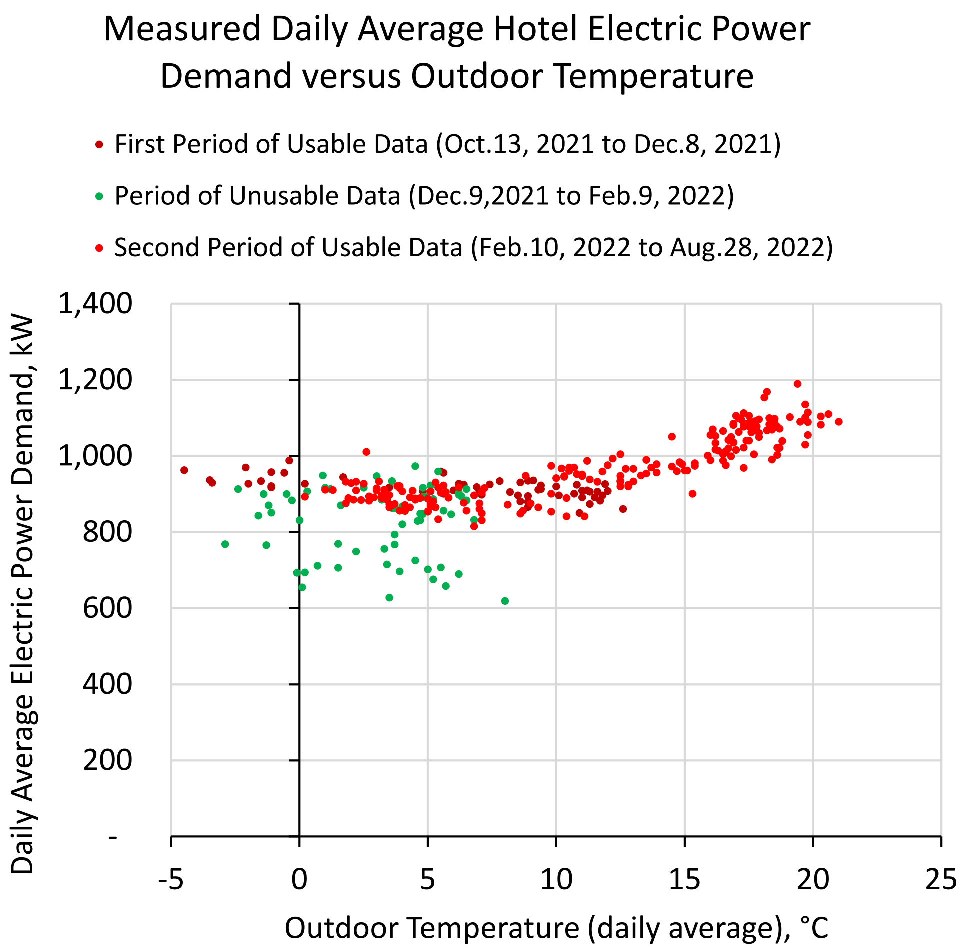

Figure 35 – Measured Hotel Electric Power versus Outdoor Temperature, All Data

Many of the green points from the period of unusable data do not align with the pattern created by the red and dark red points in Figure 35. This is as expected given the irregularity of operation caused by the COVID-19 pandemic.

Also, of note here, is that there appears to be a clear relationship between average hotel electric power demand and outdoor average daily temperature with both the red and dark red points, especially in summer conditions. With average outdoor temperatures greater than about 10°C, there is a rise in daily average electric power demand with an increase in daily average outdoor temperature. This is as expected because of an increased use of the ship's electric chillers to meet the thermal cooling demands.

Under winter conditions, that is temperatures under about 5°C, there appears to be an opposite relationship between daily average electric power consumption and outdoor temperature, but the relationship appears weaker. Since the ship's thermal heating demands are mostly met without electricity, this weak relationship is as expected. Some electric resistance heaters are known to exist on the ship, for example as backup in the hot water system or as space heaters in the car decks, so an increase in daily average electric power demand with falling outdoor temperature is not unexpected. However, these electric heaters do not make up the bulk of the space heating capacity on the ship.

It is desirable to establish a series of linear relationships between measured daily average electric power and outdoor temperature with the usable data shown in Figure 35. Figure 36 includes the same usable data shown in Figure 35, but split into two groups. The red points represent the data with daily outdoor temperatures below 7.5°C, while the blue points represent the data with temperatures above 7.5°C. The red points are considered to have electric loads that are heating-impacted, while the blue points are considered to have cooling-impacted electric loads. The changepoint of 7.5°C was selected based on visual review of the data in Figure 35. Separate linear regression equations and their respective coefficients of determination are shown for the heating-impacted and cooling-impacted loads.

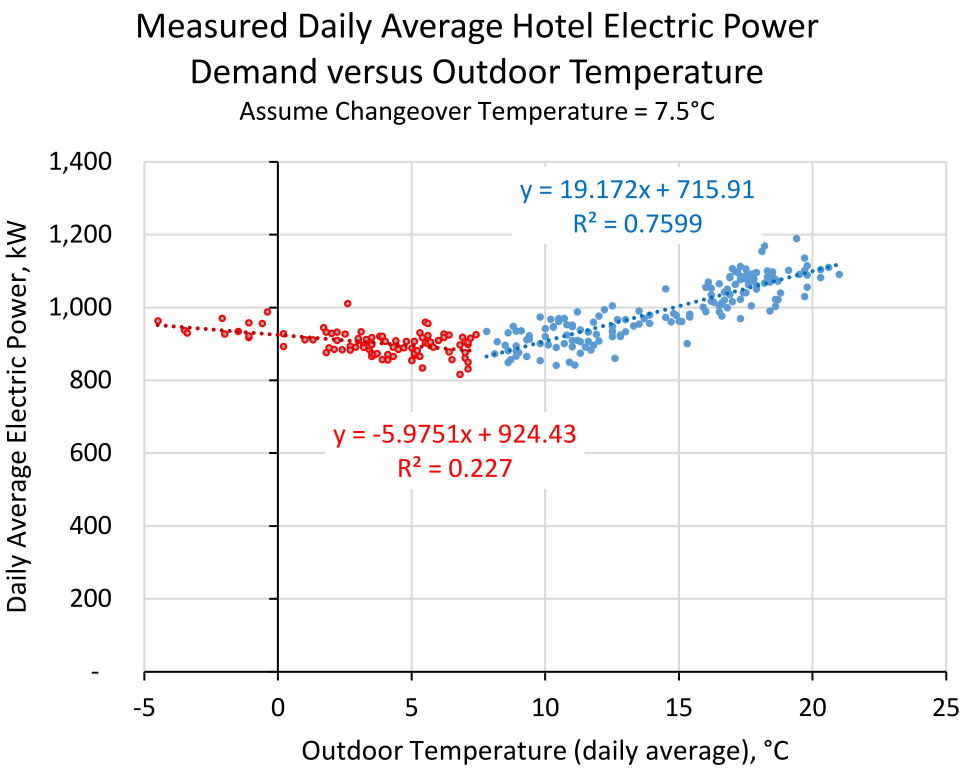

Figure 36 – Measured Hotel Electric Power versus Outdoor Temperature, Usable Data

The two key elements from this regression analysis that will be useful to the model calibration are:

1. *The electric power baseload* – From the graph, the daily average electric power baseload can be estimated to be approximately 850-900 kW. This is, under normal operating conditions, the portion of the electric power load that is not impacted by outdoor temperature.
2. *The slopes of the regression* – The slope of the regression indicates how much the independent variable impacts the dependent variable – in this case, the independent variable is the outdoor temperature, and the dependent variable is the daily average electric power. The slopes of the regression equations show that the heating-impacted portion of the daily average electric load has a nearly 6 kW increase for every 1 K drop in outdoor temperature (where outdoor temperature is below 7.5°C), and the cooling-impacted portion of the daily average electric load has just over a 19 kW increase for every 1 K increase in outdoor temperature (where outdoor temperature is above 7.5°C).

Before the regression is relied-upon, however, it is important to confirm its robustness. This can be done by considering the following three factors:

A. *The shape of the regression model* – Here, the changepoint temperature is assumed to be 7.5°C. This temperature can have an important impact on the overall results of the regressions. In this case, a V-shape curve was chosen, where just one changepoint temperature is assumed. It is also possible to have two or more changepoint temperatures to help describe a more detailed relationship between the independent and dependent variables. In this case, the chosen V-shape curve appears to be adequate based on a visual review. Additional statistical analysis could be made to optimize selection of the form of the fit and the changepoint temperature(s), but that is considered to be outside the scope of this study.

B. *The goodness of fit* – The coefficient of determination is a reasonable indicator of whether the regression is explaining the variability in the data. In the case of the cooling-impacted load, the R^2 value of 0.76 shows the regression is reasonably good at explaining the variability. The regression applying to the heating-impacted load is much lower, with an R^2 of 0.23. This is not necessarily a cause for concern given the fairly flat slope.

C. *Other factors* – Only outdoor temperature is taken into account in these regressions. Other factors that could impact the estimate should be investigated. In this case, passenger count should be investigated further, given that it had a meaningful impact on delivered energy, as shown in Section 6.1.

The robustness of the two proposed regression equations shown in Figure 36 with respect to the passenger count is made by investigating the residuals. Here, the residuals are defined as the regression-estimated daily average electric power minus the measured value. Figure 37 shows a scatter plot of these residuals versus the average passenger count per trip.

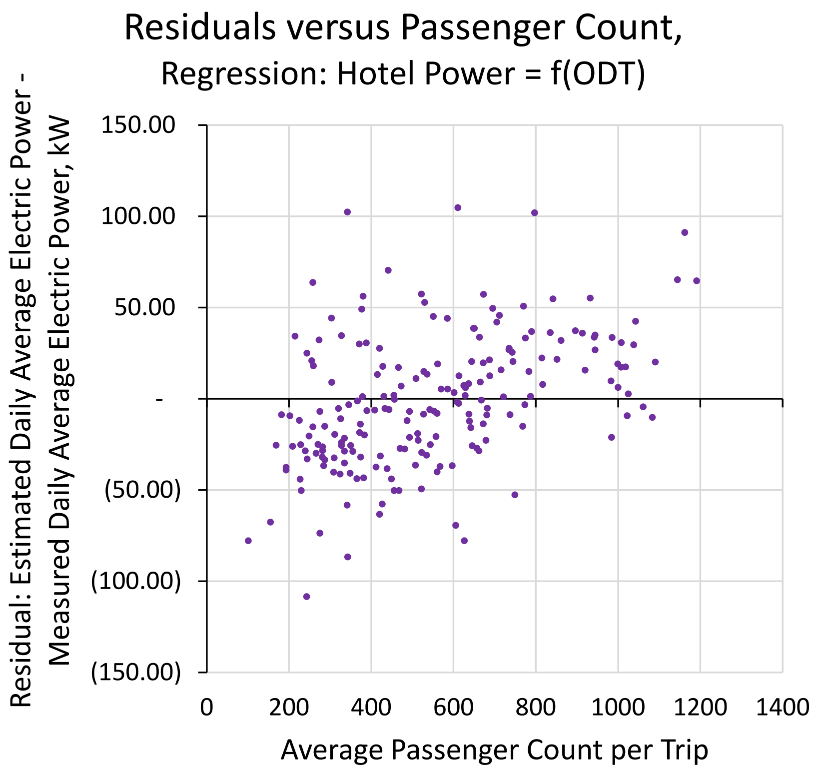

Figure 37 – Residuals of the Regression as a Function of ODT versus Passenger Count

The residual plot in Figure 37 helps show whether the regression needs revision to include passenger count as an independent variable. Here, it is fairly clear that the average residual in the 200-to-400 passenger per trip bin is negative, whereas the average residual in the 800-to-1000 passenger per trip bin is positive. This trend appears to be consistent across the passenger count range. This indicates that there is a relationship between daily average electric power and passenger count that is unaccounted for in the regression that only considers outdoor temperature.

To fix the above issue, a linear regression with two independent variables – passenger count and outdoor temperature – was generated. The results and key characteristics of this new regression scheme are shown in Table 13, including the changepoint, the coefficient for the outdoor temperature independent variable, the coefficient for the passenger count independent variable, the y-intercept, the R^2, and the number of observations.

Table 13 - Regression Characteristics: Daily Average Electric Power as a Function of Outdoor Temperature and Passenger Count

Regression Characteristic	Heating-impacted Regression	Cooling-impacted Regression	Units
Changepoint	Less than 7.5°C	7.5°C or higher	
Outdoor Temperature Coefficient	-9.71	15.06	kW/°C
Passenger Count Coefficient	0.093	0.094	kW/Passenger per trip
y-intercept	892	724	kW
R-square	0.48	0.79	
Observations	69	130	days

The equation for the heating-impacted conditions represented by the data in Table 13 is shown in Equation 1.

Equation 1 - Daily Average Electric Power Regression, Heating-impacted Conditions

$$Daily\ Average\ Electric\ Power\ [kW] = -9.71\ \times Outdoor\ Temperature\ [°C] + 0.093\ \times Passenger\ Count\ [Average\ Passengers\ per\ Trip] + 892\ .$$

The equation for the cooling-impacted conditions represented by the data in Table 13 is shown in Equation 2.

Equation 2 - Daily Average Electric Power Regression, Cooling-impacted Conditions

$$Daily\ Average\ Electric\ Power\ [kW] = 15.06\ \times Outdoor\ Temperature\ [°C] + 0.094\ \times Passenger\ Count\ [Average\ Passengers\ per\ Trip] + 724\ .$$

The R^2 for both regressions in Table 13 have improved as compared to the regressions only considering outdoor temperature. A residual scatter plot for the new regression equations versus passenger count is shown in Figure 38.

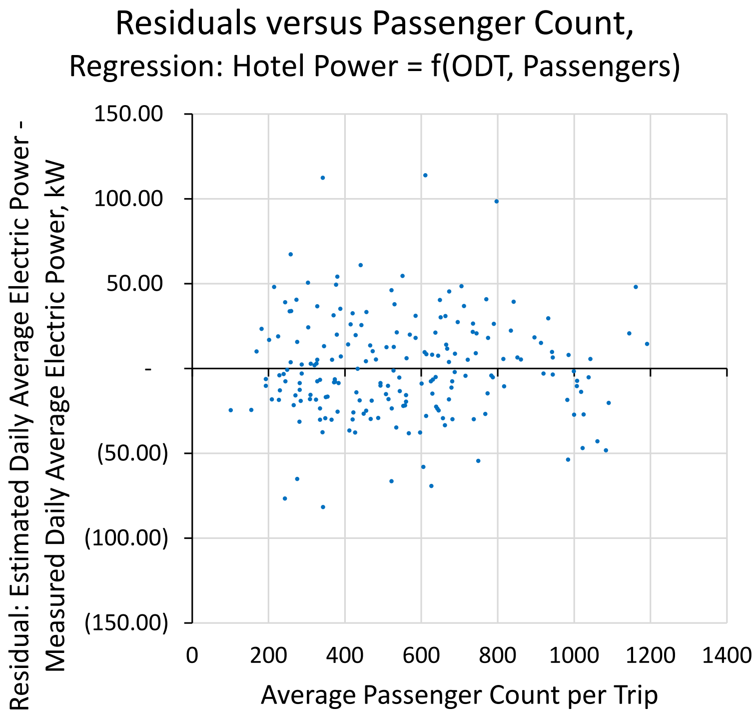

Figure 38 – Residuals of the Regression as a Function of ODT and Passenger Count versus Passenger Count

While there still exists scatter in Figure 38, there is an improved balance across the y = 0 line. This means the revised regression scheme results are sufficiently accounting for both the effects of outdoor temperature and passenger counts. Therefore, the regression represented by Table 13, Equation 1, and Equation 2 is recommended for use as a tool to calibrate the modeled electric demand of the ship.

6.3 Chilled Water Temperatures

In this section, historical chilled water temperature data are used to better understand the operation of the chilled water system. This analysis is useful for understanding the seasonal patterns of the chilled water system operation, including supply and return temperatures, as well as utilization patterns of the various water chilling devices on the ship, namely the seawater heat exchanger, absorption chiller, and the two electric chillers.

Available data for the period January 23, 2022, through January 23, 2023, include temperature readings for different positions in the chilled water system. In addition, the dataset includes fields for runtimes for the electric chillers and absorption chiller, as well as power consumption for the two chilled water pumps. However, these fields are either empty or null for the entire period. Therefore, this analysis focuses only on the available temperature data. Figure 39 shows a simplified diagram with the location of the temperature sensors in the chilled water system. The label codes, for example "4TT5" for the Chilled Water Supply sensor, are also provided. Not shown in the figure is a temperature sensor indicating the temperature of the seawater (4TT3).

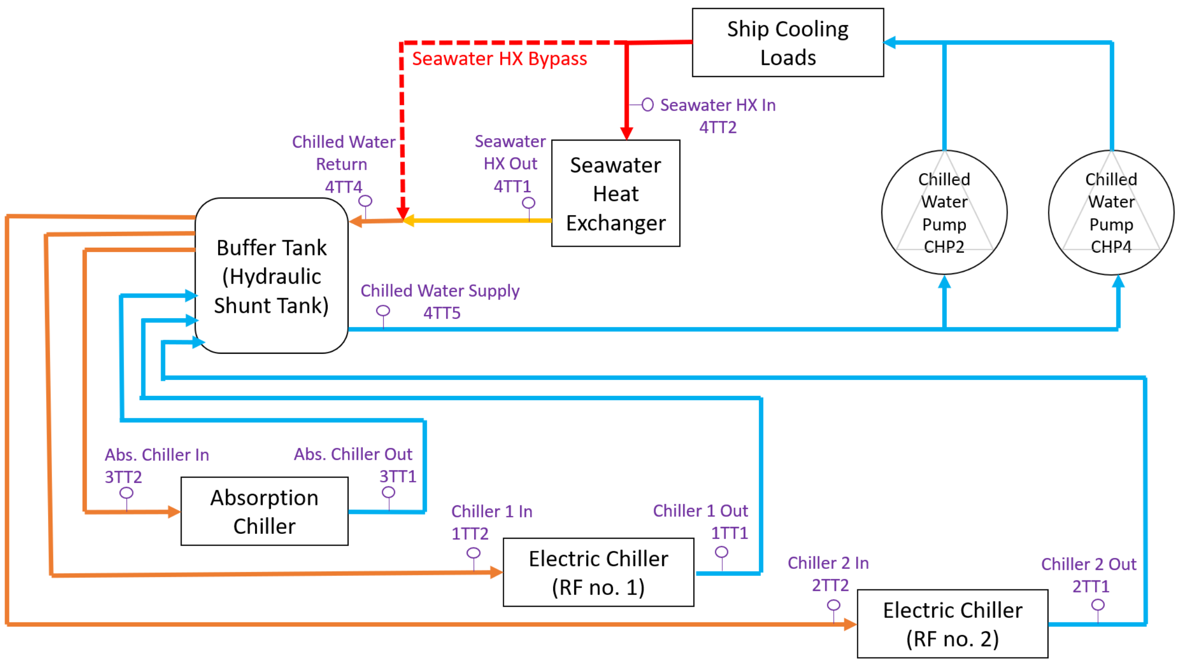

Figure 39 - Chilled Water System Temperature Sensor Locations

As indicated in Figure 39, there are two chilled water pumps that serve the ship cooling loads. The chilled water returns to a buffer tank, and then to a combination of the seawater heat exchanger, the absorption chiller, and the electric chillers. The intended control of the system is described in Section 5.4.2.

The Teknotherm monitoring and control screen, shown in Figure 7, conflicts with the detailed design drawings "Chilled Water Diagram" and "R06 CW Function Sketch"[9] in that the monitoring and control screen shows a bypass of the seawater heat exchanger for the portion of the return water coming from the loads in the forward areas of the ship. The two detailed design drawings show all return water flowing through the seawater heat exchanger, and with a normally closed valve that could act as a bypass. Shown in Figure 39 is a seawater heat exchanger bypass, with a dashed red line. While the temperature data were explored in more detail in an attempt to ascertain in which configuration the ship operates, the results were inconclusive. It is unknown whether the ship operates with all return water flowing through the seawater heat exchanger, or only a portion.

A time-series plot of all the available and usable temperature data for the chilled water system is shown in Figure 40.

[9] The detailed design drawings are not shown here, see Section 1.3.

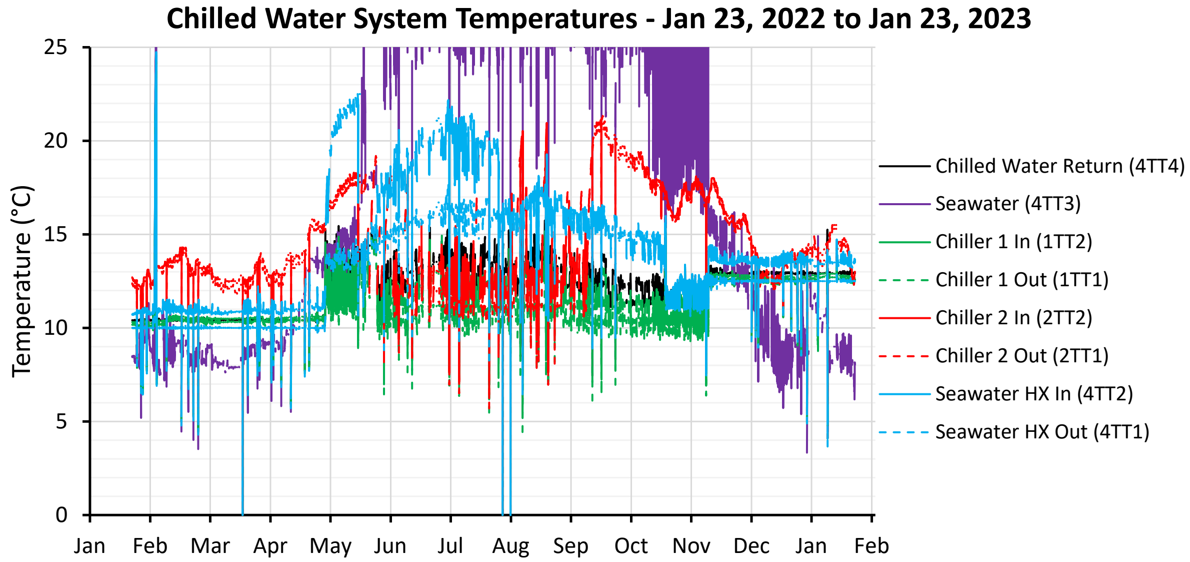

Figure 40 - Chilled Water System Temperature Data

From the view of the data shown in Figure 40, it is challenging to draw conclusions, so more detailed review occurs below. However, it can be seen that the seawater temperature sensor is not providing a proper reading starting in mid-May and lasting through early November. The seawater temperature is registered above 25°C for most of the period. The surface seawater temperature is very unlikely to be this high (See Figure 5), so this is likely an error. The improper reading could be caused by the sensor location being in stagnant water that heats up during the summer months because of a summer operating mode. Otherwise, the sensor appears to be giving reasonable values during most[10] of the winter months.

Missing from Figure 40 are data from the absorption chiller's inlet and outlet temperature sensors, "Abs. Chiller In 3TT2" and "Abs. Chiller Out 3TT1", respectively. These data are not useful because the absorption chiller was not operational during the available time period. The data are mostly null, except for a few weeks in the summer months where the two temperatures are nearly equal. The chilled water temperatures into and out of the absorption chiller's evaporator will not be further considered in this analysis.

Also missing from Figure 40 are data from the chilled water supply temperature (4TT5). After comparison with the air handler unit supply temperatures in Section 6.4, it was discovered that the chilled water supply temperature is consistently higher than the supply air temperatures in many of the air handlers. Figure 41 shows the chilled water supply temperature and the supply air temperature for one of the air handlers, AP-2.1, for a part of 2022.

[10] The causes of the large increase in temperature in early February and the large decreases over the period are unknown and not explored further here since they are of such short duration.

