Dynamic Simulation of Ship-System Thermal Load Management

Patrick T. Hewlett and Thomas M. Kiehne

Abstract — Anticipating highly dynamic and reconfigurable future ships, the US Navy has sought to develop modeling and simulation capabilities for transient, electrical-mechanical-thermal, shipboard interactions at the system level. In support of this work, an object-oriented Dynamic Thermal Modeling and Simulation (DTMS) framework written in C++ has been in use for several years. As reported in this paper, DTMS has recently been augmented to model two-phase flow and heat transfer for simulation of a shipboard vapor-compression chiller and its attendant loads. A controls methodology has been implemented in the heat exchanger models to monitor their relevant states, chilled water enthalpy, and refrigerant liquid level. These heat exchangers have been integrated with a heavily-customizable, centrifugal compressor model focused on required power input rather than the detailed dynamics of fluid compression. The heat exchangers and centrifugal compressor, along with a model of a thermostatic expansion valve, have been used to assemble a simulation of a 200-ton marine chiller predicated on baseline parameters for the Navy’s current destroyer. This chiller has been connected with thermal loads of varying magnitude to demonstrate controller response during full-load and part-load operation. The final simulation reported here consists of 22 thermal loads ranging from 8 to 256 kW with chilled water supplied by two chillers. Results are compared with both steady-state-predicted values and previous dynamic simulations using commercial software.

I. INTRODUCTION

The US Navy continues to support applied research on the design, development, and simulation of their future fleet of all-electric ships (AES). Central to the Navy’s vision is the use of electricity as the primary energy transport means for the majority of ship systems. For example, the ship’s propulsion has historically been handled by dedicated gas turbines connected directly to reduction gears which control