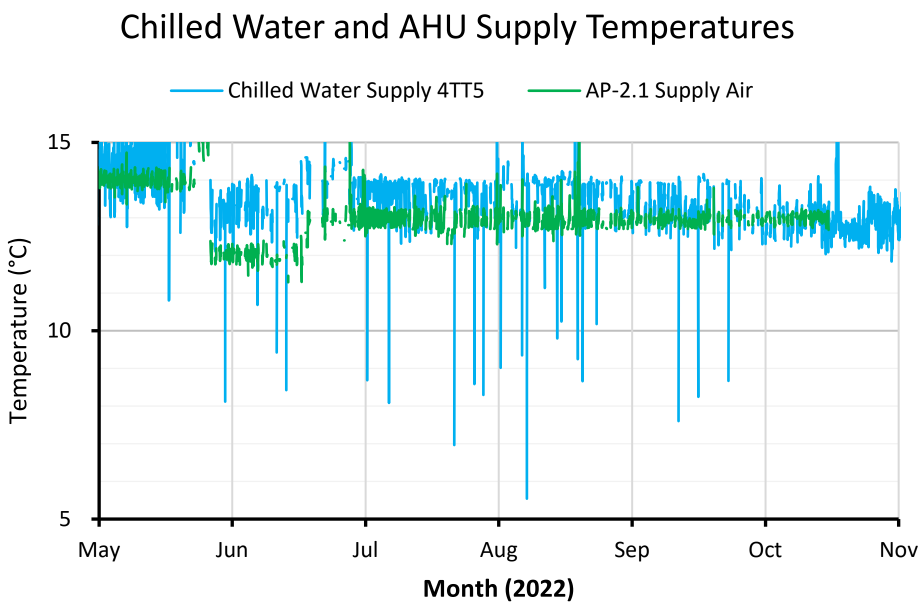

Figure 41 - Chilled Water Supply Temperature Sensor Failure

Since the only source of cooling in the air handlers is the cooling coils, which receive chilled water, the higher chilled water supply temperature shown in Figure 41 is not physically possible. Through discussions with the ship engineer, it was confirmed that the "Chilled Water Supply 4TT5" temperature sensor is malfunctioning and that a fix should take place in April 2023. The ship operators have been able to rely instead on one of the chiller outlet temperatures, depending on which one is operating, as a proxy for the chilled water supply temperature [14]. For this analysis, when the electric chillers are operating, as identified by their low outlet temperature reading, the "Chiller 1 Out 1TT1" or "Chiller 2 Out 2TT1" temperature sensor data are used as a proxy for the chilled water supply temperature. When the chillers are not in operation, the "Chilled Water Return 4TT4" has the lowest temperature and can therefore be used as the proxy for the chilled water supply temperature. This adjusted chilled water supply temperature, along with the temperature data for the supply air temperature in air handler AP-2.1, is shown in Figure 42.

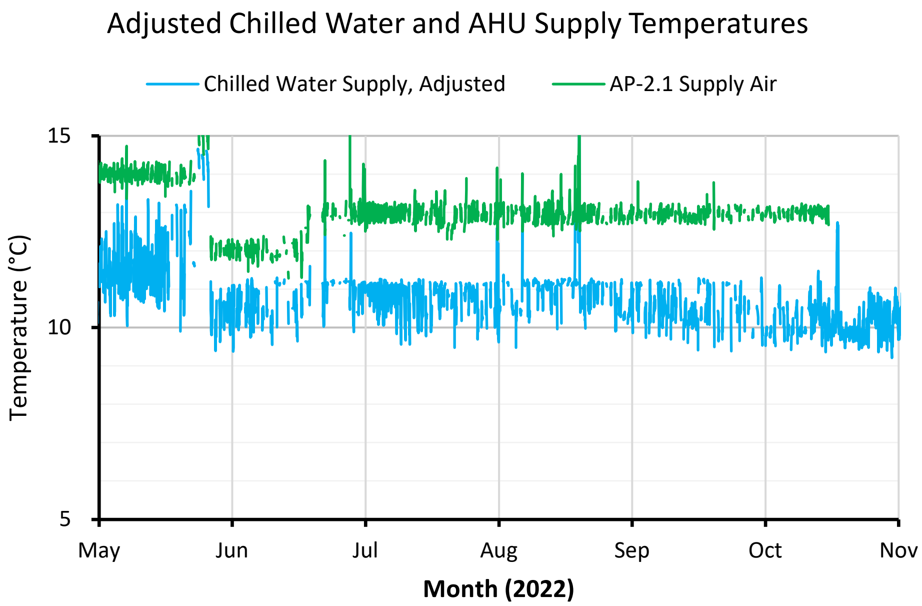

Figure 42 – Adjusted Chilled Water Supply Temperature

The results of the adjusted chilled water supply temperature, as shown in Figure 42, are promising. The supply air temperature is above the adjusted chilled water supply temperature, which now complies with the laws of physics. This adjustment will be used in the remainder of this report.

The data in Figure 40 can be used to calculate the differences in temperature across each of the three heat exchangers. This is shown in Figure 43, where:

- Chiller 1 is the temperature difference between "Chiller 1 In (1TT2)" and "Chiller 1 Out (1TT1)"
- Chiller 2 is the temperature difference between "Chiller 2 In (2TT2)" and "Chiller 2 Out (2TT1)"
- Seawater HX is the temperature difference between "Seawater HX In (4TT2)" and "Seawater HX Out (4TT1)"

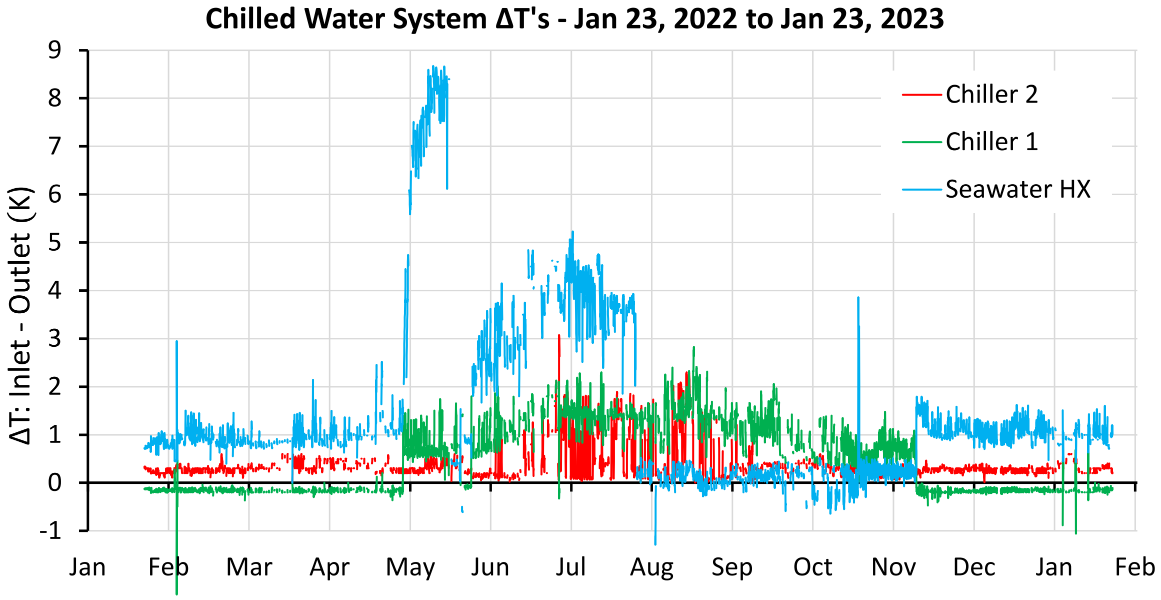

Figure 43 - Chilled Water, Temperature Differences Across Cooling Unit Heat Exchangers

Unfortunately, the chilled water flow rates through each of the three heat exchangers in Figure 43 are unknown. It is also unknown what state the chillers and seawater heat exchanger are in – whether the chillers are operating and at what speed, and the status of the pump on the seawater-side of the seawater heat exchanger. However, even with these limitations, the temperature differences shown here can provide a better understanding of the actual chilled water system operation and show the potential value of additional data collection.

Both electric chillers (Chiller 1 and Chiller 2) are likely off during the winter, given the ship's low cooling demand and the adequate cooling capacity of the seawater heat exchanger during times with low seawater temperatures. The difference between the inlet and outlet temperature readings for both chillers in Figure 43 confirm this: the temperature differences are near zero. The fact that the differences are not exactly zero is likely due to a slight difference in the calibration between the respective inlet and outlet temperature sensors, rather than an indication of heat being removed by, or added to, the evaporator.

Starting in May, the temperature difference for Chiller 1 jumps from just under 0 K to just under 1 K and then up to about 2 K in mid-August, indicating what appears to be a fairly constant use during the summer months. Chiller 2 appears to have more occasional use during the summer, especially in July and August. A clear transition from operation of electric chiller 1 to operation of the seawater heat exchanger is evident in early November, where the chiller temperature difference drops to a level near zero and the seawater heat exchanger temperature difference jumps from near zero to above 1.

The approximately 0 K difference between inlet and outlet temperatures of the seawater heat exchanger, as seen in Figure 43, indicates no operation during the months of August, September, and October, when the sea temperatures are perhaps too high to provide useful cooling to the chilled water system. For the remainder of the year, the seawater heat exchanger has a temperature difference of 1 K or higher, indicating it is in use during the majority of the year. Notably, during the first summer months, and especially in the first half of May, there is a noticeable increase in the temperature difference. In the first half of May, the seawater heat exchanger has a temperature difference of about 7 to 8 K, and in the months of June and July the difference is about 3 to 4 K.

Without water flow rates, it is not possible to estimate the energy exchange in the seawater heat exchanger, which would help understand if the seawater heat exchanger was used to its full capacity. Regardless, the high temperature differences indicate that the operational data in early-May are perhaps anomalous.

The temperature differences shown in Figure 43 can be used to estimate a duty cycle for the three different devices used for water chilling. Note that the absorption chiller has a duty cycle of 0 for the entire period, so it is not included in this analysis. A unit is considered “on” if the temperature difference across its heat exchanger is higher than its “off” position, or significantly higher than zero. Figure 44 shows the resultant estimated duty cycles for each month, where the duty cycles have been stacked on top of one another.

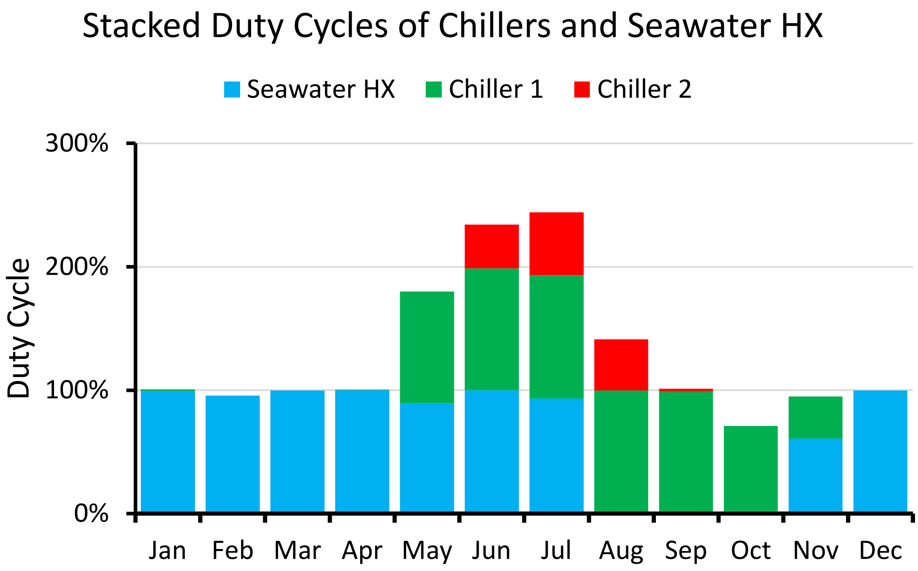

Figure 44 - Duty Cycles for Chillers and Seawater Heat Exchanger

Although the duty cycles are shown stacked in Figure 44, the duty cycles for each unit were estimated independently from one another. In addition, although the graph y-axis goes above 100%, this does not mean the duty cycle for an individual cooling device is higher than 100%, just that more than one unit operated that month. While stacking the duty cycles of the units in this manner can be counterintuitive, doing so allows for easy identification of the changes in operational patterns over the year. As clearly shown in Figure 44, the seawater heat exchanger is, in fact, used as a baseload for over 8 months of the year. Chiller 1 operates nearly continuously for the months May through October, and even somewhat into November. Chiller 2 operates more infrequently during the main summer months of June through August. Importantly, the capacities of these three cooling devices are variable, so the duty cycles shown here only indicate on/off and do not provide much insight into the size of the loads being met.

Combining the operation modes shown here with the seawater and chilled water return temperature data gives insight into energy-saving opportunities related to increased utilization of the seawater heat exchanger. The data component of this question is explored in more detail below, along with discussion of the potential efficiency opportunity.

The temperature data for the chilled water supply (adjusted as shown in Figure 42), chilled water return, and inlet and outlet to the seawater heat exchanger are shown in a time-series graph in Figure 45.

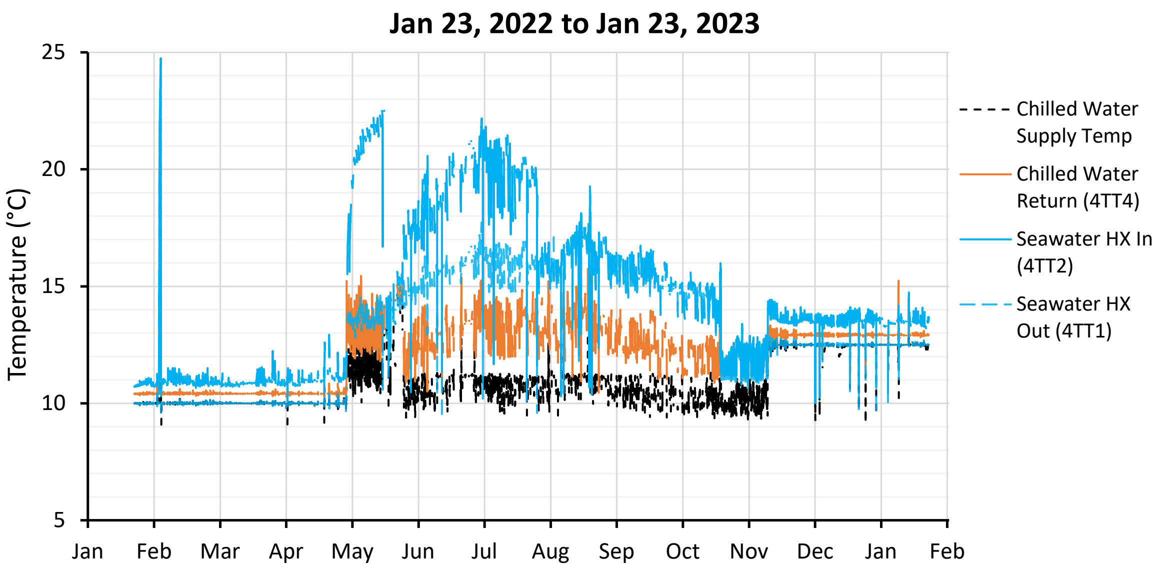

Figure 45 - Chilled Water System Temperature Data, Supply, Return, and Seawater HX

With the piping arrangement and temperature sensor locations shown in Figure 39, if all chilled water returning from the ship's cooling loads were to pass through the seawater heat exchanger, then the Seawater HX In (4TT2) temperature sensor would indicate the temperature of the chilled water returning from the cooling loads. In this case, the Chilled Water Return (4TT4) and Seawater HX Out (4TT1) temperature sensors should also give approximately the same reading. This is the case in the winter months of mid-November through most of April, where the Chilled Water Return (4TT4) temperature sensor reads about 0.4°C higher than the Seawater HX Out (4TT1) sensor. This difference could possibly be attributed to a measurement or calibration error.

During simultaneous operation of the seawater heat exchanger and one or both of the chillers, which occurs from the end of April through the majority of July, the temperature readings are difficult to reconcile with one another and the piping configuration. In this period, it appears that some of the water returning from the ship's cooling loads flows through the seawater heat exchanger and that the remainder bypasses the seawater heat exchanger. The temperature of the water flowing into the seawater heat exchanger is indicated by Seawater HX In (4TT2) and ranges from 16°C to 22°C, and the temperature of water flowing out of the seawater heat exchanger ranges from 14°C to 17°C. The temperature of the mixed water coming from both the bypass and the outlet of the seawater heat exchanger, as shown by the Chilled Water Return (4TT4) temperature sensor, ranges from 11°C to 15°C during the period. Since the temperature indicated by the Chilled Water Return (4TT4) sensor is lower than both that of the Seawater HX In (4TT2) and Seawater HX Out (4TT1) sensors, it appears as though the portion of the return water that bypasses the seawater heat exchanger is much colder than the portion of return water that flows through the seawater heat exchanger. Without additional information on the piping configuration and flow rates, the temperature of the water returning from the cooling loads is unknown. Additional research and data collection are recommended. However, even with the lack of clarity described here, it does appear that the seawater heat exchanger could be useful even with relatively high seawater temperatures. Since the return water the seawater heat exchanger receives during the early summer months from the ship's cooling loads is so warm, it is possible that similar warm return temperatures could be taken advantage of by the seawater heat exchanger in the fall, even with relatively warm seawater temperatures.

In the months of late-July through early-November, when the seawater heat exchanger is not in operation, there is no temperature difference across the seawater heat exchanger, as shown in Figure 43. However, the temperature readings for both the inlet and outlet sensors in the seawater heat exchanger show temperatures ranging between 14°C and 18°C, likely indicating the flow of chilled water continuing to return at a higher temperature through the seawater heat exchanger than through the bypass. There is still a high level of uncertainty in the interpretation of these data, but they provide some insight into possible energy-saving opportunities. The uncertainty in an energy savings analysis would be too high to present here. It is recommended that the ship operators verify that the seawater heat exchanger is being turned on in the fall as early as it is able to assist in cooling the chilled water return.

6.4 Air Handler Units

The data available, as introduced in Section 4.4, for the ship's 14 air handler units – see Table 10 for detailed characteristics – are described in detail in this section. These data are used to develop a better understanding of the operation of the ship's ventilation system. This allows verification of some important IDA ICE model parameters, as well as insight into possible efficiency upgrades.

The VAV air handler unit designated as AP-2.1 has the longest period of data availability – from October 17, 2021, through October 17, 2022. Data availability for the remainder of the air handlers is limited to the period July 9, 2022, through October 17, 2022. Figure 46 is a time-series graph of hourly average values for seven key temperature readings (upper graph) and control levels (lower graph) for AP-2.1.

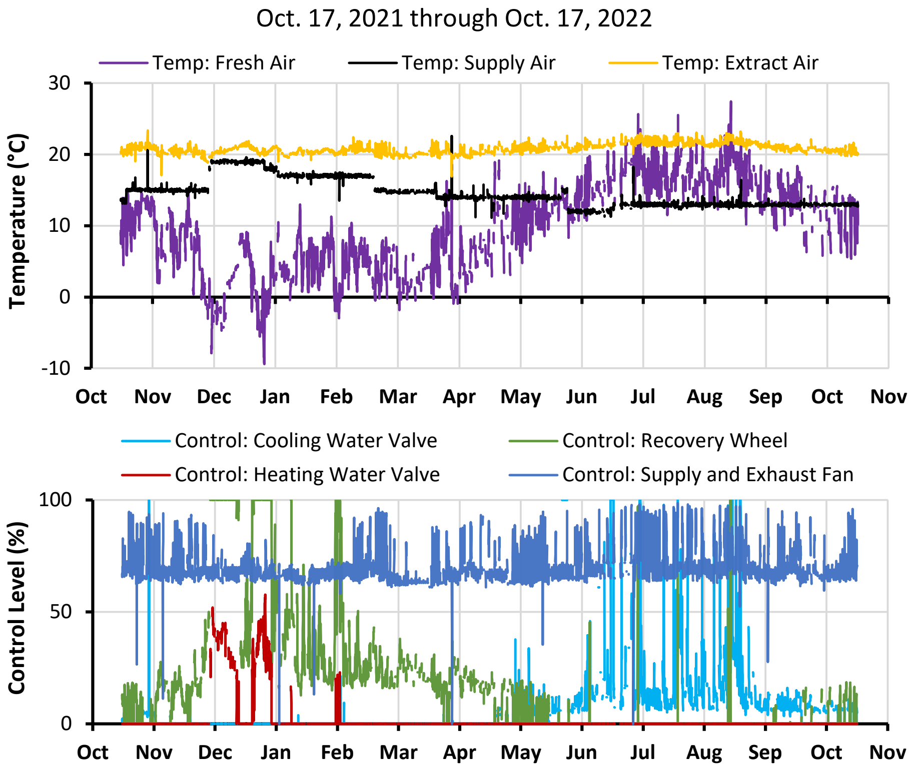

Figure 46 - Key Data Available for the Air Handler Unit AP-2.1

Not all the data available for AP-2.1 are shown in Figure 46. Instead, the variables most applicable to this work are plotted as an introduction to the detailed data provided. Temperatures are shown on the primary y-axis, while the control levels are shown on the secondary y-axis. The data shown in Figure 46 and their implications for this report are as follows:

- *Temp: Fresh Air*. The temperature of the fresh air coming into the air handler follows the expected pattern of cold temperatures in the winter and warmer temperatures in the summer. A comparison (not shown here) of the air handler fresh air temperature data to the Færder Fyr weather station show agreement, with a deviation of less than 2.5 K for most hours of the year, and an average deviation over the course of a year of less than 0.4 K.
- *Temp: Extract Air*. The temperature of the extract air coming from the rooms served by the air handler is often just above 20°C, with a noticeable slight rise in the summer months. While the extract air temperature data are not discussed further in this report, they are shown here to highlight that in future work a comparison of the modeled extract temperatures could be made to the measured data, and detailed adjustment could be made to the model's internal gains assumptions in the spaces served by each air handler (Section 10.2).
- *Temp: Supply Air*. The supply air temperature for AP-2.1 ranges between 11°C and 20°C. The supply air temperature setpoint, not shown here, was also made available. In a separate analysis, the measured supply air temperature indicates very good adherence to the supply air temperature setpoint. Therefore, the step changes seen in the supply air temperature data in Figure 46 are closely linked to changes in the setpoint. The supply air temperatures for this and the remaining air handler units are analyzed in more detail in this section.
- *Control: Supply and Exhaust Fan*. The control settings for the supply and exhaust fans in AP-2.1, which represent the fan speed as a percentage of full speed, range from approximately 70% to 100%.
- *Control: Recovery Wheel*. It is not clear whether the control signal for the recovery wheel represents speed of the wheel or desired percentage of rated efficiency. However, it is shown here in case it would be of interest in future work related to a more detailed calibration of the heat recovery component of the model.
- *Control: Cooling Water Valve.* The controlling position of the cooling water valve indicates the flow rate of cooling water to the cooling coil in the air handler, as a fraction of the design flow rate. This provides an indication of the cooling demand of the air handler, which, according to the data shown in Figure 46, increases as expected during the warmer summer months and is mostly zero through the remainder of the year. For this work, the cooling water control valve data were used to improve understanding of the operation of the air handlers. However, these data could be used in future work to perform a more detailed calibration of the model of the cooling system in the air handlers and the associated cooling loads in the zones served.
- *Control: Heating Water Valve.* The controlling position of the heating water valve indicates the flow rate of the heating water to the heating coil in the air handler, as a fraction of the design flow rate. This indicates the heating demand of the air handler. For this air handler, the heating water valve control signal is zero most of the year and rises no higher than about 50% for periods with the coldest outdoor temperatures. Similar to the cooling water valve control data, the heating water valve control data were only used in this project to improve understanding of the operation of the air handlers, but they could be analyzed in more detail in future work.

Figure 47 shows the average hourly supply air temperature for air handler unit AP-2.1 plotted against the average hourly outdoor air temperature.

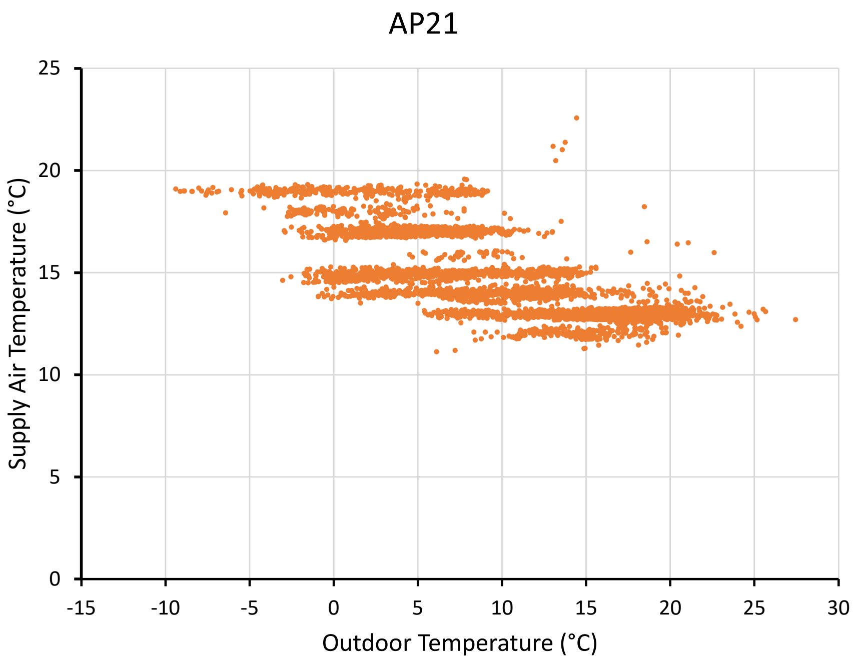

Figure 47 - Supply Air Temperature versus Outdoor Air Temperature, Air Handler AP-2.1

The trend in Figure 47 is decreasing supply air temperature with increasing outdoor air temperature. This is logical given the need for heating in the winter and cooling in the summer. However, it is not clear in either Figure 47 or Figure 46 whether the temperature setting is being adjusted manually or automatically. Unfortunately, the AP-2.1 air handler is the only unit with annual data available, so a comparison is not possible.

The average of the measured hourly supply air temperatures for each of the air handlers over the course of the period July 9th, 2022, through October 17th, 2022, is shown in Figure 48. The wheelhouse air handler, AW-2.4, is not shown here because its supply air temperature varies significantly over the course of the day, suggesting an issue with the sensor or a different control mechanism than the other air handlers. In fact, it appears the wheelhouse air handler's supply air temperature is controlled by a space temperature reading, rather than the supply air temperature itself. Investigation of the operational characteristics of the wheelhouse air handler could be performed in future work and is not discussed further in this report.

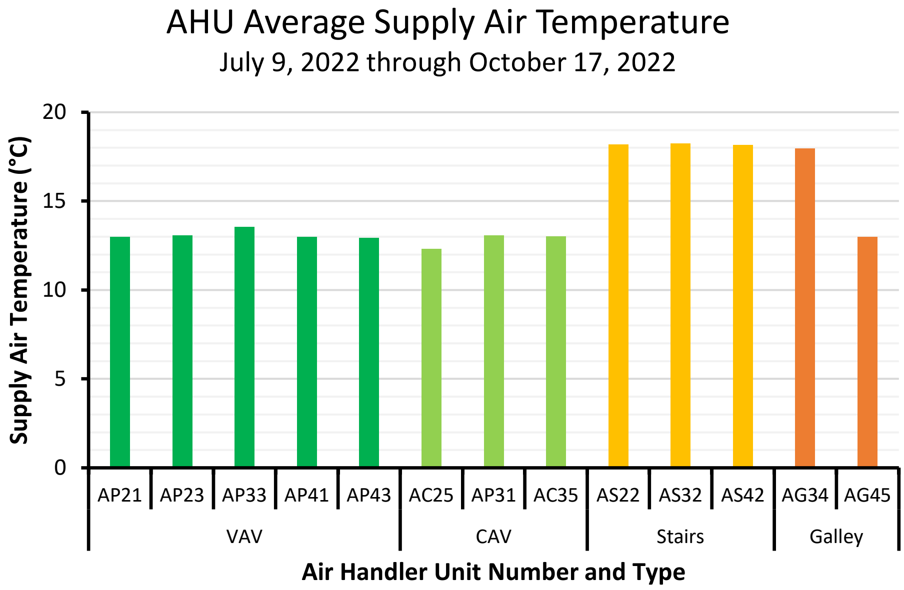

Figure 48 - Air Handler Average Supply Air Temperatures, Summer through Fall 2022

The VAV and CAV air handlers operated with supply air temperatures of approximately 13°C in the summer and fall of 2022, as seen in Figure 48. Air handlers serving the stairs operated on average at about 18.2°C. The two galley air handlers, AG-3.4 and AG-4.5, had average supply air temperatures of 18°C and 13°C, respectively.

With the exception of the three air handlers serving the stairs – AS-2.2, AS-3.2, and AS-4.2 – which each had an initial setpoint of 19°C changed to 18°C in late July, the supply air temperature settings were not changed during the period represented by Figure 48.

Since both the supply air temperature measurement and the supply air temperature setpoint are provided in the dataset, analysis of the deviation of the supply air temperature from its setpoint can be made. An example of this is shown graphically for four of the ship's VAV air handlers in Figure 49. Here, the supply air temperature deviation is defined as the hourly average supply air temperature setpoint subtracted from the hourly average measured supply air temperature.

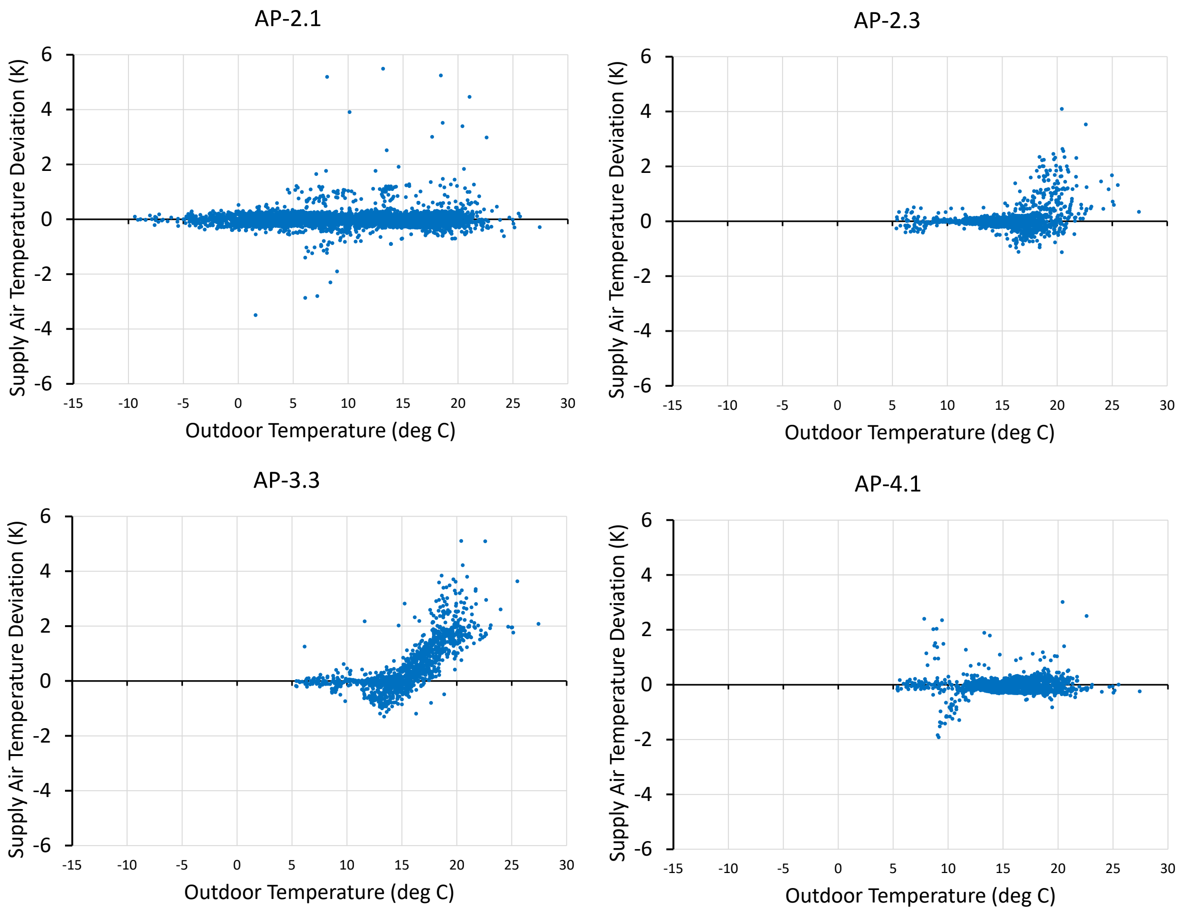

Figure 49 - Supply Air Temperature Deviation Versus Outdoor Air Temperature, Four Examples

In Figure 49, positive deviations correspond to instances where the measured supply air temperature is higher than the setpoint. For example, air handlers AP-2.1 and AP-4.1 tend to be capable of maintaining the correct supply air temperature, independent of outdoor temperature, given that most of the measured points are close to the 0 K deviation line. On the other hand, there are a number of instances at the higher outdoor temperatures around 20°C where air handler AP-2.3 does not appear to have met the supply air setpoint temperature. The data for air handler AP-3.3 show an even clearer pattern of deviation from the setpoint. The supply air temperature begins departing from the setpoint at rising outdoor temperatures, beginning at an outdoor temperature of about 15°C. This indicates a possible issue with overheating of the spaces served by AP-3.3 or lack of adequate capacity of the cooling coil in AP-3.3. Investigation of this issue is discussed later in this section.

A simple indication of the summer and fall supply air temperature deviations can be made by taking an average of all of the hourly deviation values for each air handler unit. This is shown in Figure 50.

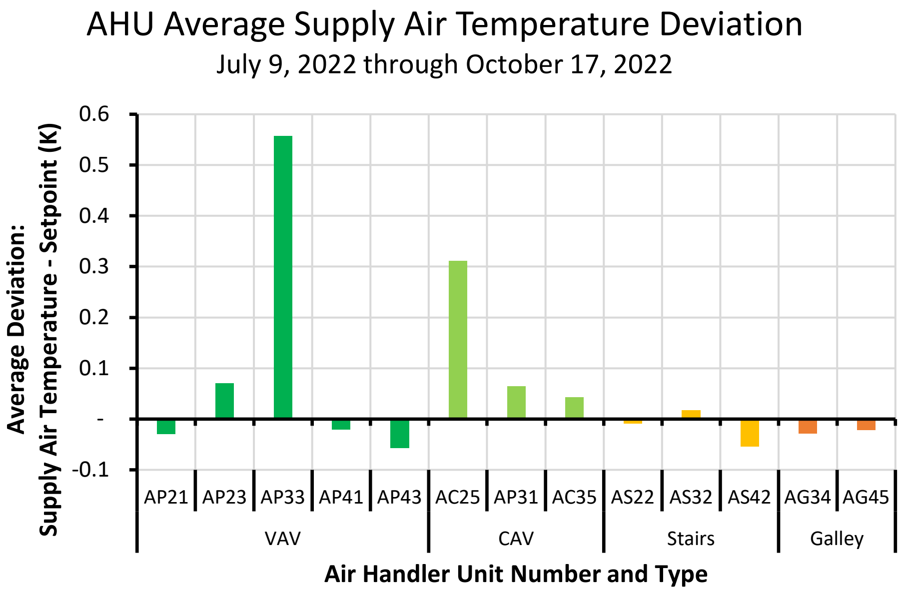

Figure 50 - Air Handler Average Supply Air Temperature Deviation, Summer through Fall 2022

As seen in Figure 50, air handler AP-3.3 had the largest average deviation in supply air temperature over the summer and fall 2022 period. For this reason, the data for AP-3.3 are explored in more detail to develop a better understanding of the operational limitations and to look for potential efficiency opportunities. Air handler AC-2.5 had the second highest average deviation; its data could be explored in future work in a similar manner. The remaining air handler units had only small positive or negative supply air temperature deviations, so it appears they were operating with adequate cooling capacity during this period.

Analysis of the available data for air handler AP-3.3 shows a likely failure in its fresh air inlet temperature sensor. Figure 51 shows the difference in inlet air temperature measured for each air handler and the inlet air temperature measured for air handler AP-2.1. The selection of air handler AP-2.1 as the baseline is not important.

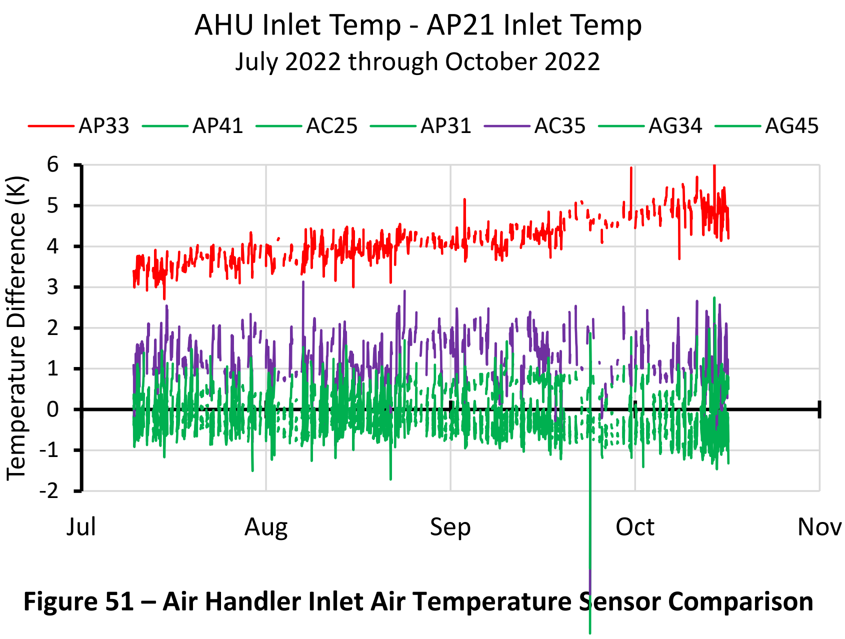

Figure 51 – Air Handler Inlet Air Temperature Sensor Comparison

When compared to the fresh air inlet temperature data for the other air handlers, as seen in Figure 51, the temperature readings for AP-3.3 show a consistently higher reading. Its inlet air temperature reading differs from air handler AP-2.1 by about 3 to 5 K and the deviation gets worse with time. The sensors for the other air handlers are mostly within 1 K of AP-2.1's sensor, with the exception of AC-3.5, which ranges between about 1 and 2 K difference. Review of the duct inlet locations for the air handlers does not indicate a reason for AP-3.3's large temperature difference. The ship's first engineer confirmed that it is possible the temperature sensor has failed [14].

Figure 52 shows a simplified sketch of the VAV air handler units on the ship, including the locations of some of the sensors whose logged data were made available to this project. Temperature sensors in the extract, inlet, and supply airstreams are indicated with a red "T". Relative humidity of the supply air is shown with a blue "RH". The valve position of the chilled water coming out of the cooling coil is indicated with a green "V", as is the valve associated with the heated water used for the heating coil. The control signal to the recovery wheel is shown in purple. Many other sensors are not shown in this figure for simplification.

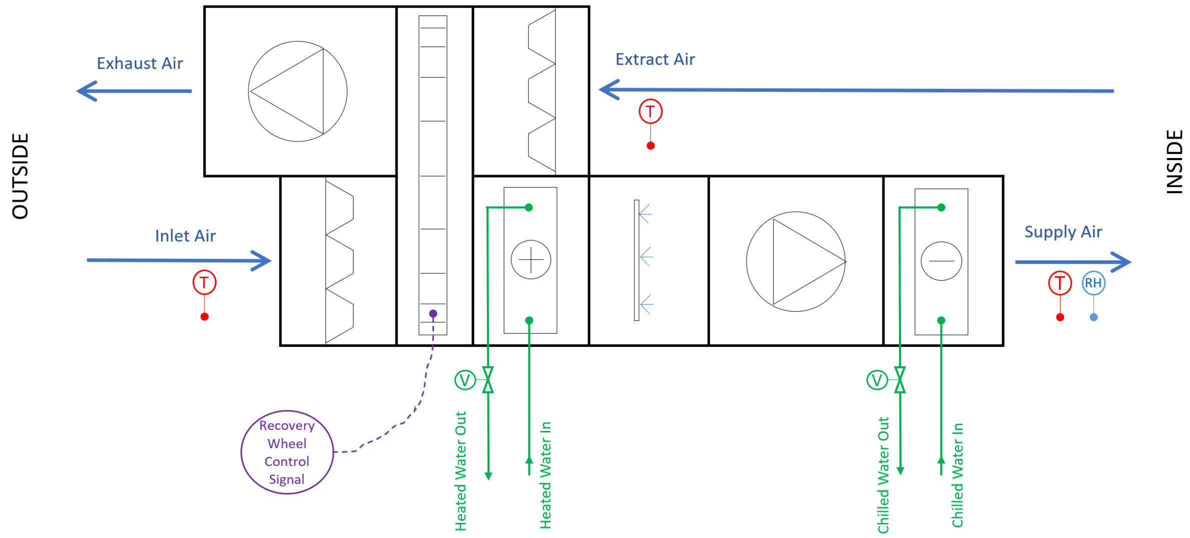

Figure 52 - VAV AHU Simplified Sketch

In summer conditions, when the setpoint temperature is lower than the inlet air temperature, the ventilation air requires cooling before reaching the interior zone. This can be accomplished with the recovery wheel, the cooling coil, or a combination of both. However, the recovery wheel can only provide effective sensible cooling when the extract air temperature is lower than the inlet air temperature. If this is not the case, the rotation of the recovery wheel should be stopped, effectively stopping heat transfer between the two air streams.

For the summer and fall 2022 period discussed above, Figure 53 reveals the operation of the recovery wheel during an intended ventilation air cooling condition for the AP-3.3 air handler. Shown on the x-axis is the hourly average recovery wheel control level, in percentage. The y-axis shows the hourly average temperature difference, or delta T, between the inlet and extract air temperatures. The points in blue are the delta T when the measured inlet air temperature from AP-3.3's inlet air temperature are used. The points in orange are the same conditions but re-calculated with the "actual" inlet temperature. The inlet air temperature from air handler AP-2.1 is used as the proxy for the "actual" inlet air temperature. Not shown here are the data from conditions where the actual inlet air temperature is less than 15°C.

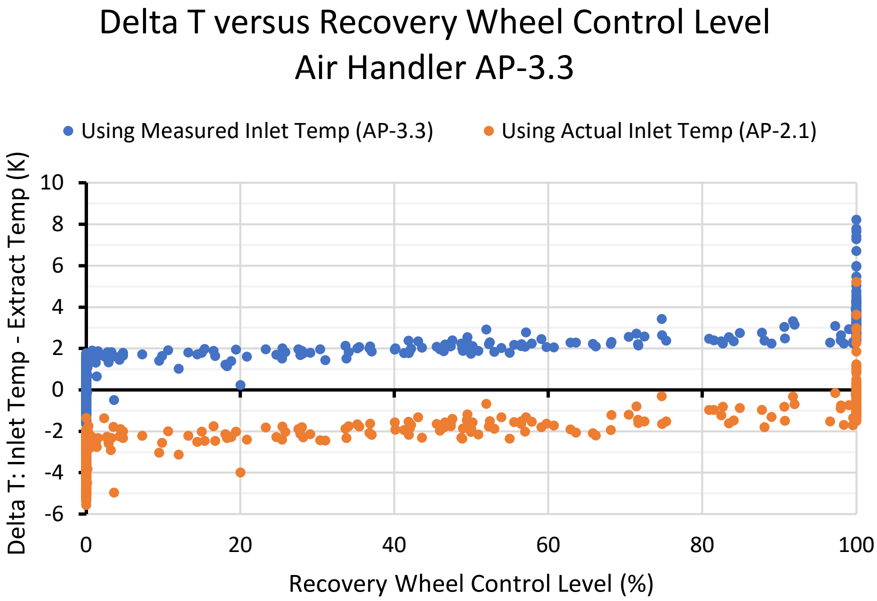

Figure 53 - Recovery Wheel Control, Air Handler AP-3.3

In Figure 53, the blue points using the measured inlet air temperature from AP-3.3 show the same information the air handler's controller had access to. The recovery wheel control level began increasing when the delta T between the inlet air and extract air temperature was just under 2 K. The control signal increased steadily until the delta T reached about 3 K, at which point the recovery wheel control signal was set to 100% to cool down the inlet air as much as possible. If the error in the inlet air temperature sensor for AP-3.3 is ignored, the air handler appears to have good control of the recovery wheel.

However, as indicated by Figure 51, air handler AP-3.3 was working with faulty inlet air temperature data. Since the recovery wheel control level is increased while the delta T is actually negative, the orange points in Figure 53 show that the air handler was "asking" for heat to be removed from supply air stream during conditions when the higher temperature of the extract air stream made that not possible. For the majority of the hours where the ventilation air needed cooling during this period, the improper operation of the recovery wheel caused heating of the inlet air.

Figure 54 shows a histogram of the increase in temperature caused by the improper use of the recovery wheel. The average temperature gain is shown by 0.5 K bins of supply air temperature deviation. Also shown, on the secondary y-axis, is the number of hours with improper use of the recovery wheel, as a percentage of all the hours in the bin. The data in this graph have been filtered for cases where the outdoor air temperature is greater than 15°C.

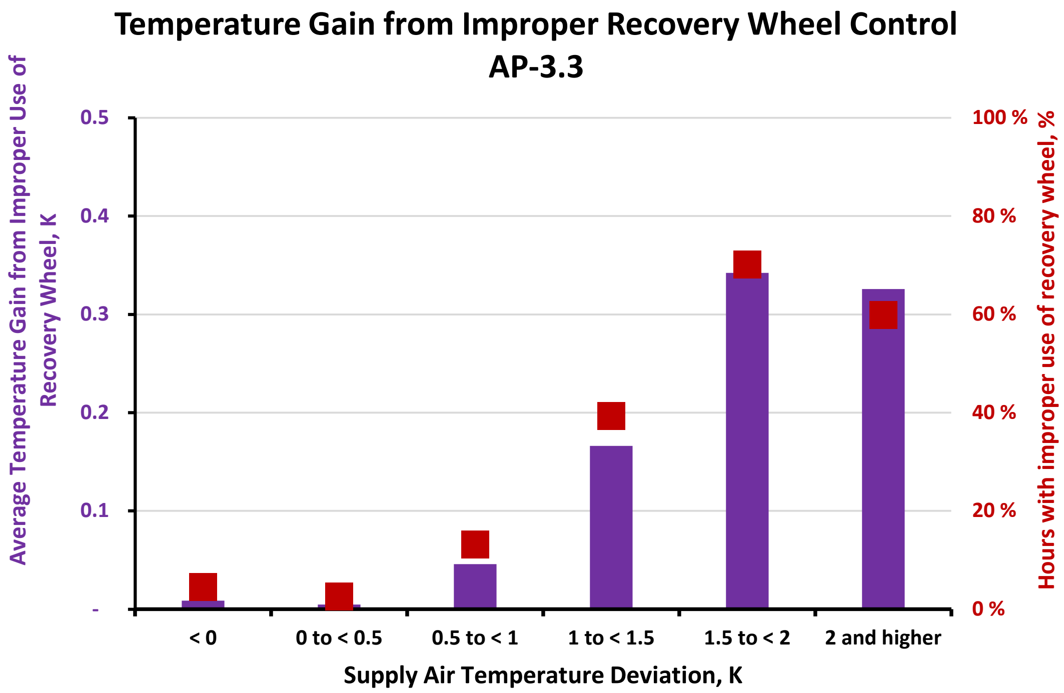

Figure 54 - Temperature Gain from Improper Recovery Wheel Control, AP-3.3

As seen in Figure 54, the improper temperature gain from the recovery wheel increases with increasing supply air temperature deviation, which aligns with the AP-3.3 graph in Figure 49. The temperature gain is never more than 0.4 K. This means the improper operation of the recovery wheel only accounts for a portion of the supply air temperature deviation. For example, in the "1.5 to <2" supply air temperature deviation bin, the recovery wheel should not have been in operation over 70% of the time. If the temperature sensor had been functioning properly, the average temperature deviation could have been reduced by about 0.35 K, or about 20% of the supply temperature deviation problem would have been solved.

The data above show that the air handler AP-3.3 likely did not properly utilize its recovery wheel during summer conditions in the 2022 summer and fall period: The recovery wheel was utilized during conditions where the extract air temperature was higher than the inlet air temperature. This appears to be caused by a faulty inlet air temperature sensor. This is likely one of the causes of the high supply air temperatures for AP-3.3, and placed an unnecessarily high demand on the chilled water system. The operation of air handler AP-3.3, and namely the accuracy of the inlet air temperature sensor, should be investigated further by the ship operators. Since this likely only partially accounts for the lack of capability of the air handler to maintain its setpoint temperature, additional thorough investigation should be made.

An estimate of the energy savings from repairing the temperature sensor is described in Section 8.4.

7 Simulation Model Results

The model provides a large variety of results. Shown in this section are high-level results from the initial model along with some specific results related to the air handler units and indoor temperatures. The subsections that follow describe the model validation (Section 7.1) and the model adjustments (Section 7.2). For the final adjusted model results, see Section 7.2.

The results from IDA ICE, given in terms of delivered energy, represent the electric and fuel energy that needs to be delivered to the ship in order to maintain the requirements of the hotel system. The annual energy use and energy demand per floor area for each of the major end-uses is shown in Table 14.

Table 14 - Annual Energy Demand by Major End-Use, Initial Model

End-Use	Annual Energy, kWh	Annual Energy, kWh/m^2
Lighting	323,349	17
Space and AHU Cooling	214,243	11
AHU Auxiliary	1,520,409	81
Electric Heating	-	-
AHU and Space Heating	1,318,420	70
DHW Heating	564,147	30
Other Electric Loads	118,380	6
Total	**4,058,947**	**217**

The end-use with the largest energy demand is the AHU Auxiliary category, which includes fans, pumps, and humidification related to the heating, cooling, and ventilation system. This high energy demand is expected given the low efficiency of the air handler unit fan and duct systems, caused by the smaller ductwork required to fit within the tight spaces on the ship. The second highest energy demand is the air handler and space heating requirements. This is also as expected given the cold climate.

The modeled hotel energy demands, by major end-use, for each month are shown as average energy demand per day in Figure 55. AHU Auxiliary includes the electric energy used to power the fans, pumps, and humidifier. Electric Heating is zero, but there is a placeholder for it in this graph.

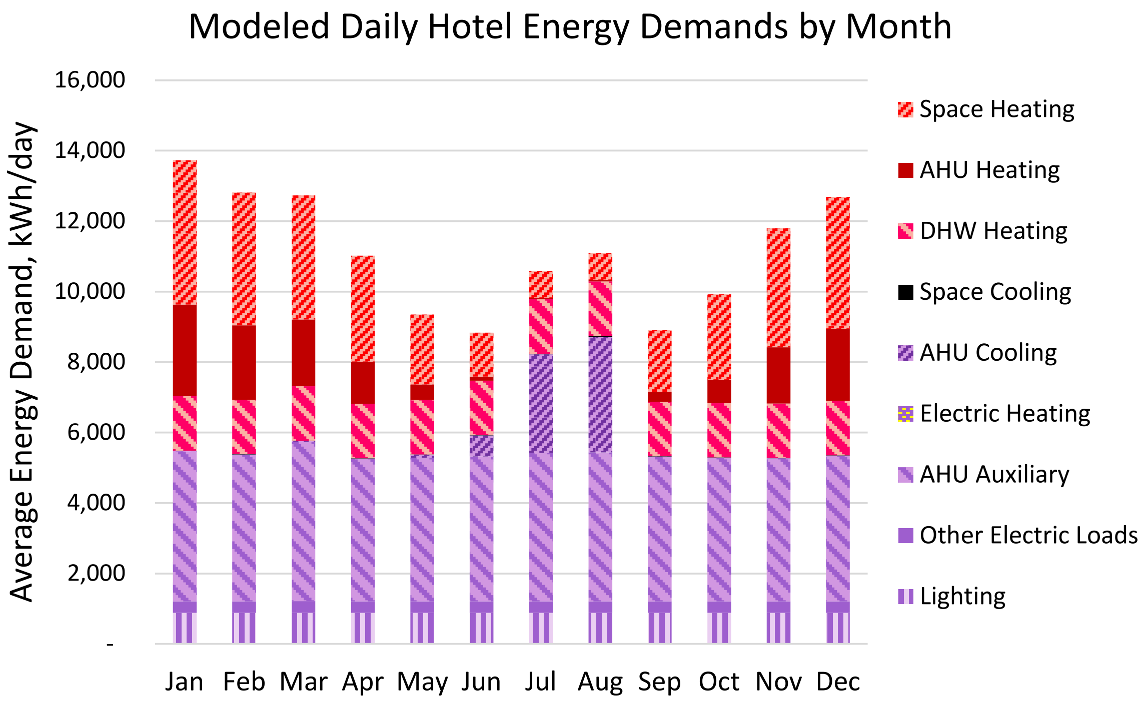

Figure 55 - Hotel Energy Demands by Month, Initial Model

In Figure 55, the results in red represent the hotel demands that are expected to be met with thermal energy. The model assumes backup electric resistance heaters are not used. The DHW heating is uniform across the year, as expected based on the model inputs. The space heating and AHU heating are higher in the winter months than in the summer months because of their dependency on outdoor temperature. These results should be compared to the results of thermal energy use data if it becomes available (see Section 0).

The results shown in purple in Figure 55 represent loads expected to be covered by electricity. These results are discussed further in Section 7.1.2.

A detailed look at the energy use of various end-uses can be made, depending on the level of detail entered in IDA ICE. One category of end-uses with a high level of detail is the ventilation system. The model results related to the six air handler units are given in Figure 56 and Table 15. All the values are in kWh, including heating and cooling energy demand of the ventilation air, fan energy, humidifier energy, and the heat and cold recovered by the heat recovery systems.

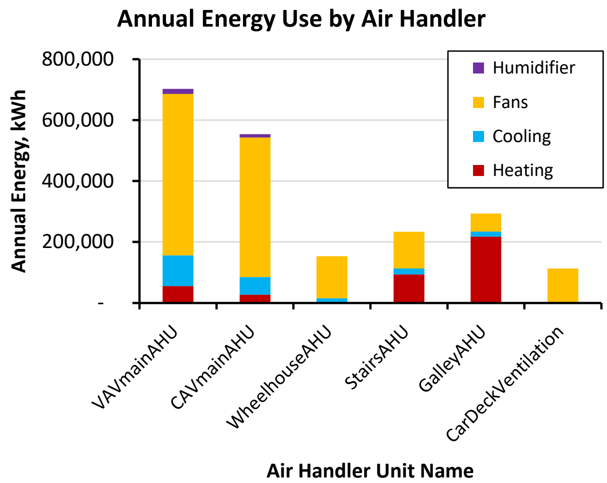

Figure 56 - Annual Results for AHUs, Initial Model

Table 15 - Annual Energy (kWh) for AHUs, Initial Model

AHU	Heating	Cooling	Fans	Humidifier	Heat Recovery	Cold Recovery
VAVmainAHU	55,456	100,676	529,518	16,608	1,323,671	1,292
CAVmainAHU	27,394	57,703	458,067	10,431	849,594	4,958
WheelhouseAHU	0	15,457	138,168	-	807,074	-
StairsAHU	93,443	19,632	120,059	-	462,765	-
GalleyAHU	217,453	16,902	58,944	-	-	-
CarDeckVentilation	0	-	112,721	-	-	-
Total	**393,746**	**210,368**	**1,417,477**	**27,038**	**3,443,104**	**6,250**

The difference between the AHU auxiliary energy in Table 14 – 1,520,409 kWh – and the sum of the total AHU fan energy –1,417,477 kWh – and humidifier energy – 27,038 kWh – shown in Table 15 – represents 75,894 kWh of pumping energy for both the chilled water and hot water circulation. This amounts to an average annual load of 8.7 kW. Limited information is available for the pressure losses in the piping systems and especially the control strategy of the hot water and chilled water pumps, so the validity of the model's estimate of pumping energy is questionable. The high fan energy use shown here confirms the discussion above regarding the low fan system efficiencies. The heat recovery of the air handler units with a heat recovery system is significant. The galley AHU does not have heat recovery, so it has the highest heating load, even though its fan energy (and underlying overall airflow) is the lowest of these systems. This area for possible energy efficiency improvement is clearly visible in the red heating component of the GalleyAHU shown in Figure 56.

The results of the model can also be reviewed by checking the temperatures of the spaces. Figure 57 shows the minimum and maximum temperatures of the zones over the course of the annual simulation. The space temperatures were ranked and plotted against their cumulative floor areas.

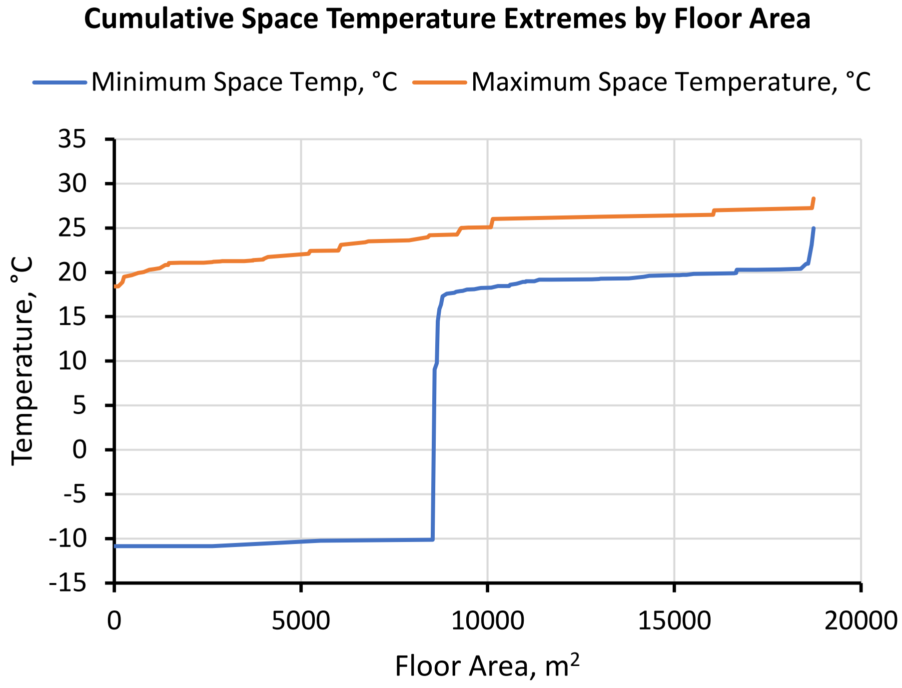

Figure 57 - Cumulative Space Temperature Extremes by Floor Area, Initial Model

The following example is given to help with interpreting Figure 57: There is a total of 10,000 m^2 of floor area that has a minimum space temperature less than about 18°C and a maximum space temperature less than 25°C. The large floor area that has an approximately -10°C minimum temperature is made up by the three car decks, which do not have heating in the model. Ignoring these areas with extremely low minimums, these results still show that during the most extreme conditions of the annual simulation, the space heating system did not meet the heating setpoint of 21°C for nearly all the floor area of the ship. A deviation of about 1°C is common in this simulation. Further investigation into the sizing of the space heaters is given in Section 7.1.1.

7.1 Model Validation

Validation can be defined as comparing model results to a previously unseen dataset, whereas calibration is an iterative process where the model parameters are adjusted to cause alignment with a training dataset [20]. This section describes model validation versus two datasets: space heater sizing and hotel electricity.

7.1.1 Space Heater Sizing

The ship designer is expected to select space heaters according to the estimated net heat loss of the space during design conditions. In the process of selecting space heaters for each room, the ship designer has first-hand knowledge of the expected operation of the ship, as well as heat gains and losses in each room. A comparison of the as-installed space heater sizing to the modeled heating load in each zone provides a useful reference point because it allows a comparison of the modeler's (less-developed) understanding of the physics of the ship with that of the ship designer's (well-developed) understanding. This comparison should not be used for model calibration because its source data represent the ship designer's estimates, rather than measured as-operated data. However, the comparison can lead to a deeper understanding of the model in relation to the physical characteristics and operation of the ship, which can increase confidence in the validity of the model.

The two data sources for this analysis include:

1. Installed heating capacity for each zone, based on the design data; and
2. Modeled design heat load for each zone, where a revised version of the model was made by replacing the space heaters that were sized according to design data (described in Section 5.4.3) with "ideal heaters" that are set to effectively unlimited capacity.

For the modeled case's design conditions, the outdoor temperature is assumed to be -25°C and the supply water temperature for the heating water is set to about 69°C. This uses the same outdoor temperature found in many of the available design data materials. The heating water supply temperature in the modeled case is slightly lower than the design-data-indicated 70°C supply temperature because the model uses a linear outdoor temperature compensation curve that assumes a 70°C supply water temperature at a -26°C outdoor temperature and a 20°C supply water temperature at a 20°C outdoor temperature. This difference is not expected to be meaningful to the results.

Ideal heaters are a tool in IDA ICE that allows heating energy to be added in the exact amounts needed to maintain the temperature setting, without consideration of the hot water loop temperature, room temperature, or thermostat control method. In other words, an ideal heater is 100% efficient, and therefore its energy consumption can be considered to be the net heating energy needs of the room.

A "heating load" analysis was performed in IDA ICE with the -25°C fixed ambient temperature, no solar radiation, and ventilation fans set to operate according to their schedules. The results of the analysis are plotted in Figure 58. The x-axis shows the design heat load with ideal heaters. The y-axis shows the installed heater capacity. These values come from the manufacturer's data, where the rating condition has a supply water temperature of 70°C, a return water temperature of 50°C, and a room air temperature of 20°C. While not exact, these are near the modeled values of 69°C supply water temperature and 21°C room setpoint temperature (from Figure 17), so the results of this analysis are considered to be approximate. Each red point represents one of the 58 zones in IDA ICE. Figure 59 shows the same graph but zoomed in to show the details of the first 10 kW on both the x- and y-axes.

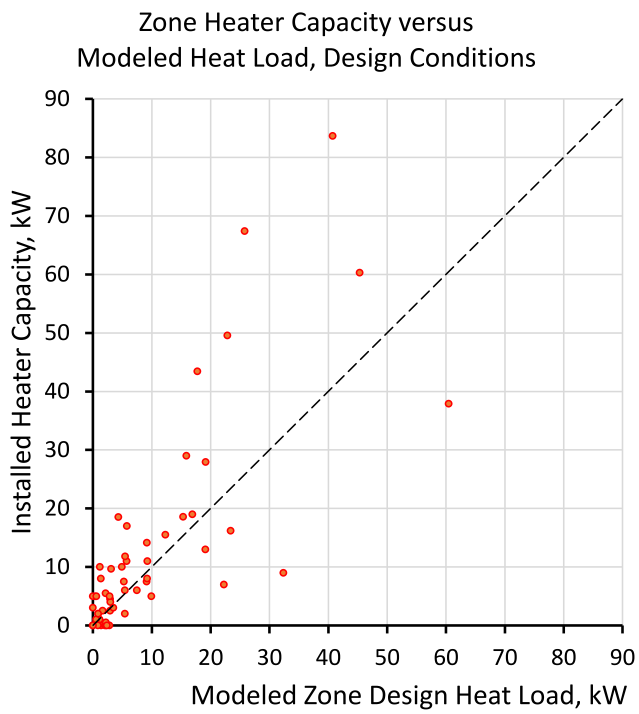

Figure 58 – Zone Space Heater Sizing (All)

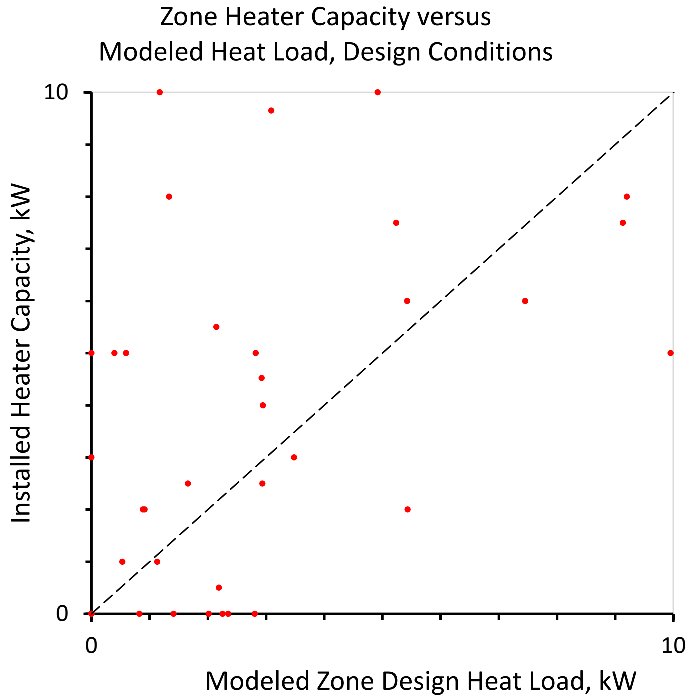

Figure 59 – Zone Space Heater Sizing (<10 kW Subset)

Figure 58 and Figure 59 are useful in determining whether the model considers the as-installed heaters to be over- or under-sized. If the red point is above the dashed line, the installed heater capacity is higher than the modeled design heat load for that zone, meaning the as-installed heater has adequate capacity to meet the design heat load. The majority of the points in both figures are above the dashed line. Heaters are often sized with a safety factor greater than 1. Therefore, the expectation of this analysis was that the model should result in the heaters appearing to be oversized.

There are, however, several points below the dashed line. These indicate that the model considers the as-installed heaters to be under-sized. These areas deserve additional investigation because they call into question the model developer's understanding of the physics of the zone in relation to that zone's design heating load. Given both the complexity and limitations of the model development process, however, perfect adherence to the oversized heater expectation is not necessarily desired. Instead, the goal is to use these deviations as a tool to focus the investigation and build confidence in the model.

In total, 20 of the 58 zones are shown by the model to have undersized heaters. All these zones are shown in Table 16. The IDA ICE zone name is shown here. The first two numbers indicate the zone's deck level. Zones on decks 6 through 8, and some zones in deck 9, are those that represent the hotel portion of the ship. The zones are sorted in descending order of modeled design heat load, given in the second column. The third column provides installed heater capacity. The fourth and fifth columns give an indication of the size of the under-sizing, with a ratio of installed heater capacity-to-design heat load and the size of the heater under-sizing, respectively. The last two columns show the room temperature as estimated by the model and the deviation of the room temperature from the 21°C setpoint.

Table 16 – Zones with Undersized Heaters

IDA ICE Zone Name	Modeled Design Heat Load, kW	Installed Heater Capacity, kW	Ratio: Installed Heater Capacity-to-Design Heat Load	Magnitude of Heater Undersizing, kW	Room Temp., °C	Room Temp. deviation, K
08f-Wheelhouse	60	38	0.63	23	18	- 3
03a-ChangeOfficeStore	32	9	0.28	23	4	- 17
07a-ColorShopBistroPlayArea	23	16	0.69	7	18	- 3
03f-PaintBeer	22	7	0.31	15	2	- 19
03a-SopepChemFire	19	13	0.68	6	16	- 5
03a-HPU	10	5	0.50	5	10	- 11
08f-PS-Cabin-3xOffsm-4xOfflrg-1xSenoff-1xChiefeng	9	8	0.87	1	20	- 1
08f-SB-Cabin-3xOffsm-4xOfflrg-1xSenoff-1xChiefeng	9	8	0.82	2	19	- 2
09f-CrewgymLaundryMaleFemaleStairsHallway	7	6	0.81	1	20	- 1
06a-StairsGentsHallway	5	2	0.37	3	18	- 3
07a-FashionShopStairs	3	3	0.86	0	20	- 1
08m-Cabin-5x1C-ExteriorPort	3	3	0.85	0	19	- 2
09m-Computerroom	3	0	No Heater Installed	3	14	- 7
02f-StoreHousekeeperstore	2	0	No Heater Installed	2	17	- 4
08m-HallNursHcCl	2	0	No Heater Installed	2	17	- 4
08f-Hallwaymid	2	1	0.23	2	18	- 3
08m-HallwaystarboardDispChairsLinenElroom	2	0	No Heater Installed	2	18	- 3
09a-HallwayStairs	1	0	No Heater Installed	1	8	- 13
08m-StairsHallwayMisc	1	1	0.89	0	20	- 1
07f-Stairs	1	0	No Heater Installed	1	18	- 3

The first five zones have a modeled design heat load greater than 10 kW, so they are the focus of this investigation. The zones with lower modeled design heat loads are not expected to appreciably affect the results. Therefore, they are not further discussed.

The wheelhouse zone has the largest indicated under-sizing magnitude, a deficit of 23 kW. The wheelhouse has a dedicated heating system used to blow warm air on the windows to keep them free of frost. This is already included in the model and is sized according to the design data. Review of the model's design calculation shows the heater is in fact providing its maximum heat during the design day. No additional heating devices were found in the design data, so the modeled heater capacity in the wheelhouse model appears to match the design data. Two reasons could explain this deviation. First, it is possible that the ship designer miscalculated the heating load for the wheelhouse and selected an undersized heater. Secondly, the model could somehow be incorrect. The wheelhouse could have heat-producing equipment that is otherwise unaccounted for in the model. Heat producing equipment amounting to 23 kW is significant, however, and this is therefore unlikely to account for the entire deviation. A third explanation is that the window heat loss rate used in the model could be too high. As discussed in Section 5.2.2, the wheelhouse windows are stated in the design data as having an inefficient U-value of 5.8 W/m^{2}K, but with some ambiguity as to the actual value. Therefore, a downward adjustment to the wheelhouse window U-value could improve alignment with the space heater sizing in this zone. Note an adjustment to the internal gains in the wheelhouse is made as described in Section 7.2, so a U-value adjustment may not be necessary. This is discussed further as future work (Section 0).

In Table 16, the zones outside the hotel section of the ship are colored in blue. These areas are expected to have irregular occupancy and therefore have unknown heating setpoints that could be lower than the heating setpoints in the occupied areas. For example, it is possible the heaters installed in these areas are sized to avoid freezing temperatures, rather than providing occupant comfort. This makes these areas less reliable and of less interest when it comes to comparing the heater sizing and the model design heat load. For this reason, they will not be further discussed.

The third listing in Table 16, "07a-ColorShopBistroPlayArea", shows an installed heater capacity-to-design heat load ratio of 0.69, which represents a 7-kW heater under-sizing. This is significant, so it receives further investigation here.

As discussed in Section 5.1, after investigating other options, the large atrium in the aft of the ship on decks 6 and 7 was modeled as two separate zones, with a floor/ceiling connecting them. The three-dimensional model is shown in Figure 60.

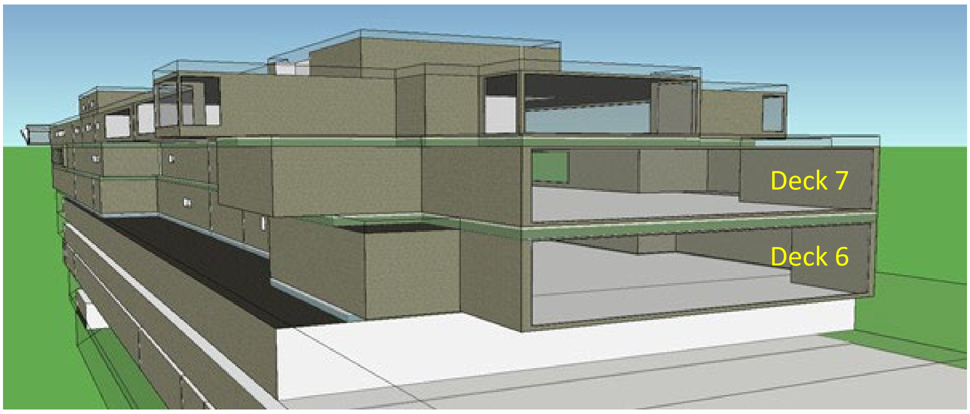

Figure 60 – Aft-section of Decks 6 and 7

Of significant note is that this is not how the ship is constructed. Instead, there is a large opening through this floor/ceiling connection. The ship was designed with convective heaters along the bottom of the tall windows, which means the heaters are only located on deck 6. These heaters have been originally accounted for in the model as occurring on deck 6. The modeled design heat load shows this area of deck 6, "06a-AftShop", has about 26 kW more capacity than needed to meet the design conditions. This easily covers the 7-kW deficit on deck 7. The solution is to adjust the sizing of the heaters in the model to move some of the heating capacity from deck 6 to deck 7 in the aft section of the ship. This adjustment was performed as part of the model adjustments (Section 7.2).

7.1.2 Hotel Electricity

The results shown in purple in Figure 55 represent the modeled hotel demands expected to be met with electricity. The AHU auxiliary, other electric loads, and lighting energy demands sum to approximately 5350 kWh/day, or an average electric load of 223 kW. This demand is relatively consistent throughout the year. These all-electric loads are compared to the measured hotel electricity data (Section 6.2) in this section.

Since the ship's AHU cooling and space coolers use chilled water supplied *in part* by the electric chillers, the modeled cooling energy demand can also *partially* be included in the comparison to the measured hotel electricity data[11]. AHU cooling demands peak, as expected, in the summer months and are zero throughout the rest of the year. Space cooling is almost non-existent, with a consistent 9 kWh/day throughout the year, with the exception of 12, 18, and 18 kWh/day in June, July and August, respectively. This extremely light usage of the space cooling systems indicates a possible issue with the model that should be investigated further. As indicated in Table 9, there is 575 kW of zonal cooling capacity in the ship, compared to 1368 kW of cooling capacity in the air handler units, as shown in Table 11. This would indicate that the ship designers expected the zonal cooling systems to be in higher demand than only 18 kWh/day (an average load of only 0.75 kW) in August, for example. Improper specification of internal gains in the model is expected to contribute to this issue and is discussed further in this section. Proper calibration of the model to electric cooling consumption, however, cannot take place until after thermal energy and cooling system control data are collected and analyzed (Section 0).

Figure 61 shows daily average power versus daily average outdoor temperature for the modeled electric loads and for the regression (Section 6.2) developed from the measured electric demand. The purple points represent the modeled electric loads that include the AHU fan, other equipment, lighting, and 1/3 of the modeled AHU cooling demand. The reason for this adjustment to the modeled AHU cooling demand is to approximate the delivered electrical energy to the electric chillers, assuming they were used to meet the entire cooling demand. To get the delivered electrical energy, the cooling energy demand must be divided by the COP, which is assumed to be 3.0. The black points represent the measured hotel electric demand, as calculated by Equation 1 for outdoor temperatures below 7.5°C and Equation 2 for outdoor temperatures 7.5°C and higher. The number of passengers per trip used in the equations is the same as that assumed in the model (Section 5.3.2): 1600 passengers per trip per day.

[11] The portion of cooling load met by each of the cooling sources is unknown. Section 5.4.2 describes the electric chillers, absorption chiller, and seawater cooling system in more detail.

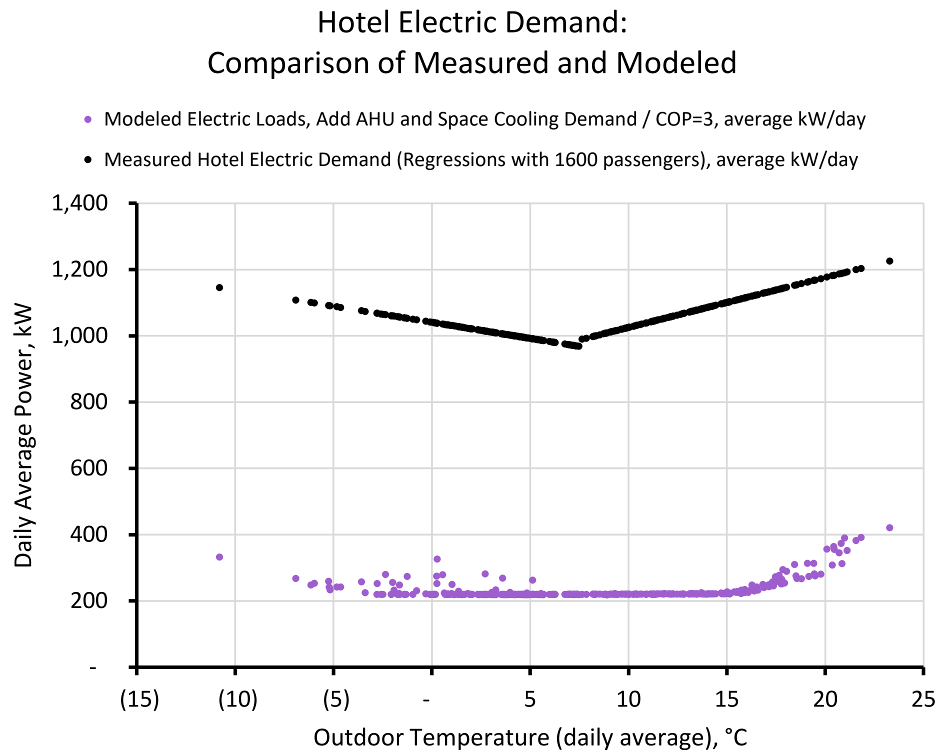

Figure 61 – Comparison of Measured and Modeled Hotel Electric Demand

From Figure 61, it is clear that relatively large adjustments are required to align the model's electric loads with the measured electric demand data. The following four areas deserve attention:

1. *The modeled electric baseload is about 750 kW lower than the baseload based on measured data.* Here, "baseload" is estimated as the power demand at the lowest level of the curve. The baseload of the measured regression is just under 1000 kW and the baseload of the model is just over 200 kW, resulting in a roughly 750 kW difference. This is a significant deviation that indicates the ship has significantly higher electric loads than originally included in the model. This is discussed further below, where model inputs are compared to the design electric load estimates made by the ship builder (from Section 4.2).
2. *Below the changepoint temperature, the modeled electric load should show a relationship to outdoor temperature in order to align with the measured electric demand's slope.* The mostly flat slope of the purple points in Figure 61 below 15°C indicates the original model has a weaker relationship to outdoor temperature than the measured data. The only reason for the increased load during cold outdoor temperatures is because of the electric humidifiers in the air handlers. There is no electric heat was assumed in the original model. There are electric heaters in the hot water system (Section 5.4.1) of the ship, as well as electric heaters in the car decks (discussed below). Inclusion of these loads would improve the match. Since control of the relatively large electric heaters in the hot water system is unknown, addition of this aspect of the model should occur when further data are collected (Section 0).
3. *The modeled changepoint temperature should be lower.* The changepoint temperature of the modeled electric demand (purple) is approximately 15°C, as seen in Figure 61, whereas the measured data showed a changepoint temperature of 7.5°C (Figure 36). If the additional electric loads, as indicated necessary to be added to the model by item 1 above, are located within the heated space, they could cause a reduction in the heating demands of the ship

and therefore a downward shift in the changepoint temperature. The changepoint of the adjusted model and how well it fits the measured data is suggested as a part of future work (Section 0).

4. Above the changepoint temperature, the slope of the modeled electric load aligns fairly well with the measured electric demand's slope. There are, however, many adjustments that need to be made to the accounting of the three types of cooling systems (Section 5.4.2) and how their control systems interact with each other and the ship's cooling needs before this portion of the model is considered validated. Additional validation, including an improved understanding of the actual operation of the cooling systems, should be undertaken if the additional data collection and analysis suggested in future work is performed (Section 0).

Investigation of the first item in the list above continues here. As indicated in Section 0, the model development involved review of a lot of data and it is possible that some electric loads were omitted during that process. A more detailed review was undertaken to understand whether the electric demands in the winter design-day electric loads analysis in Section 4.2 should be included in the model.

Figure 62 compares the winter design-day electric demand estimate in Figure 3 ("weighted average") with an adjusted estimate, and with the modeled average electric demand on an average winter day. The adjusted design-day estimate comes from a more detailed analysis of the winter design-day electric load than that made in Section 4.2. A downward adjustment was made to the original design-day estimate by removing electric loads that are not expected to have an impact on the heated and cooled areas of the ship. This includes all of the loads categorized as "ship systems". It also includes fans that supply air to and exhaust air from the engine room. The reason for this adjustment is that the IDA ICE model is intended to represent the hotel portion of the ship. Therefore, a reasonable goal is for the modeled electric loads to align with the adjusted design-day electric loads estimate. Here, the other hotel equipment category includes the humidifiers in the air handlers.

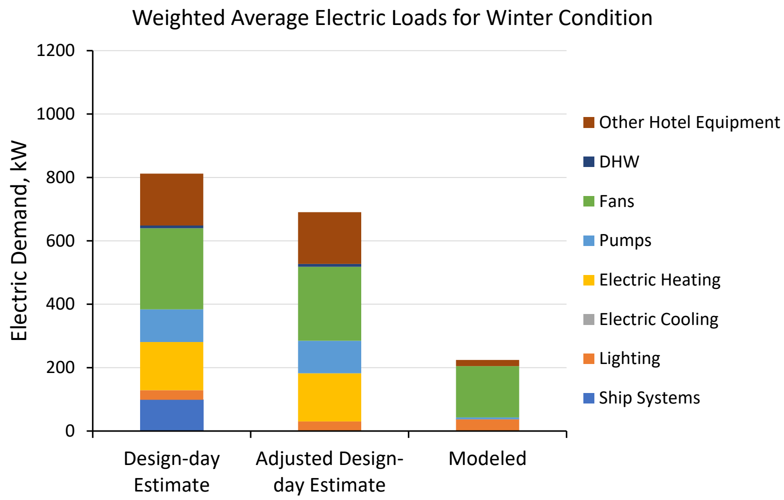

Figure 62 - Average Winter Electric Loads Comparison, Initial Model

There is a large deviation between the modeled and adjusted design-day estimates. Based on Figure 62, it is clear that the modeled estimate is missing electric heating demand, a large portion of pump demand, and a large portion of other hotel equipment electric loads. The modeled fans electric load is slightly lower than the adjusted design-day estimate, while the modeled lighting load is slightly higher than the adjusted estimate. Table 17 shows the electric loads for the goal (adjusted design-day estimate) and the model results. The deviation between them is also shown.

Table 17 - Average Winter Electric Loads Comparison, Initial Model

Load Type	Adjusted Design-day Estimate, kW	Modeled, kW	Model's Deviation, kW
Ship Systems	0	0	0
Lighting	30	37	7
Electric Cooling	0	0.5	0.5
Electric Heating	152	0	-152
Pumps	103	6	-97
Fans	233	162	-71
DHW	9	0	-9
Other Hotel Equipment	164	20	-144
TOTAL	**691**	**224**	**-466**

The model is lacking 466 kW of electric load and it should be adjusted with these loads. Before adding them to the model, however, care needs to be taken to ensure they are added in the appropriate manner. The details of the missing systems should be understood well enough to ensure that the ship's daily electrical load profile is appropriately represented.

A comparison of the average daily load profile for each month for the measured (Section 7.1.2) and modeled electricity demand is shown in Figure 63. The winter months are shown as red lines, the summer months as blue lines, and the spring and fall months are shown as green lines.

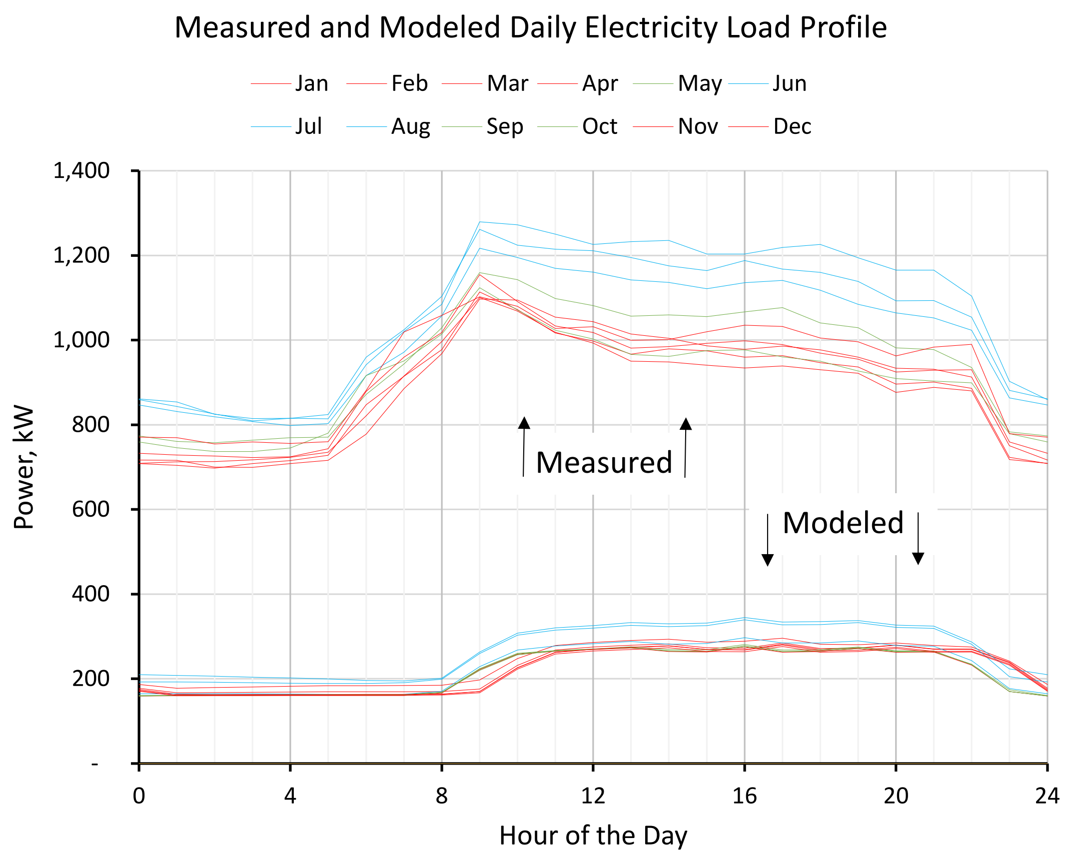

Figure 63 - Measured and Modeled Daily Electric Load Profile by Month, Initial Model

This comparison also shows a significant mismatch between the measured and modeled electricity demand. The modeled daily load profile rises much later in the day, starting around 8:00 am in the summer and 9:00 am in the winter, compared to the measured load profile indicating that the increase in electricity demand starts at about 5:00 am.

The measured curves shown in Figure 63 are un-adjusted, so they do not represent the target load profile for the model. It is also not reasonable to expect the model to match the monthly curves exactly given the expected variability in the operation of the ship. Figure 64 shows the average of the load profiles for the winter months shown in Figure 63, for both the measured and modeled cases, in red. In addition, the concept of a target has been introduced. Target A, the purple dashed line, represents the measured line adjusted by removing the electric loads from the winter design-day estimate that were not expected to influence the heated and cooled areas of the ship. The blue dashed line, Target B, is the daily load profile of the adjusted design-day electric load estimate implied by the ship's schedule.

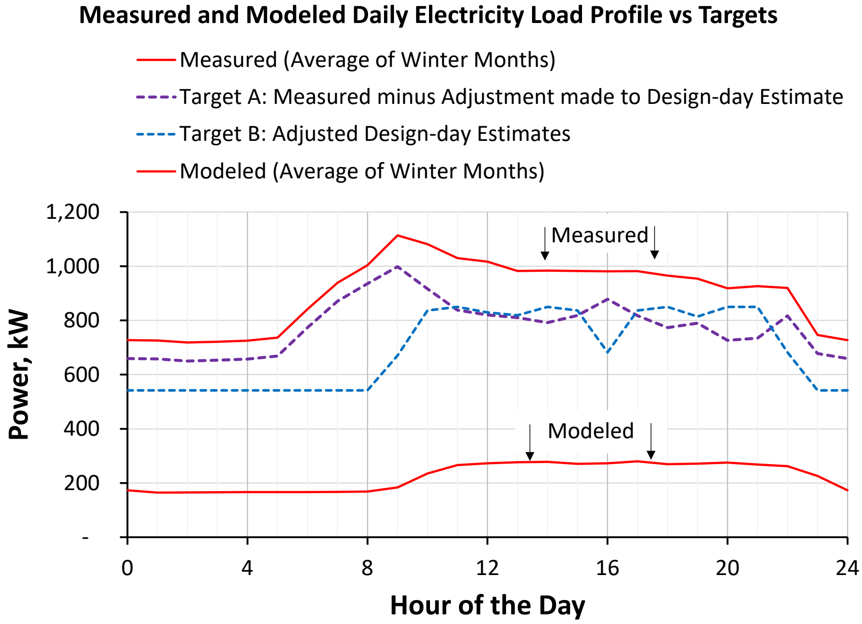

Figure 64 - Average Measured and Initial Modeled Daily Electric Load Profile with Targets

Interestingly, the two targets are very similar to one another starting around 10:00 am, when the ship begins its first daily journey. This gives confirmation that the design-day estimates do a good job of representing the actual ship operation. However, before 10:00, there is a significant difference between Target A and Target B. The higher electric power of Target A, which is based on measured data, shows that the ship is operated with a higher electric demand than expected by the authors of the design-day estimate. This indicates possible energy efficiency opportunities. There is an approximately 100 kW difference in Target A and Target B during the night and early morning hours, indicating electric equipment is being used on the ship that might not be required. In addition, the difference in timing of the morning warmup of the ship indicates a possible shift in the scheduling of electric systems to save energy. These ideas can be explored further in future work.

7.2 Adjusted Model

In this section, the results of adjustments to the model are shown and compared to the model validation results. This is meant to provide an overview of the methodology used to ensure the robustness of the adjusted model, as well as provide an indication of the remaining limitations of the model. More detailed model adjustments are recommended as future work (Section 0), and this would ideally be performed after collection and analysis of additional thermal energy use and operational data. The efficiency measure analyses shown in Section 0 use the model results in this section as the baseline case, with the exception of the galley ventilation rate measure described in Section 8.1.

Based on the results of the model validation (Section 7.1), the following adjustments were made:

- Moved 8 kW of water-borne space heater capacity from deck 6 to deck 7;
- Added electric internal gains (Other Hotel Equipment) of 150 kW as follows:
    - 20 kW internal gains in wheelhouse to represent window heaters;
    - 130 kW of other equipment loads evenly distributed based on floor area; and
- Added 150 kW electric car deck heaters set to a 15°C thermostat setting.

In addition, after these adjustments were made, it was identified that an additional iteration of adjustments was required in order to fix space overheating caused by the additional equipment loads. This process is explained in detail in this section.

Additional electric pump energy was not added to the hot- and cold-water loops. This would require a more detailed mapping of sizing and control of the pumps according to the installations in the ship. The fan parameters were also kept the same. Since the air handler unit fans are currently modeled in detail, it is unclear how closely the design estimates should be followed. Depending on the future needs of the model, the control of the electric internal gains and their exact locations within the ship could be reviewed with detailed ship data. In addition, the control of the car deck heaters and electric water heaters in the hot water system should be verified with sub-metered data if possible.

Table 18 shows the annual energy demand for each major end-use in the adjusted model. The difference between results of the original model (Table 14) and the adjusted model are also shown.

Table 18 - Annual Energy Demand by Major End-Use, Adjusted Model

End-Use	Annual Energy, kWh	Difference between original and adjusted model
Lighting	323,291	0 %
Space and AHU Cooling	307,244	43 %
HVAC Fans and Pumps	1,493,974	-2 %
Electric Heating	896,251	(infinite)
AHU and Space Heating	861,699	-35 %
DHW Heating	564,010	0 %
Other Electric Loads	1,228,341	938 %
Total	**5,674,810**	**40 %**

There is a notable increase of 43% in space and AHU cooling and a decrease of 35% in AHU and space heating. This is caused by the increase in internal gains from other electric loads, which increased by 938% with the addition of the 150 kW. The electric heating has increased from 0 kWh/year in the original model as a result of the addition of car deck heaters. Overall, the annual energy demand has gone up by 40%.

Updates to the air handler energy use are shown in Figure 65 and Table 19. These can be compared directly to Figure 56 and Table 15 for an understanding of the impact of the model adjustments.

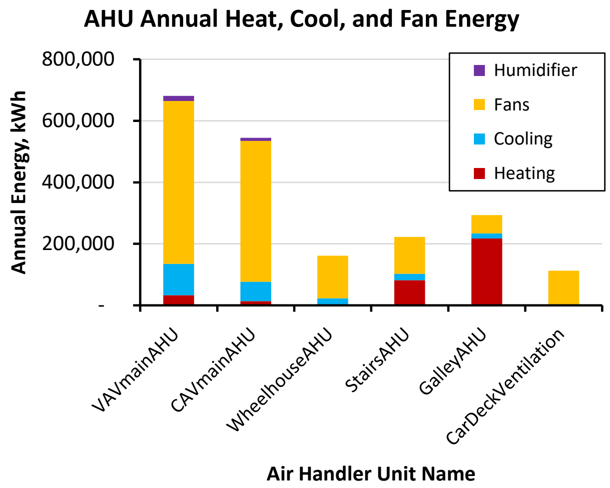

Figure 65 - Annual Results for AHUs, Adjusted Model

Table 19 - Annual Energy (kWh) for AHUs, Adjusted Model

AHU	Heating	Cooling	Fans	Humidifier	Heat Recovery	Cold Recovery
VAVmainAHU	33,157	102,126	529,279	16,620	1,346,189	156
CAVmainAHU	13,426	63,213	457,860	10,438	863,728	665
WheelhouseAHU	1	23,094	138,168	-	658,666	-
StairsAHU	81,641	20,795	120,059	-	448,936	-
GalleyAHU	217,421	16,901	58,933	-	-	-
CarDeckVentilation	0	-	112,721	-	-	-
Total	**345,645**	**226,128**	**1,417,020**	**27,058**	**3,317,518**	**821**

Figure 66 shows, for the first iteration of the adjusted model, the ranked minimum and maximum space temperatures of the zones over the course of the annual simulation, plotted against their cumulative floor areas. The results for the original model, which are shown in Figure 57, are shown here with dotted lines.

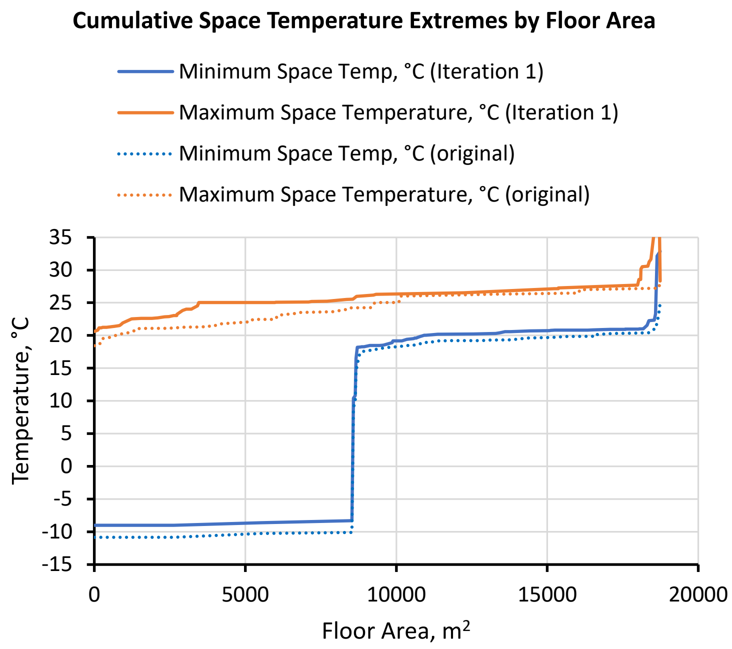

Figure 66 – Cumulative Space Temperature Extremes by Floor Area, Adjustment Iteration 1

The addition of space heaters in the car decks and additional internal gains throughout the ship caused an increase in both the minimum and maximum space temperatures throughout the ship. According to Figure 66, most of the ship zones (ignoring the car decks) are closer to meeting their heating setpoint of 21°C under extreme heating conditions. At the far right of the graph, however, there are some zones that have extremely high minimum and maximum space temperatures. These are storage rooms that do not have active cooling or ventilation, so their internal gains are causing unrealistically high space temperatures. Figure 67 shows the results of the final model, where the internal gains for the storage rooms with abnormally high space temperatures were evenly redistributed to the other rooms in the ship.

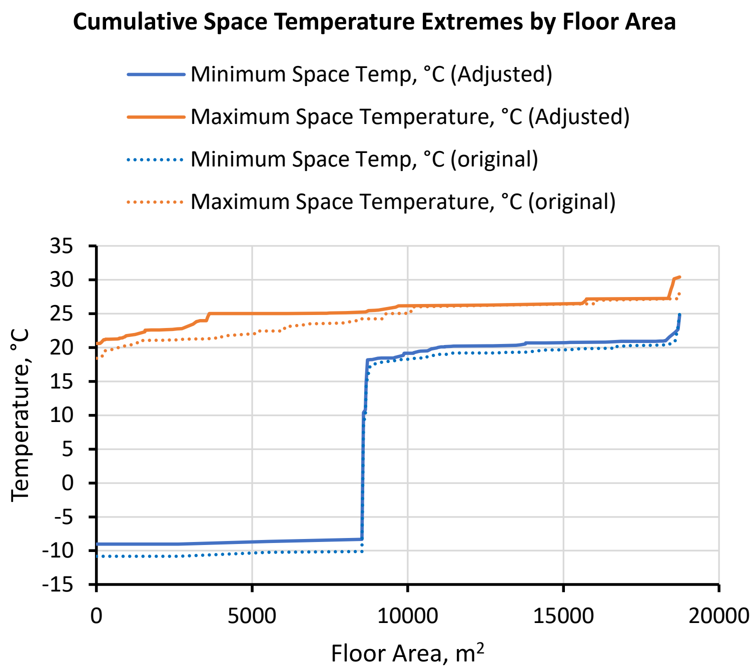

Figure 67 - Cumulative Space Temperature Extremes by Floor Area, Adjusted Model

As seen in Figure 67, the space temperatures are more reasonable in the adjusted model. While additional refinement could be made to the internal gains to improve some of the remaining overheated spaces, this is not expected to have an appreciable impact on the results.

The minimum temperatures for the wheelhouse changed from 17.7°C in the original model to 18.5°C in the adjusted model. The additional 20 kW to represent the window heaters had the expected impact of keeping the space warmer on a cold day. However, there still remains a necessary adjustment to the model in order to keep the wheelhouse at the 21°C heating setpoint temperature.

Similarly, the aft section of deck 7 responded positively to the additional 8 kW of space heaters in the adjusted model. The minimum indoor temperature in that zone increased from 19.7°C to 20.7°C. This is near the setpoint temperature of 21°C and therefore indicates that this was an improvement in the model. Note that the temperature in the aft section of deck 6, where 8 kW of space heaters was removed in the adjusted model, did not see a decrease in minimum space temperature.

The measured winter electricity use and design-day estimates provide a meaningful target for model adjustments. The goal is to adjust the electric loads so they are in closer alignment with the adjusted loads in Table 17, while ensuring that the modeled daily load profile lies near the two targets shown in Figure 64.

Figure 68 shows the average hotel energy demands estimated by the adjusted model, by major end-use, for each month. This can be compared directly to Figure 55, which shows the same results for the original model.

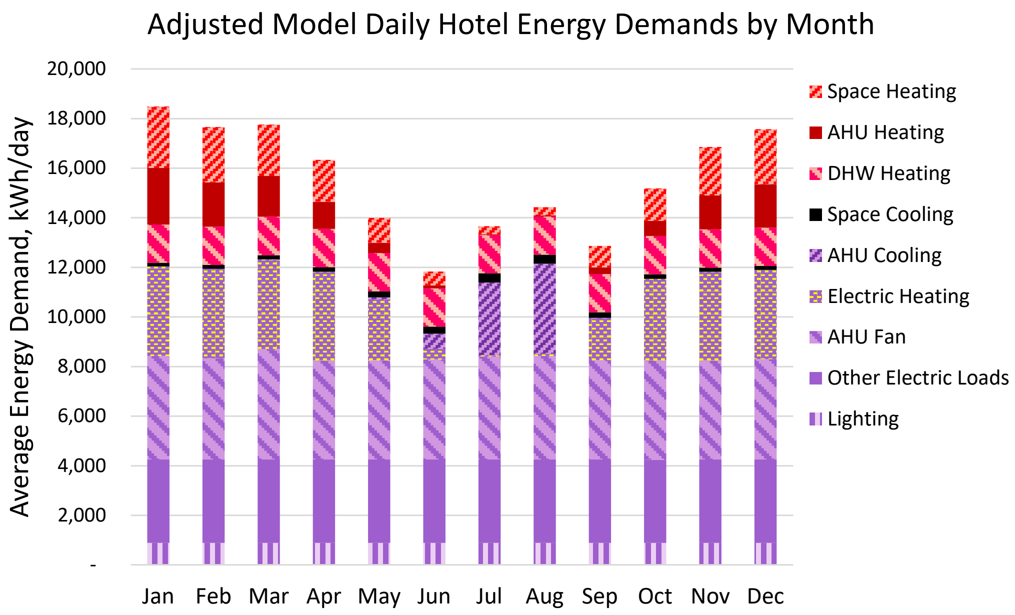

Figure 68 - Hotel Energy Demands by Month, Adjusted Model

The primary differences between the results in Figure 68 for the adjusted model and the results in Figure 37 for the original model are:

1. A significant increase in the other electric loads category, caused by the 150 kW addition;
2. The addition of the electric heating category, which represents the 150 kW car deck heaters;
3. An increase in space cooling, caused by necessary cooling of spaces with more internal gains; and
4. A decrease in space heating, also caused by the additional internal gains.

Points 3 and 4 above highlight the interaction between electrical energy use and thermal energy use. This shows that model adjustments should be made with knowledge about the thermal energy use.

Figure 69 includes the same data as Figure 61: Average daily measured hotel electric demand, as guided by the regression described in Section 6.2, is shown in black points; average daily electric loads from the original model are shown in purple points. In addition, the results from the adjusted model are shown in red points.

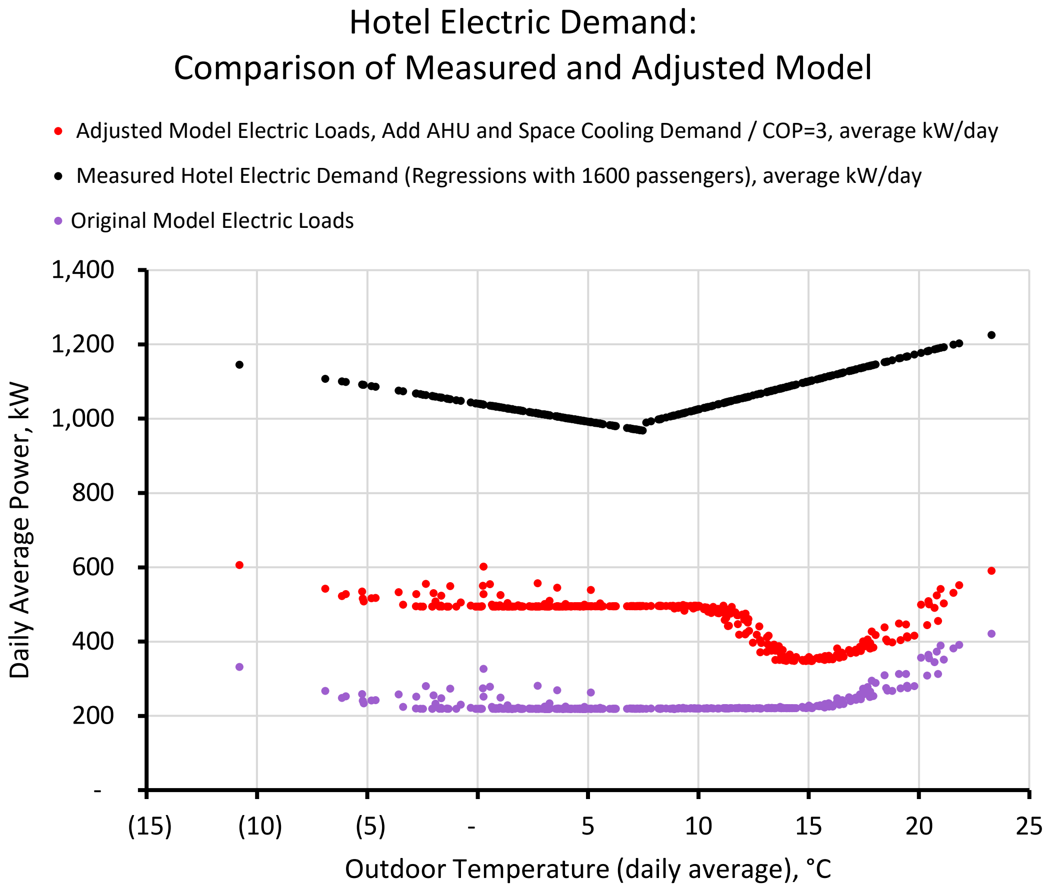

Figure 69 - Comparison of Measured, Modeled, and Adjusted Model Hotel Electric Demand

The goal with the model adjustment is not to move the red points all the way up to the black points because the black points assume full passenger loading and because the electric demand data include loads that are outside the scope of the model (ship systems, as described in Section 7.1.2). However, the results shown in Figure 69 are disappointing for two reasons:

1. *As outdoor temperature falls below 10°C, the adjusted model's daily average power does not increase in the same manner as the measured data.* There is a new relationship to outdoor temperature between 10°C and 15°C. This is the 150 kW of thermostatically controlled electric heaters in the car decks that were added to the adjusted model. They begin to engage as the outdoor temperature falls below the assumed thermostat setting of 15°C. Since the car decks are well-ventilated and have no insulation to the exterior, the heaters end up operating full-time when the outdoor temperature is less than 10°C, in an attempt to meet the setpoint. This does not reflect the curve shown by the black points. This is therefore an area that requires additional research into electric loads that would cause a dependency to outdoor temperature like that shown by the black points.
2. *The addition of the internal gains in the adjusted model did not cause a reduction in the changeover temperature.* The red points still show a changeover temperature near 15°C, rather than the changeover temperature implied by the measured electrical data of 7.5°C. It was expected that adding internal gains would have caused a downward shift in the changeover temperature. This result was thoroughly investigated, and could perhaps benefit from further investigation.

Figure 70 shows the same information as Figure 62 – weighted average electric loads for the design-day estimate, the adjusted design-day estimate, and the original model – but also with the results from the adjusted model.

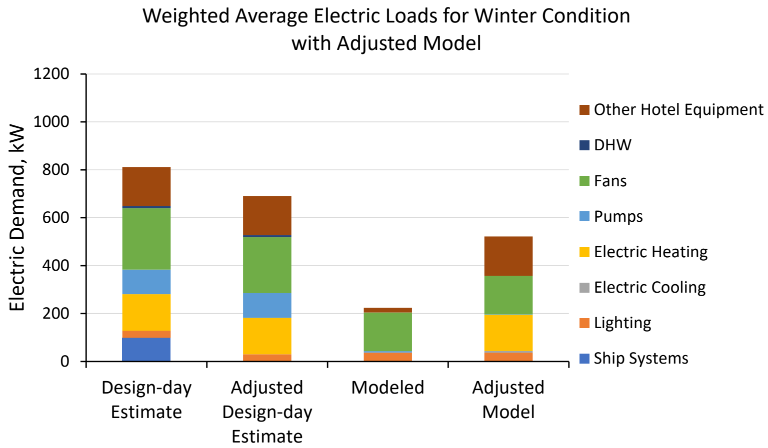

Figure 70 - Average Winter Electric Loads Comparison, Adjusted Model

The adjusted model is much improved over the original model with regard to the total average electric demand. The primary differences are the addition of electric heating from the car deck heaters, and a large increase in the other hotel equipment category. There is still a deficit of approximately 170 kW in the adjusted model's total average electric demand compared to the adjusted design-day estimate, which is the target for the model. The pump category, which makes up about 100 kW in the adjusted design-day estimate, is an order of magnitude lower in the adjusted model, and the fan demand is low by about 100 kW. Both of these issues were explored in significant detail, but without measured data, it would be impossible to properly tune the model. The primary issue is that the pumps and fans are driven by variable speed motors, which causes a large variation in power demand depending on the actual pressure and flow conditions, which are unknown. This points to the fact that the model is not appropriately calibrated with respect to the fan and pump energy consumption, so care must be taken in making conclusions about energy efficiency measures affecting these end-uses.

The daily load profiles from Figure 64 are shown again in Figure 71, with the addition of the average daily electric load profile for winter months from the adjusted model, shown with a thick dashed red line.

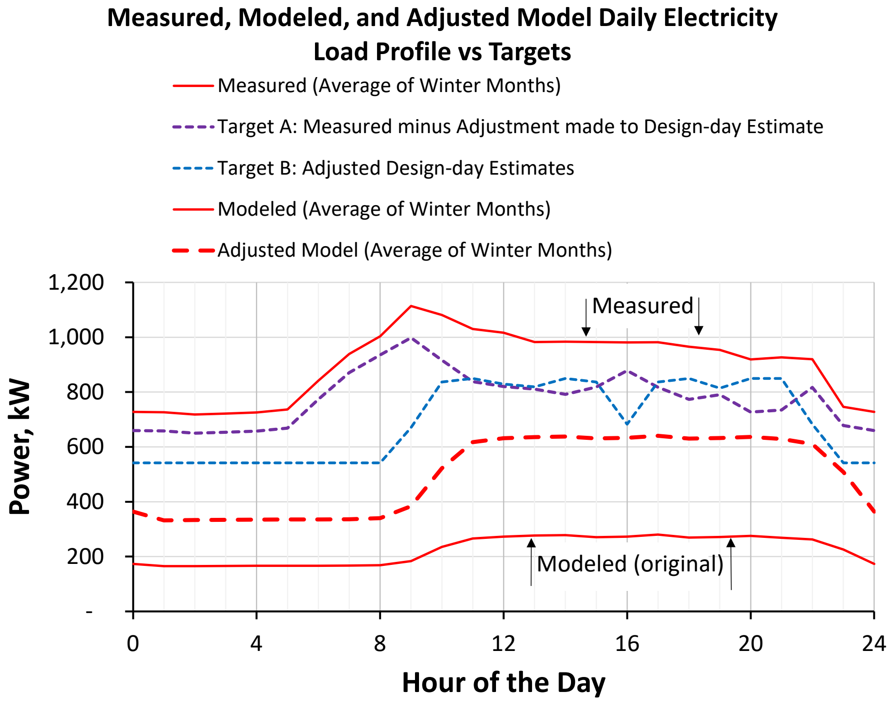

Figure 71 – Average Measured, Initial Modeled, and Adjusted Model Daily Electric Load Profile with Targets

During the daytime hours, the adjusted model has a noticeable increase of nearly 400 kW. In the early morning, prior to the ship's scheduled maneuvering and departure, the adjusted model shows an increase of slightly more than 150 kW in electric power versus the original model. This difference between times of day is because the additional 150 kW of other electric loads have the same schedule as the other equipment (described in Section 5.3.4 as using the lighting schedule shown in Figure 15). There still remains a difference of approximately 200 kW in the daytime and 200-300 kW at night between the adjusted model and the targets.

A significant effort went into a detailed review of the ship design data and the electrical loads data in an attempt account for the missing electric loads apparent in Figure 71. Draft modifications were made to the modeled electrical loads through an adjustment to their scheduling and addition of electrical loads that are known to not be included in the model. Table 20 describes the loads not included in the model and their estimated electrical demand. The majority of these values were from the ship design data, including the winter and summer electrical load estimates. The estimated electrical demand for the galley cooking equipment is from e-mail communication with the ship's first engineer [14].

Table 20 - Electric Loads Not Included in the Model

Load	Estimated Electrical Demand (kW)
Heat Recovery Circulation Pumps	21
Heated Water Circulation Pumps	14
Provisional Cooling Plants	41
Network and Navigation Equipment	34
Engine and Other Ventilation Fans	60
Galley Cooking Equipment	200
Total	**370**

This approach to adjusting the electrical loads brings with it too much uncertainty. One primary issue with making significant modifications to the electrical loads in the model is that the electrical loads can affect the heating and cooling loads of the hotel system of the ship in different, but important ways. For example, adding equipment loads within the conditioned spaces causes a reduction in space heating needs and an increase in space cooling needs. A second, but similarly important issue is that there is not enough data available to represent either the actual loads or their schedules since most of the loads are variable and run at varying times of the day. For these reasons, it would be a mistake to add to the adjusted model line in Figure 71 the total value of 370 kW in Table 20.

In addition, the two electric chillers – which combined represent a rated full-capacity electrical load of 205 kW – and the electric heaters within the hot water system – which combined represent a full-capacity electrical load of 1638 kW – have large enough electrical demands to easily override any scheduling and loading estimates regarding the smaller electrical loads in Table 20. So, while initial efforts to align the modeled electrical loads with the targets were successful on the surface, the recommended final model does not include these adjustments. By leaving the model as is (that is, as represented by Figure 71), this is a useful reminder for users of the model to both be careful with the model results related to the electrical energy use and to seek out additional trustworthy data that can be used to improve the model in a manner that represents the actual ship operation.

A closer look at the model's estimate of the hotel's heating demand can be made through investigation of the daily heat demand signature, which is shown in Figure 72. The average hourly heat demand over the course of a day is shown for each individual month of the year. Winter months are shown in red, where the month with the highest heat demand, January, is shown with a dashed red line. Shoulder months of May and September are shown in green. Summer months are shown in blue, with August shown as a dashed blue line. The hotel heat demand consists of space heating in the zones, air handler unit heating, and domestic hot water heating.

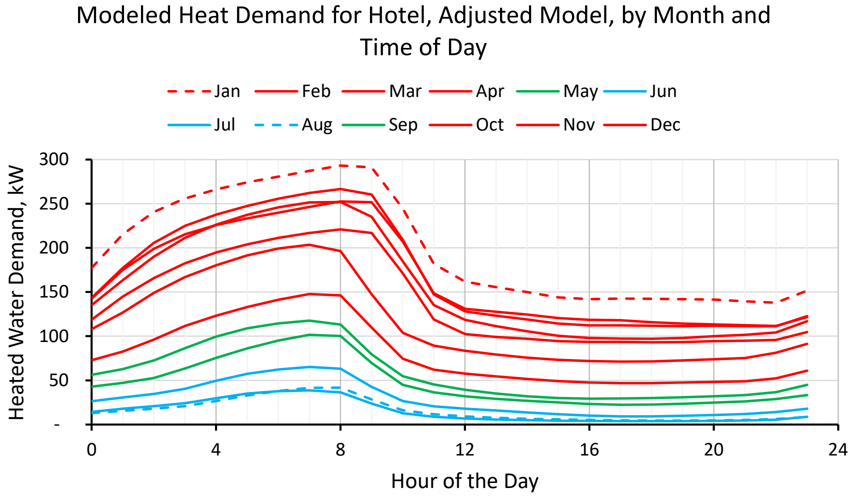

Figure 72 - Daily Modeled Heating Demand, by Month

As expected, the months with the coldest outdoor temperatures have the highest heat demand. Figure 72 also shows that the heat demand is highest at night and in the early morning hours, and then falls during the day when the internal gains from equipment and occupants increase, the daytime outdoor temperature rises, and heat gains from solar radiation increase.

The sum of the energy shown under the January curve in Figure 72 from midnight until 10:00 am is 2.8 MWh. The heated water system's thermal storage tanks have a capacity of just under 4.9 MWh, as described in Section 5.4.1. This means that during the average night in January, about 60% of the thermal storage capacity is required to meet the heating needs of the hotel system before the engines start providing heat and re-warming the storage tanks, according to the model results. There are two nights in the annual simulation where the nighttime heating needs of the hotel exceed 100% of the thermal storage capacity, and four nights where the nighttime heating needs exceed 85% of the thermal storage capacity.

In a similar fashion, the daily cooling demand signature can provide more insight into the model's estimate of hotel cooling loads. Figure 73 shows the cooling signature with the same color coding by month as that shown in Figure 72. Cooling demand represents the combination of space and air handler unit cooling needs.

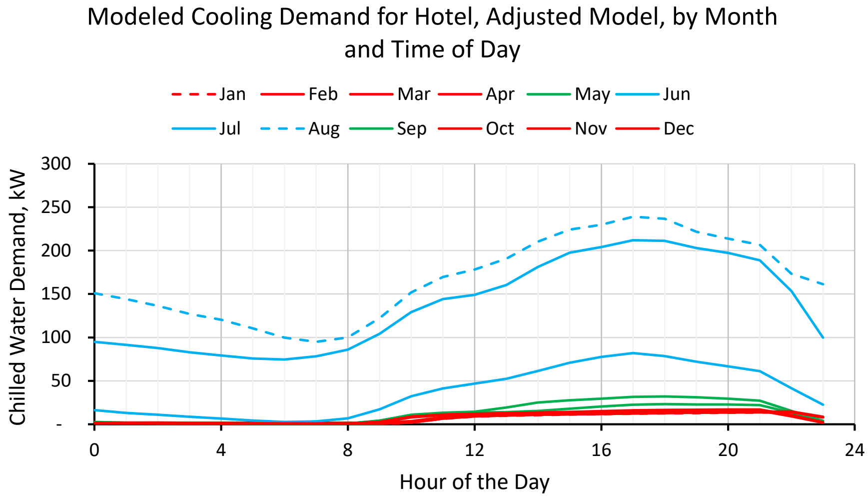

Figure 73 - Daily Modeled Cooling Demand, by Month

The month of August has the highest cooling demand, as shown in Figure 73. The cooling loads for July follow a similar trend, but with a demand about 25 kW to 50 kW lower throughout the day. The month of June is the next line, with significantly lower cooling demand, followed by even lower demands from the shoulder months and winter months. Therefore, only the months of July and August are of significant concern regarding cooling loads in an average year. The model shows the cooling demand peaks late in the day, around 4:00 pm to 7:00 pm. The maximum modeled hourly cooling demand occurs at 8:00 pm on August 1st and is 903 kW, which is just less than the nominal capacity of *one* of the ship's two electric chillers described in Section 5.4.2. This means, according to the model, the ship has more than adequate cooling capacity relative to the estimated cooling loads.

The modeled heating and cooling loads and their daily load shapes described in this section, while based on a model developed with detailed ship design information, have not been validated and would benefit from comparison to energy data collected as described in future work in Section 0.

8 Efficiency Measures

As discussed in Section 2.2, the model development was inspired by possible energy efficiency measures spanning a wide spectrum of the ship's hotel end-uses. The intention is to create a model flexible enough to consider as many energy efficiency measures as possible, both in this report and in possible future analyses. The energy efficiency upgrades suggested here are a result of a combination of the detailed analysis of the available data described in Section 0 and discussions with the ship engineering team during the ship visit (Section 4.5). Using a combined data- and needs-driven approach to selecting efficiency measures allows prioritization of measures that are both relevant and whose savings can be more reliably estimated through comparison with the available data. This process also allows for improved refinement and understanding of the model and its capabilities.

Table 21 provides a summary of the six energy efficiency measures analyzed in this report. A short description of the baseline and efficient-case model are given, along with an estimate of the resultant heating, cooling, and electrical energy savings.

Table 21 - Summary of Energy Efficiency Measures

Measure Description	IDA-ICE Model Description		Hotel Delivered Energy Savings		
	Baseline	Efficient-case	Heating	Cooling	Electricity*
Galley Ventilation Rate	Adjusted Model **edited** with 80% galley ventilation	**Adjusted Model** (50% day/10% night galley ventilation rate)	12 %	3.2 %	2.2 % *
Galley Heat Recovery	**Adjusted Model** (no galley heat recovery)	Adjusted Model **edited** with galley AHU heat recovery with 72% effectiveness	15 %	0.4 %	0.1 % *
Nighttime Ventilation Rate	**Adjusted Model**	Adjusted Model **edited** by reducing VAV, CAV, stairs, and wheelhouse AHU ventilation rates to 20%	15 %	4.5 %	8.3 % *
Recommissioning	IDA-ICE model not used in this analysis. Operational data are used to compare percent energy savings from correcting a faulty inlet temperature sensor on one air-handler		Savings estimates are instructive for only one example and do not represent hotel-wide savings		
Windows Solar Heat Gain	**Adjusted Model** (0.41 SHGC)	Adjusted Model **edited** with window SHGC of 0.61	1.1 %	-4.2 %	0 % *
Absorption Chiller	Adjusted Model loads used in a separate calculation that assumes only electric chillers	Adjusted Model loads used in a separate calculation that includes the absorption chiller	0 %	0 %	0.9 %

* Cooling energy savings will have an unknown, but positive, impact on the delivered electricity energy savings. Because the efficiency and utilization of electric chillers is unknown, the component of delivered electricity energy savings caused by delivered cooling energy savings have not been included here.

Each of the measures in Table 21 are described in more detail in their respective sub-sections below. Their interaction with one another can be inferred from Table 21 based on comparison of the baseline and efficient-case descriptions. For example, the efficient-case for the first measure – galley ventilation rate – is the same as the baseline for three of the other measures – galley heat recovery, nighttime ventilation rate, and windows solar heat gain – so the savings for the galley ventilation rate measure could be added to the other three. However, since the baselines for the other three measures are the same, they are not additive with one another. Also of note is that the IDA ICE model was not used in the same manner for all measures, if at all. For example, with the absorption chiller measure, the loads modeled in IDA ICE were used in a separate calculation because the interaction between the various equipment providing cooling to the chilled water system has not been developed in the model.

The impact of an efficiency measure on the operational costs of a particular ship can vary considerably based on the characteristics of the ship's energy supply and storage systems, and the purpose the ship serves (e.g., journey length, number of passengers, time in harbor, etc.). By providing the delivered energy savings here, these energy efficiency measure results can be more easily adapted to the specific characteristics of other ships. The efficiency measure sub-sections below provide further insight into the measures' impacts specific to the operation of the Color Hybrid. Efficiency measure-caused impacts to each of the three primary delivered energy sources can affect a ship's operational costs as follows:

- *Heated water energy impacts.* In Color Hybrid, heated water energy savings that occur at night in the winter effectively increase the capability of the thermal storage tanks to provide heat to the ship. This is because the ship utilizes the thermal storage tanks when docked at night in Sandefjord, when the diesel engines are not generating waste heat. The crew reported that on cold winter nights, the electric resistance heaters often need to be used to maintain adequate temperatures in the heated water system [13]. Therefore, heated water energy savings that occur on winter nights would offset the use of electric shore power to run the electric resistance heaters. The annual shore power savings would depend on how often the savings coincide with the use of electric resistance heat to supplement the stored engine waste heat. On the other hand, given the excess of available heat from the engine exhaust heat recovery system, daytime heated water energy savings would not impact the Color Hybrid as it is operated today. However, in the design process of other ships, namely those without excess waste heat from the burning of fossil fuels, lower heating demands throughout the entire day could be equally beneficial depending on the specifics of the storage and heat generation systems on board.
- *Chilled water energy impacts.* In the Color Hybrid, the impact of cooling energy demand savings on the operational costs of the ship also depends on timing of the savings. In the winter, the seawater heat exchanger is used as the sole-provider of cooling. This means the cooling demand is met for nearly no cost. In the heat of the summer, however, when the temperature of the seawater is too high to provide useful cooling, the cooling energy demand will have a more direct impact on the operational cost. A reduction in the cooling energy demand during these conditions will cause a reduction in the electricity supplied to the electric chillers based on the operational COP of the marginally operating chiller. In other ships, the cooling equipment will likely have different operational patterns and efficiencies, so these would need to be considered on a case-by-case basis to ascertain the operational cost impacts.
- *Electricity impacts.* The Color Hybrid is supplied electricity through a combination of the shore power in Sandefjord and the electricity generated through the diesel-powered generators on board. Therefore, a reduction in electricity demand will cause a varying impact on operational costs depending on whether the savings coincide with a connection to shore power or with generation of electricity on board. The cost of the shore power varies with time of day and with time of year, while the cost of onboard electricity generation varies with the cost of diesel and the efficiency of the diesel-powered electricity generation. The complexity of the Color Hybrid is compounded by the fact that the ship has a large battery storage system, which effectively shifts diesel electricity generation to shore power up to the storage limit of the battery. These complexities are not analyzed further in this report, but it is important to remember that all savings in delivered electricity on the Color Hybrid result in some form of operational cost savings. On other ships, with under-utilized renewable electricity generation sources, this may not be the case. Therefore, a case-by-

case operational cost analysis should also be performed on electrical energy demand savings.

The chosen measures represent only a small subset of the possible efficiency measures that could be explored with the model. As needs arise and as more data are made available, using the model to explore additional efficiency measures would likely result in both a better understanding of how to improve a ship's hotel energy efficiency and enhancements to the model itself.

8.1 Galley Ventilation Rate

The galleys are equipped with variable speed ventilation control, both manually adjustable and adjustable from the control center. Since the galley air handler units are not equipped with heat recovery, their heating energy demand is expected to be quite high during the winter. An important consideration for operation of the ship is to always maintain a galley ventilation flow rate that meets the needs of the space and no higher. The operational data suggest the galley ventilation rates are often higher than necessary, so an analysis of the energy impacts of reducing the galley ventilation rate is made in this section.

As described in Section 5.4.4, the galley AHU ventilation rate is assumed in the original and adjusted model to operate at 50% flow during employee occupied hours and 10% during the remaining hours of the day. The air handler unit operational data show consistent operation at 80% flow or higher, without much deviation throughout the day and night. Since the data are only available for a four-month period from the summer and fall of 2022, it is not clear whether this operation represents normal ship operating conditions. The high ventilation rate witnessed in the data is likely not required to meet the ventilation needs of the galleys, especially at night, when no cooking activities take place. Therefore, the model is kept at the 50%-day and 10%-night flow rate assumption to establish a reasonable baseline condition for other measures tested with the model. For this measure, however, the baseline condition uses the higher 80% flow rate and the efficient case uses the 50%-day and 10%-night flow rate assumptions, which are assumed to be more closely aligned with the average ventilation needs of the galleys over time.

Figure 74 and Table 22 show the results of the simulation for both the baseline and efficient cases, in terms of annual energy use by the galley air handler units. The efficient case for this measure aligns with the adjusted model, which means the efficient case values shown in Figure 74 match the galley AHU values shown in Figure 65.

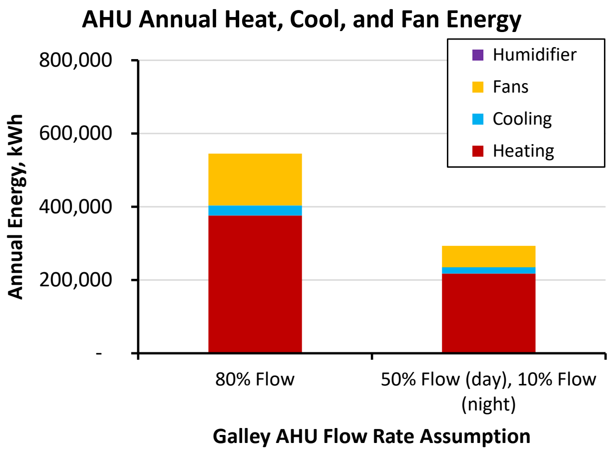

Figure 74 – Annual Model Results for Galley AHUs, by Flow Rate Assumption

Table 22 – Annual Energy (kWh) for Galley AHU, by Flow Rate Assumption

Ventilation Assumption	Heating	Cooling	Fans
80% Flow	376,124	27,310	141,630
50% Flow (day), 10% Flow (night)	217,421	16,901	58,933
Savings	*158,703*	*10,410*	*82,697*

The annual total energy use of the galley air handler units is reduced by nearly 50% with this measure. Heating water energy demand is reduced by 42%, cooling energy demand is reduced by 38%, and fan energy use is reduced by 58%. The annual air handler heating energy use for the entire ship is reduced by 32%.

A better understanding of the impact of the heated water savings can be made by reviewing the hourly average time signature of the savings. Figure 75 shows the average heated water savings for each month in the simulation. Winter months are shown with red lines, summer months in blue, and shoulder months in green. The months with maximum and minimum demands – January and August, respectively – are shown with dashed lines.

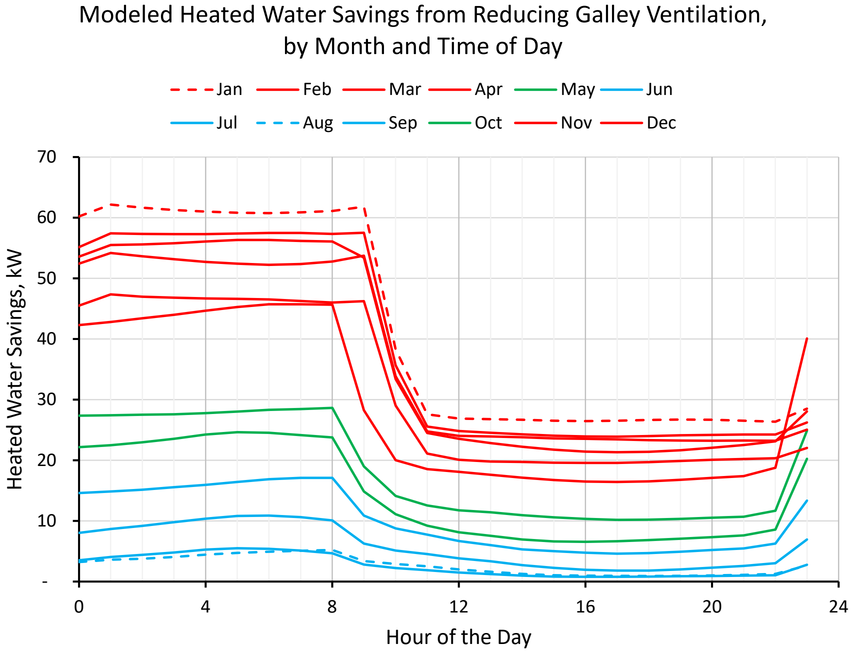

Figure 75 – Daily Modeled Heating Savings from Reducing Galley Ventilation, by Month

Heated water system savings, shown in Figure 75, are highest at night and in the winter when the Color Hybrid needs to make the most of its thermal energy storage tanks. The 60-kW savings during the average night in January amounts to 540 kWh per night, which represents over 11% of the storage capacity of the thermal storage tanks. Therefore, this measure should have a meaningful impact in reducing the operation of the electric heaters in the heated water system.

The purpose of this analysis is not to provide exact energy savings, but instead to show that the magnitude of savings can be relatively large. The specific savings will depend heavily on how much the ventilation rate in the galleys can be reduced, for how long, and under which outdoor weather conditions. It is important to point out that close monitoring and adjustment of the galley ventilation rates should ensure that minimum indoor air quality levels are always maintained and never traded off to achieve higher energy savings.

8.2 Galley Heat Recovery

As shown in Figure 65, since the air handlers serving the galleys do not have heat recovery, the high ventilation airflow rates in the galleys require a significant amount of heat. This presents itself as an obvious energy efficiency opportunity. The IDA ICE model is used to estimate the energy savings, and potential issues with a galley heat recovery measure are discussed in this section.

The exhaust hoods in galleys tend to have large amounts of cooking particles, moisture, and heat in their airstreams. These conditions would quickly foul a traditional energy recovery wheel, like those used in the VAV and CAV air handlers in the ship. In galley applications, the preferred choice is a glycol run-around loop with a heat recovery coil specifically designed to handle the harsh conditions in cooking exhaust air streams. While the air handler serving the galleys in IDA ICE could be modified to use a glycol run-around loop option, this increase in complexity is not expected to improve the certainty of the results. Therefore, the standard simplified air-to-air heat exchanger was used to simulate this efficiency measure. An air-side effectiveness of 0.72 was chosen, which matches the value used for the VAV air handler unit. While this does not represent any particular product on the market and is likely higher than expected in a typical glycol run-around loop system, it is important to also mention that the heat energy in the extract airstream is expected to be higher than that shown in the simulation. This is because the heat rejected from the cooking equipment is not included in the model. IDA ICE does not lend itself well to modeling the localized effect of cooking appliances, with high heat rates and exhaust fans within close proximity. Instead, if the galley cooking appliances' electric loads were added to the galley zones, they would improperly impact the entire zone rather than the local area around the cooking appliance and its exhaust hood. Therefore, the assumptions here are expected to lead to reasonable energy savings results, given the uncertainties.

The results of the simulation, specific to the air handler units, are shown in Figure 76 and Table 23. The simulation assumes that the ventilation system operates at 50% capacity during the day and 10% capacity at night, in both the baseline and efficient cases. These results can be compared directly with Figure 65 and Table 19, respectively.

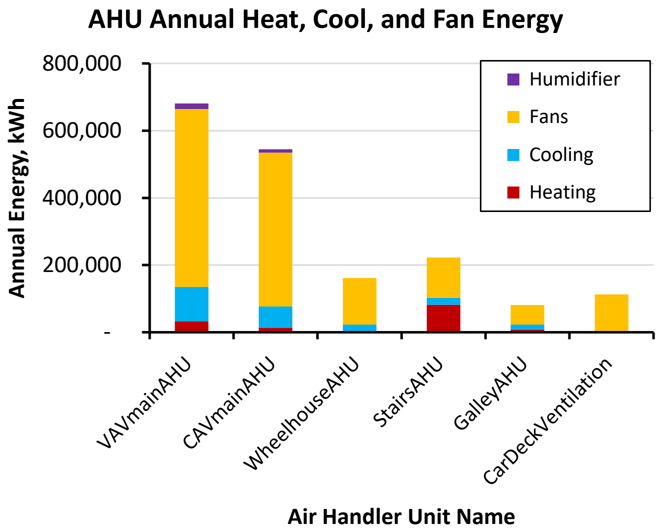

Figure 76 – Annual Model Results for AHUs, with Galley Heat Recovery Measure

Table 23 – Annual Energy (kWh) for AHUs, with Galley Heat Recovery Measure

AHU	Heating	Cooling	Fans	Humidifier	Heat Recovery	Cold Recovery
VAVmainAHU	33,156	102,124	529,280	16,621	1,346,197	156
CAVmainAHU	13,427	63,213	457,860	10,436	863,728	665
WheelhouseAHU	1	23,098	138,168	-	658,627	-
StairsAHU	81,639	20,794	120,059	-	448,966	-
GalleyAHU	7,854	15,523	57,634	-	209,725	1,100
CarDeckVentilation	0	-	112,721	-	-	-
Total	**136,076**	**224,751**	**1,415,722**	**27,057**	**3,527,242**	**1,921**

Immediately noticeable when comparing the results in Figure 76 with those of the adjusted model in Figure 65 is the large reduction in the annual heating energy for the galley air handler unit. The heat recovery unit caused a 96% reduction in modeled annual heating energy use. The heating energy consumption for all air handlers on the ship was reduced by 61% with this measure.

Figure 77 shows the hourly average time signature of the savings, averaged for each month in the simulation.

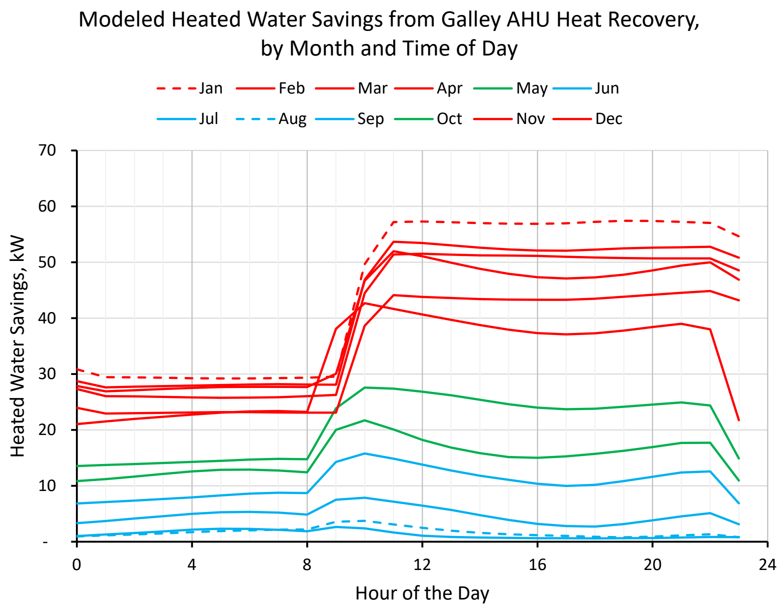

Figure 77 – Daily Modeled Heating Savings from Galley AHU Heat Recovery Measure, by Month

The results confirm the impact of the underlying assumed ventilation schedule, with higher savings during the higher daytime ventilation rate. However, even with only 10% of rated ventilation airflow during the night, the average nighttime savings in January are significant – nearly 30 kW, or 270 kWh per night. These energy savings amount to more than 5% of the storage capacity of the thermal storage tanks.

While these savings estimates verify that this exhaust air heat recovery measure is promising, it is important to remember that the Color Hybrid likely would not be able to implement this measure without significant work. The measure would require remodeling of the exhaust duct systems and installation of new piping to bring heat from the exhaust air heat exchanger portion of the run-around loop heat recovery unit back to the air handler's inlet air streams on Deck 9. It is also critical that fire protection measures are taken with any design of kitchen exhaust ventilation systems. This is, however, a useful measure for ship designers to keep in mind when considering the ventilation requirements of galleys in new ships since the potential energy savings are so high. A similar measure could be considered for the air handlers serving the stairs, which also do not have ventilation heat recovery.

8.3 Nighttime Ventilation Rate

Based on the available operational data, the Color Hybrid ventilation systems run throughout the night with no reduction in the flow rate of the CAV systems and only minimal reduction in flow rates of the VAV systems. Crew members remain on board in their cabins at night, but there are no passengers. The fans in the air handlers have variable speed control, so nighttime turndown of each air handler's flow rate should be achievable with minimal effort, using the existing equipment. The energy impacts of reducing the ventilation rates of the unoccupied spaces are explored further in this section.

The ship classification rules call for a minimum fresh air supply per person. However, there are no requirements for ventilating unoccupied spaces [18]. Norwegian regulation TEK17 provides minimum ventilation rates for unoccupied commercial buildings as a means for ventilating air pollution from materials within the space. Where low-emitting materials are used, the guidance provided is 0.7 m^3/h of fresh air per m^2 floor area [21].

For the modeled savings analysis, it is assumed that the VAV, CAV, Stairs, and Wheelhouse air handlers can be turned down to 20% of their rated flow[12] between midnight and 9:00 am. The exception is the two CAV air handlers that serve the crew areas, AC-2.5 and AC-3.5, which are not assumed to be turned down[13]. This reduced airflow rate still results in all zones receiving fresh ventilation air at a rate above the 0.7 m^3/h per m^2 floor area guidance specified in TEK17, with the exception of a 37 m^2 stair and hallway zone with only 0.56 m^3/h per m^2 floor area. This exception should be insignificant with respect to the savings estimates results shown here, but it serves as a reminder that implementation of this measure should include measures to ensure adequate ventilation rates for all zones.

The modeled energy demands of each type of air handler for the efficient case of this measure are shown in Figure 78 and Table 24. The adjusted model, whose corresponding results are shown in Figure 65 and Table 19, serves as the baseline for this measure.

[12] Savings measures for the galley air handlers, which are assumed to have ventilation rates turned down at night in the original and adjusted models, are discussed separately in Sections 8.1 and 8.2.

[13] For the efficient case model, a new air handler specific to the AP-3.1 CAV air handler was added to the IDA ICE model in order to distinguish the control of the CAV air handler serving the public zones from the two air handlers serving the crew zones.

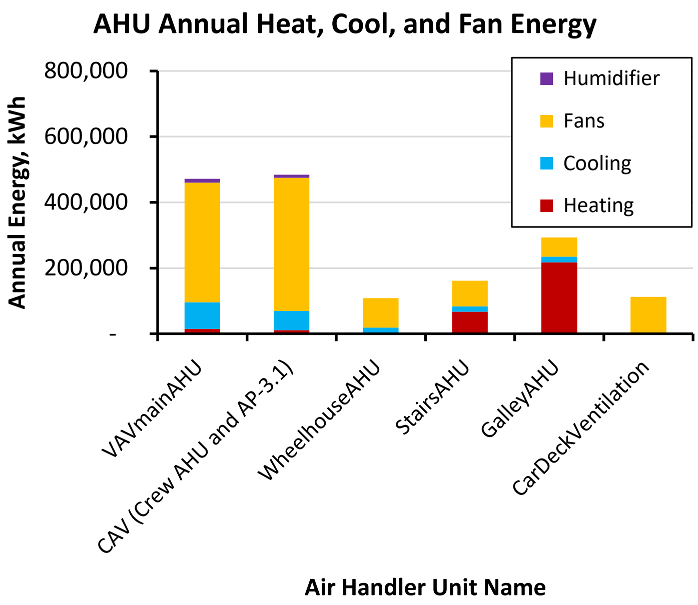

Figure 78 – Annual Model Results for AHUs, with the Ventilation Night Setback Measure

Table 24 – Annual Energy (kWh) for AHUs, with the Ventilation Night Setback Measure

AHU	Heating	Cooling	Fans	Humidifier	Heat Recovery	Cold Recovery
VAVmainAHU	14,963	80,934	363,802	11,953	956,484	44
CAV (Crew AHU and AP-3.1)	10,927	58,715	405,319	9,420	776,142	504
WheelhouseAHU	8	18,682	89,760	-	425,998	-
StairsAHU	67,258	16,331	77,998	-	272,925	-
GalleyAHU	217,403	16,899	58,929	-	-	-
CarDeckVentilation	0	-	112,721	-	-	-
Total	**310,558**	**191,561**	**1,108,527**	**21,373**	**2,431,549**	**548**

Annual energy savings for all air handlers on the ship are as follows:

- Heating water demand savings are 10%;
- Chilled water demand savings are 15%;
- AHU fan electricity demand savings are 22%; and
- Humidifier electricity demand savings are 21%.

Figure 79 shows the hourly average time signature of the heated water demand savings, averaged for each month in the simulation.

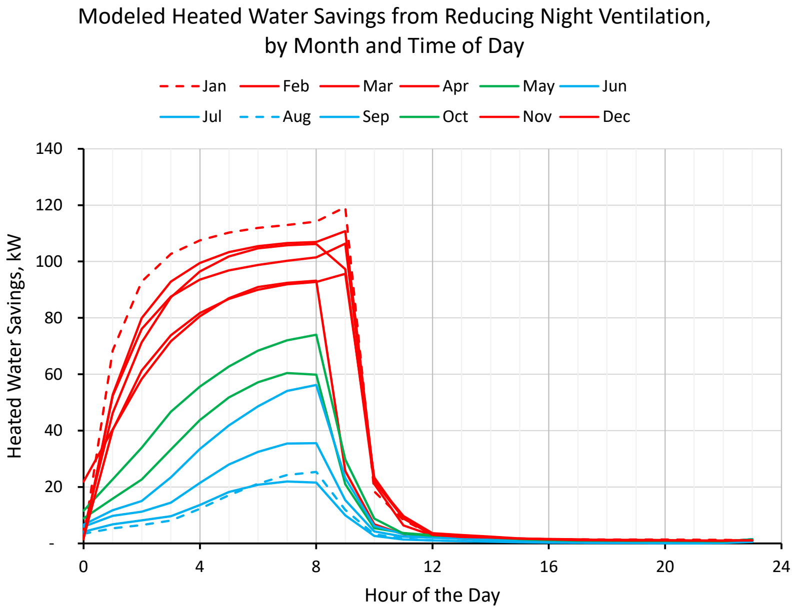

Figure 79 – Daily Modeled Heating Savings from Ventilation Night Setback Measure, by Month

The model results show significant nighttime savings in the winter months. The heated water savings of approximately 100 kW, or 900 kWh per night, in January represent more than 18% of the storage capacity of the thermal storage tanks on the Color Hybrid.

Turning down the ventilation rates in this manner is a low-effort way to save a significant amount of energy with existing equipment on the Color Hybrid. Additional attention could be given to the resultant ventilation rates in order to further improve savings, since a constant reduction in flow across all zones, as suggested here, leaves the zones with higher levels of daytime occupancy with higher night ventilation rates than necessary.

With this measure, the utmost care should be taken to ensure the indoor air quality is not adversely affected. This is especially important in areas that do not have low-emitting materials or that have other reasons to expect higher than normal levels of indoor air contaminants. The ventilation rates in these zones should be considered and adjusted accordingly.

8.4 Recommissioning

The error found with the inlet air temperature sensor for AHU AP-3.3, described in Section 6.4, highlights the importance of an ongoing methodical recommissioning program for a ship like the Color Hybrid with its many complex systems. Such errors can cause comfort issues, compounding operational difficulties, and energy efficiency degradation. The energy efficiency improvement caused by recommissioning can be very difficult to quantify, but the data analysis described in Section 6.4 lends itself to an example energy savings calculation, which will be described here.

The energy impact of the temperature sensor error cannot be calculated in detail without additional knowledge of the humidity levels of the inlet and extract air. However, an estimate of the scale of the sensible heat component of the cooling energy impact can be made by comparing the supply air temperature deviation to the temperature lift caused during improper operation of the energy recovery wheel. Figure 80 shows a histogram of the sensible heat impact in 0.5 K bins of supply air temperature deviation. Also shown is the count of hours in each bin. The data in this graph have been filtered to include only cases where the outdoor temperature is greater than 15°C. The sensible energy impact, which can be interpreted as an additional sensible cooling load, is defined as the average temperature increase caused by the improper use of the recovery wheel as a percentage of the total temperature difference between the inlet air and supply air for AP-3.3.

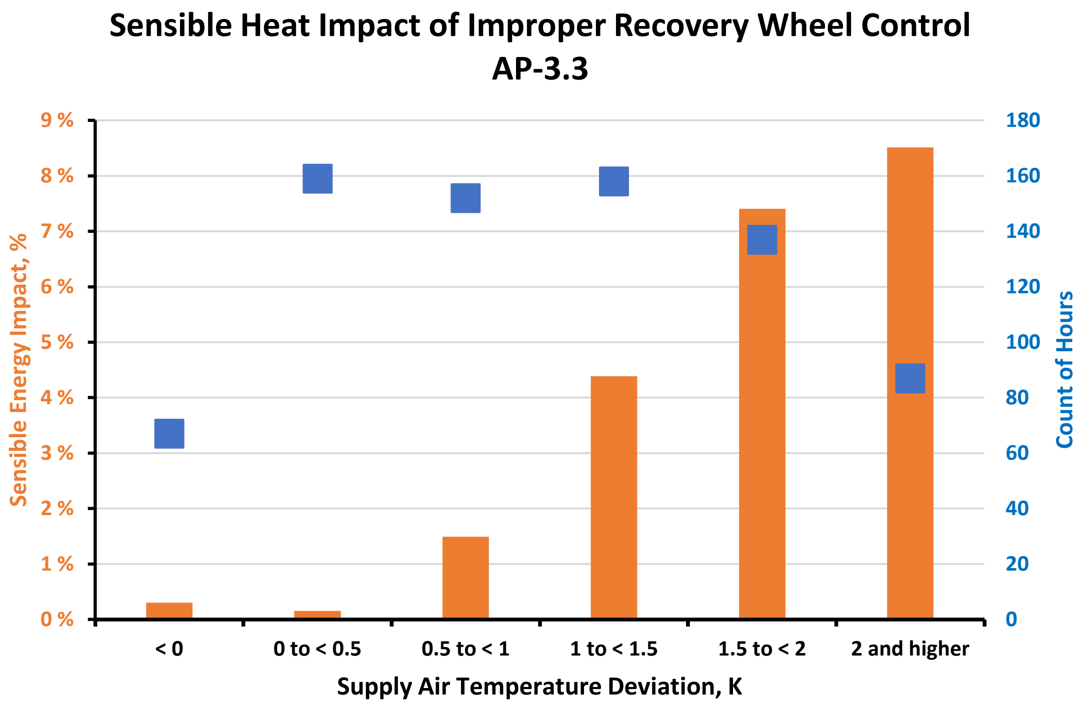

Figure 80 – Sensible Heat Impact of Improper Recovery Wheel Control, AP-3.3

At the conditions where the supply air temperature deviation was above 1.5 K, having a correct inlet air temperature sensor on AHU AP-3.3 could have saved over 7% of the air handler's sensible cooling energy, according to Figure 80.

In addition to the sensible cooling energy savings, there are also implications with a faulty inlet air temperature sensor during the heating mode. During a call for heating, the air handler would engage the recovery wheel when the temperature difference between the inlet air and the extract air appears beneficial to do so. An artificially high inlet temperature sensor would cause the recovery wheel to be disengaged during conditions where it would otherwise be beneficial to recover heat from the exhaust airstream. These additional savings are not calculated here but should be acknowledged when considering this efficiency measure.

8.5 Window Solar Heat Gain

This energy efficiency measure analysis explores the energy impacts of using windows with different solar heat gain coefficients (SHGC). The issue with local thermal discomfort near the windows in the summer is the primary reason for this analysis, but since SHGCs can also impact heating energy in the winter, this analysis explores the annual impacts. The ship documentation did not give a clear indication of the SHGC for the windows, so the adjusted model was assumed to have windows with a SHGC of 0.41. Since that is a fairly low SHGC given the windows' U-value, a model with an assumed SHGC of 0.61 will be used as a comparison. Changing only the SHGC does not represent a realistic measure for the Color Hybrid because there are no products that affect only the SHGC. However, this analysis is intended to help understand the sensitivity of the energy use results related to the original model's SHGC assumption, as well as convey the capabilities of the model to provide insight into the local thermal discomfort caused by solar gain through the windows.

Table 25 shows the model results with the higher SHGC of 0.61, in terms of annual energy demand by major end-use, along with the energy savings compared to the adjusted model with a SHGC of 0.41.

End-Use	Annual Energy, kWh	Savings from Increasing Window SHGC
Lighting	323,289	0 %
Space and AHU Cooling	320,202	-4.2 %
HVAC Fans and Pumps	1,493,680	0 %
Electric Heating	896,233	0 %
AHU and Space Heating	846,128	1.8 %
DHW Heating	564,010	0 %
Other Electric Loads	1,228,330	0 %
Total	**5,671,872**	**0.05 %**

Table 25 – Annual Energy Demand by Major End-Use, Increased Window SHGC

The high-level results clearly show that the SHGC impacts the "space and AHU cooling" category and the "AHU and space heating" category. Cooling savings are negative, meaning that the cooling energy demand increases with increasing SHGC. The opposite relationship holds for heating energy demand and SHGC. These relationships are as expected, since a lower SHGC means less solar energy can make its way through the glazing. The impacts of this change in SHGC on the other end-uses are insignificant. A closer look at the annual heating and cooling loads disaggregated by zonal loads and AHU loads is shown in Figure 81.

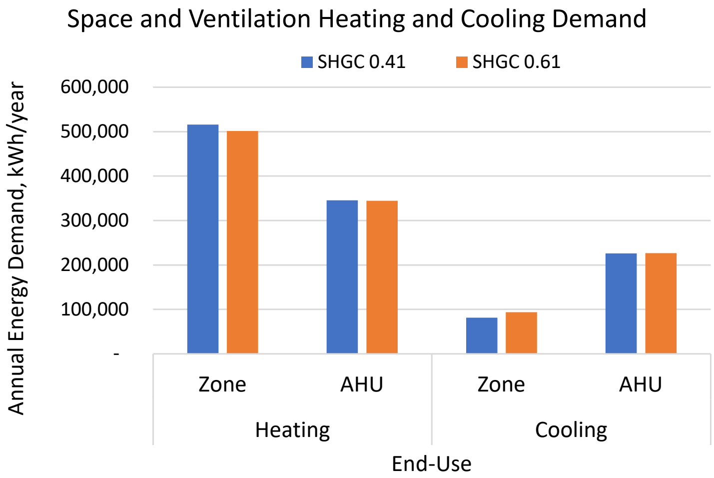

Figure 81 – SHGC Impact on Space and Ventilation Heating and Cooling Annual Demand

Figure 81 shows that the heating and cooling load impacts occur at the zonal heaters and cooling units, and not in the air handlers. This is as expected since the zonal conditioning equipment is used to maintain the space temperatures. It is also evident that this 0.20 increase in SHGC did not make a large impact on the zonal heating load, but the increase in the zonal cooling load is more evident.

As discussed in Section 0, on the Color Hybrid, heating demand savings during cold winter nights help the thermal storage tanks supply enough heat through the night, thereby reducing the need for backup heating. Figure 82 shows, averaged for each month in the simulation, the hourly average time signature of the heated water demand savings from increasing the SHGC.

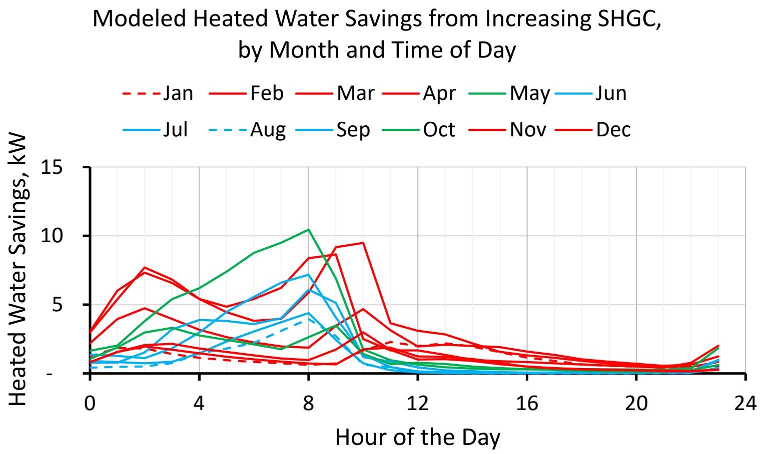

Figure 82 – Daily Modeled Heating Savings from Increasing SHGC, by Month

Interestingly, Figure 82 shows that the months with the highest impact on heated water savings from this change in SHGC are the months with the mildest temperatures. This is the result of the interaction between the varying solar heat for each month of the year, the thermal mass within the structure of the ship, and the heating load associated with the months. This complex result highlights the usefulness of an energy simulation tool like IDA ICE, which can account for these overlapping influencing factors affecting the ship's energy use. Since the ship's heated water storage tanks have enough thermal storage capacity to supply heat to the ship through the milder nights of the shoulder-season months, this measure is unlikely to have a meaningful impact on the operating costs of the Color Hybrid's heated water system.

Figure 83 shows similar results as Figure 82, but with the hourly average time signature of the *chilled* water system.

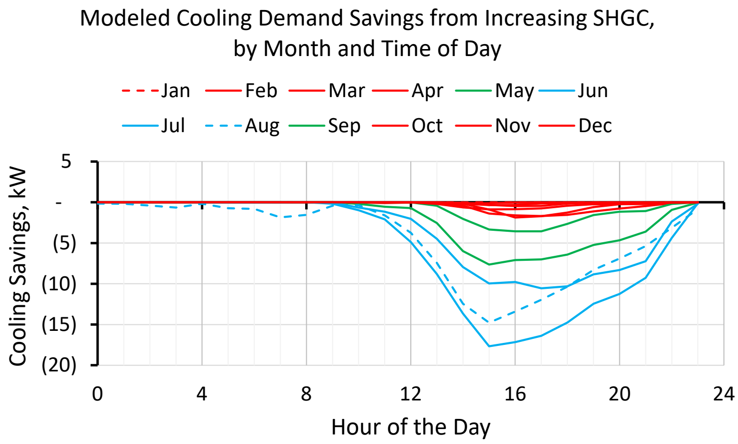

Figure 83 – Daily Modeled Cooling Savings from Increasing SHGC, by Month

In contrast to the heated water demand savings, the chilled water demand savings in Figure 83 show that the month with the second highest cooling load, July, is impacted the most by an increase in SHGC. Since a higher SHGC allows more heat in through the windows, the cooling demand savings are, of course, negative. If one wanted to lower the cooling demand of the Color Hybrid without affecting the operational costs of the heated water system, a reduction in the SHGC could be a reasonable solution. A cooling demand reduction in the warmest summer months would have a meaningful impact on the operation of the electric chillers, as made evident by the duty cycles of the chillers and seawater heat exchanger shown in Figure 44.

Since the solar gains through the windows are reported to have a negative impact on local thermal comfort, a more detailed look into the comfort levels of the occupants can be made using the model. In addition to the air temperature, the operative temperature uses the mean radiant temperature to account for its impact on the thermal comfort of an occupant. The IDA ICE models provide an estimate of the operative temperature, and specifically, an accounting of the number of hours the operative temperature exceeds 27°C. The large aft shop on deck 6 is one of the few occupied zones that show unmet cooling hours in the model results, meaning that the cooling equipment in that zone has inadequate capacity to meet the setpoint temperature. This provides a good way to identify the impact of the change in SHGC on the thermal comfort of the occupant: During the annual simulations, the operative temperature was above 27°C for zero hours in the model with a SHGC of 0.41 and 72 hours in the model with a SHGC of 0.61. Admittedly, this analysis points to the increasing lack of cooling capacity of the equipment in the zone[14] with a rising SHGC rather than the SHGC's impact on local thermal comfort, but it does show that the model can in some cases provide a qualitative indication of changes in local thermal comfort.

To refine this analysis in future work, additional model runs could be made with different ship orientations to reflect the fact that the ship moves throughout the day and its angle with respect to south changes accordingly. In addition, a more detailed analysis could be performed on the actual zones known by the crew to have local thermal comfort issues. For example, occupant thermal comfort could be studied through the re-positioning of occupants closer and farther from the windows within the zones in question.

The above analysis shows that reductions in the window SHGC cause an increase in modeled heating energy demand and a decrease in modeled cooling energy demand, as expected. Depending on the ship's energy supply and storage equipment, the tradeoff can be one-sided to the point where either heating or cooling impacts can be ignored and the SHGC can be maximized one way or the other. In the case of the Color Hybrid, the ship has excess engine waste heat during the mild heating conditions where a higher SHGC has the most impact on the heating demands. On the other hand, the ship requires costly electricity to run electric chillers during the summer conditions where a lower SHGC has the most impact on the cooling demand. Therefore, the design benefits from windows with as low a SHGC as possible. An additional benefit of this design is improved occupant comfort levels. Note that it is not clear what the SHGC of the windows on the Color Hybrid is, nor whether retrofit measures to lower the SHGC would be viable.

8.6 Absorption Chiller

As discussed in Section 4.5, the absorption chiller has mostly been out of operation since the ship was commissioned. In this section, energy savings from operating the absorption chiller are conservatively estimated using a combination of the IDA ICE cooling load estimates from Section 7.2 and the operational data from Section 6.3.

The duty cycles shown in Figure 44 suggest significant opportunity for the absorption chiller to save electricity. Much of, and in many cases all of, the cooling load met by the electric chillers can instead

[14] A more detailed study of the cooling system equipment sizing should be made after cooling energy use data have been collected, as described in Section 11.

be met by the absorption chiller. The absorption chiller has a cooling capacity of 400 kW that it can provide in addition to the cooling provided by the seawater heat exchanger. As a reminder, the operation of the seawater heat exchanger is prioritized when seawater temperatures are low enough to allow its use. The absorption chiller should be second in priority when enough waste heat is available for its operation. The priority of operation of the three types of cooling equipment serving the chilled water system is described in more detail in Section 5.4.2.

However, the duty cycle data do not indicate the cooling load covered by each of the cooling units. Specifically, the flow rate through the seawater heat exchanger is unknown, so during operation it is unclear how much cooling the seawater heat exchanger provides, and therefore it is not possible to estimate the remaining cooling load that could be covered by the absorption chiller. Luckily, there are three months with effectively no use of the seawater heat exchanger.

The following savings estimates use the IDA ICE-estimated cooling loads for August, September, and October. It is assumed that the absorption chiller could cover up to 400 kW of the average hourly cooling load, with the electric chillers covering any remaining load. This is an admittedly conservative savings estimate since the remaining months where electric chillers are used could also benefit from offsetting the use of the electric chillers with the absorption chiller.

The IDA ICE-estimated cooling loads for the three months are shown in Figure 84 for both the baseline case, where only the electric chillers provide cooling, and the efficient-case, where the electric chillers only provide cooling when the cooling load exceeds the capacity of the absorption chiller.

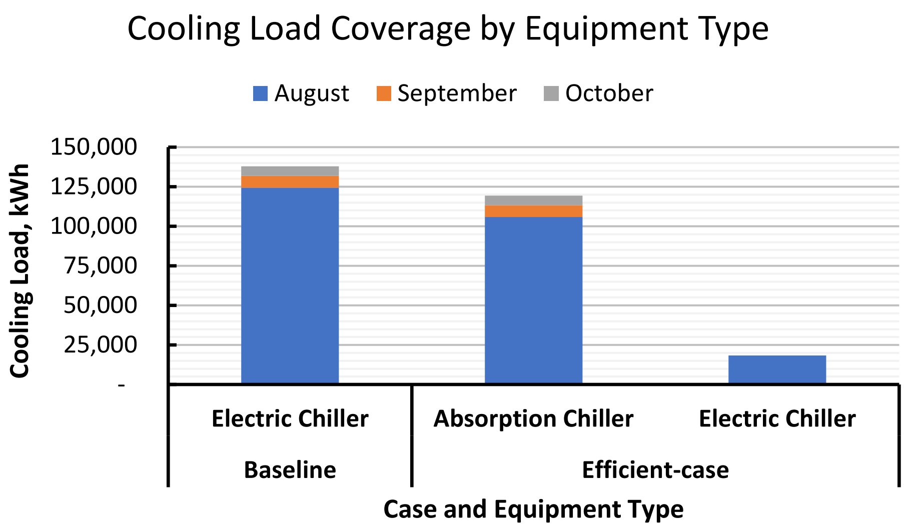

Figure 84 - Cooling Load Coverage, Electric and Absorption Chillers, August to October

It is clear from Figure 84 that the month of August has a much higher cooling demand than the other two months. A look back to Figure 73 confirms this and also serves as a reminder that the month of July has a significant cooling load that could likely be partially covered by the absorption chiller. Note that the cooling load for the baseline case equals the sum of the cooling loads for the absorption chiller and electric chiller in the efficient case.

The electricity consumption of the electric chiller is calculated by dividing the hourly IDA ICE cooling loads by an assumed COP. The design documents provide manufacturer-claimed COP values for different entering freshwater temperatures and chiller loading levels. The range of COPs provided in the design document is 2.13 to 6.6, which corresponds to a freshwater cooling water temperature range of 38°C to 18°C and a loading of 25% to 100%. Since the freshwater temperature and loading conditions are unknown, a mid-to-low range COP of 3.3 was assumed, which corresponds to a freshwater inlet temperature of 38°C and 75% loading. The design documents also provide electricity consumption of the absorption chiller of 2.3 kW, which is used in the estimate of the energy consumption of the absorption chiller in the efficient case. The results of the baseline and efficient-case energy consumption estimates are shown in Figure 85, again by month.

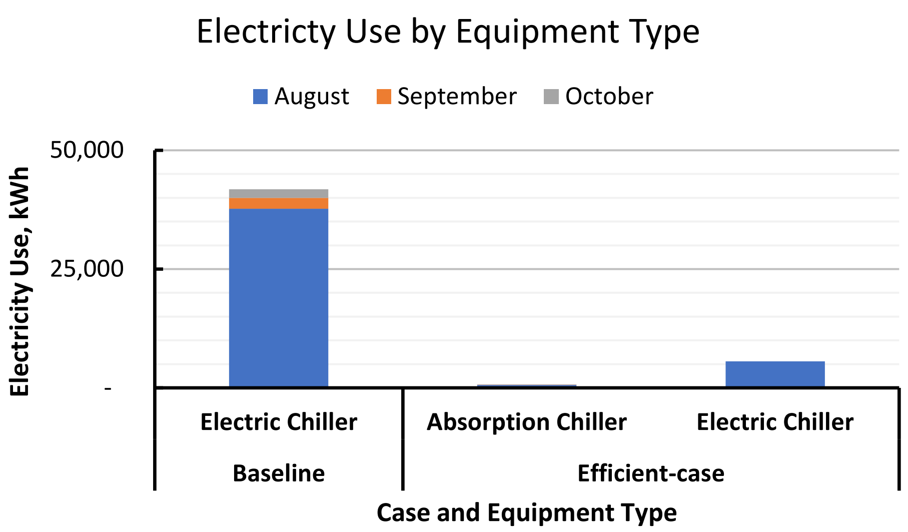

Figure 85 - Electricity Use, Electric and Absorption Chillers, August to October

The stark difference between electrical energy use of the electric chiller and absorption chiller is obvious in Figure 85. The absorption chiller's electrical energy use is nearly invisible in the figure, which is a reminder that the cooling provided by an absorption chiller is nearly "free" where waste heat is available. Table 26 shows the same data in tabular form, along with the electrical energy savings over the course of this three-month period.

Table 26 - Absorption Chiller Measure, Load, Electricity Use, and Savings, August to October

Case	Equipment	Cooling Load Met by Equipment, kWh				Electricity Use, kWh				Electricity Savings, kWh
		August	September	October	TOTAL	August	September	October	TOTAL	
Baseline	Electric Chiller	124,426	7,401	6,014	137,841	37,705	2,243	1,822	41,770	n/a
Efficient-case	Absorption Chiller	105,957	7,401	6,014	137,841	609	43	35	686	**35,487**
	Electric Chiller	18,469	-	-		5,597	-	-	5,597	

It is important to emphasize the over 35 MWh electrical energy savings shown here are lower than what is expected over the course of the year, since the absorption chiller can offset electric chiller use in other months of the year. With additional data collection specific to the operation of the seawater heat exchanger and electric chillers, a more concrete estimate could be made. However, the estimated savings are impactful in terms of reduced operational costs since electricity savings translate directly into a mix of lower shore power demand and lower diesel consumption.

9 Conclusion

One primary conclusion of this master's thesis is that model validation plays a critical role in developing a building energy simulation model, especially in the case of this innovated passenger ship with its many complex and interactive energy systems.

Although a significant effort was made to use detailed design data to guide the original model input parameters, the validation process revealed important differences in the modeled and measured electricity consumption of the hotel system. These kinds of deviations can have detrimental impacts on subsequent model-guided efficiency measure analyses. For example, the original modeled winter electrical demand was only 30% of the demand indicated by the measured data. Without the measured data, it could have been argued that the model should be "trusted" because of the effort to incorporate the detailed design data in the model's original development. If the model had been used at that point, however, the subsequent analysis could easily have misjudged efficiency measures by a similarly large margin.

It is for this reason that future work should be performed before the model's representation of the ship's energy consumption can be fully trusted. Thermal energy use data are needed for full validation of the model. In addition, the adjusted model's electricity consumption results should be reviewed during this second validation procedure because of the interaction between thermal and electrical loads.

It is possible, however, to use the model in its current form as long as care is taken to carefully understand the model's limitations and their possible impact on the trustworthiness of the results. Even without the thermal energy use data required to thoroughly validate the model, the use of detailed design data for the model development, and validation of results based on electrical and operational data, provide some confidence in the model, its energy demands, and the savings estimates for the evaluated efficiency measures.

The second major conclusion of this thesis is that, perhaps surprisingly, efficiency opportunities exist even in a state-of-the-art ship like the Color Hybrid. The analysis shows that relatively simple measures such as reviewing controls of ventilation rates and repairing failed temperature sensors can have meaningful energy savings and operational cost savings for the Color Hybrid. The other measures explored here are likely more useful for ship designers since they require equipment that is difficult to retrofit. The efficiency measures identified in this work and the description of their delivered energy savings should provide a useful starting point for ship operators and designers to estimate potential operating cost impacts of similar measures for their specific applications.

10 Future Work

There are three primary categories of future work identified: data collection and analysis, model validation and adjustment, and analysis of other possible efficiency measures. These are described in more detail below.

10.1 Data Collection and Analysis

Historical hotel system thermal energy consumption data were not available for analysis in this master's thesis. This represents a major deficit in connecting the model's outputs with the real world. Luckily, the equipment already installed as a part of the ship's energy management system should be adequate to capture all the necessary data to provide a useful estimate of the heating and cooling energy demand. The issue is that the data are only available in real time. Implementing a method to log the data over time, which appears to only require some re-programming, is recommended.

Other data collection efforts to understand individual systems on the ship could also be undertaken to improve the model's representation of the ship. Operation of the galley air handler units was identified as an area that could use an improved understanding (Section 5.4.4). Longer-term supply air temperature data that align with the thermal energy use data would help better understand the split between thermal energy requirements of ventilation air versus the space heaters (Section 5.4.4). There are numerous other examples that would benefit from further refinement of the model through additional data collection or visits to the ship. Some examples include use-patterns of wheelhouse defrosters, understanding of car deck heaters, understanding of use-pattern and space conditioning impact of the major equipment loads, the chilled water system piping configuration with respect to the seawater heat exchanger bypass, and use-patterns of electric resistance heaters in the hot water systems.

10.2 Model Validation and Adjustment

The primary model validation should take place after collecting the thermal energy consumption data. Validation and adjustment of the total hotel system energy use, including both the electrical and thermal consumption, will allow for a more holistic approach than that shown in Sections 7.1 and 7.2 for the electricity consumption. The interactions between the electrical and thermal loads are expected to be important enough that the adjustments shown in Section 7.2 will need reconsideration.

In addition to the primary model validation and adjustment, the following smaller efforts have been identified as future work that could strengthen the model's usefulness and validity:

- *Modify the IDA ICE model to include detailed modeling of the energy supply systems*. The model currently uses a COP of 1 for the energy supply systems. This means that the model results provide only the energy demands of the hotel system (Section 5.4). Performing this upgrade to the model would allow it to also provide estimates of delivered energy. Given the significantly different efficiencies of many of the energy systems (for example, the seawater cooling system is tremendously more efficient than the electric chillers), the model results will be highly sensitive to the control strategies employed on the ship, and model validation with thermal energy use data is therefore critical.
- *Use AHU operational data to validate the model.* The AHU operational data available from the ship (Section 4.3) are rich with supply and return temperatures, heating and cooling water valve positions for the heating and cooling coils, fan speeds, and CO_2 levels. These data have been cautiously used to tune the model's AHU parameters to more realistically

match their expected operation. However, when combined with the reassurance of simultaneous thermal energy demand data, the adjustment process will be even more valid.

- *Perform an analysis of the design electrical loads on a design summer day*, like the analysis of the design winter day (Section 4.2). This work can be performed to further assist in model validation. It would be most beneficial to perform this analysis after collection of the thermal energy use data for the cooling water system.
- *Revision of the schedules for other equipment.* Upon receipt of reliable schedule and demand data, adjust schedules of other equipment accordingly. The other equipment is currently set to align with the lighting schedule, but some equipment likely runs throughout the night, for example. Care should be taken to avoid the urge to adjust the modeled and measured power demand curves without reliable data (Section 7.2).
- *Perform a more detailed analysis of the other equipment internal gains in each zone*. This applies especially to the zones with more heat generating equipment than normal (i.e., casino, equipment rooms, etc.) (Section 5.3.4). This analysis would be most beneficial after surveying the rooms with major electrical equipment during a ship visit.
- *Consider adding missing space cooling equipment to the otherwise unoccupied zones* (Section 5.4.3).
- *Further investigate the battery discharge rate in the Sandefjordsfjord to estimate propulsion energy*. Battery discharge during transit in the Sandefjordsfjord could be compared with battery charging in Sandefjord, and adjusted to account for hotel electricity consumption, to possibly estimate the speed versus propulsion energy characteristics for the ship. The transit in the Sandefjordsfjord has additional maneuvering, however, so this approach might not allow extrapolation to the speed versus propulsion energy characteristics of regular sailing.
- *Investigate the CO_2 levels and airflow rates served by the VAV air handler units*. The purpose here is model validation by confirming realistic results in the model. Unrealistic results likely point to areas needing refinement (Section 5.4.4).
- *Investigate wheelhouse energy systems in more detail*. Consider whether the wheelhouse window U-value should be reduced to improve alignment with the space heater sizing in the wheelhouse (Section 7.1.1), or whether the additional other equipment added in the model adjustments is adequate (Section 7.2).
- *Compare modeled to measured extract air temperatures.* Internal gains, airflows between zones, and solar gains are examples of areas that could be investigated to fix any misalignments (Section 6.4).
- *Validation of efficiency measure savings.* The model's results of various energy efficiency measures would benefit from validation of the energy savings themselves. Additional study should be performed to gather measured energy savings of efficiency measures installed in other ships or even buildings. Validation of the energy savings could also be performed by, for example, investigating the data in order to sufficiently understand the impact that the ship's control settings (e.g., space temperature settings, hot- and cold-water temperature settings, cooling plant operation strategy) have on the measured energy use. The model could then be used to estimate energy savings from an adjustment of these control settings, as long as the data sufficiently corroborate the range of the new adjustment.

10.3 Other Possible Efficiency Measures

The breadth of efficiency measures discussed in detail in Section 0 were aligned with the limitations of the model and available data. After further development and adequate validation with thermal and electrical energy use data, as discussed in Section 10.2, the model can be used to investigate a wider variety of efficiency measures. Examples of efficiency measures that would be applicable to the Color Hybrid include:

- *Heat Pumps.* The waste heat from the freshwater cooling system and chilled water system, and the cooling to the heated water system caused by the accommodation heating loads, can be mutually beneficial to one another if the quality of energy is increased by the use of a heat pump. This exchange of heating and cooling by the various systems could greatly reduce the hotel system's overall delivered energy demand. While the heating and cooling systems in Color Hybrid take advantage of an abundance of waste heat from the engines, this heat pump concept would be particularly interesting in a ship that does not have available waste heat, for example an all-electric ship.
- *Improve control to reduce use of resistance heaters.* In the Color Hybrid, the use of resistance heaters in the heated water system has a direct impact on operational costs, either with the required purchase of shore power or the need to use more diesel to generate electricity. Efforts to better understand the conditions where electric resistance heaters are used, and methods to optimize their control in order to reduce their use as much as possible, would be of particular value to the ship's operators. Control optimization should take place automatically, or at minimum, after implementation of measures that impact the effective storage capacity of the thermal storage tanks.
- *Increase thermal storage capacity.* This is not necessarily an efficiency measure, but instead an operational optimization measure. Increasing the thermal storage capacity would result in the ability to sail longer distances or dock longer without a heat source. This could be a useful concept for ships that have no waste heat source but that receive a re-charging of their thermal storage from a land-based energy source while docked.
- *Investigate efficiency improvements in galley cooking equipment.* The galley cooking equipment is all-electric, which is known to result in electrical demands of 200 kW (Table 20). Attention to minimizing this load by improving the control of the equipment, and perhaps replacing inefficient equipment, could prove valuable to the ship's operation. This analysis could take place outside the model.
- *Begin using seawater heat exchanger earlier in the fall.* The faulty seawater temperature reading identified in Figure 40, the high chilled water return temperatures coming into the seawater heat exchanger identified in Figure 45, and lack of seawater heat exchanger operation as shown in Figure 44, all point to the possibility that the seawater heat exchanger could be re-engaged earlier in the fall, sooner after the seawater has cooled down from its high summer temperatures. This would impact the electricity requirements of the hotel, assuming the absorption chiller is still not in operation.

There are, of course, many other possible measures that the model could be used to analyze. A fully calibrated model would also be a useful tool for conceptualizing efficiency measures applicable to ships other than the Color Hybrid.

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NTNU

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Science and Technology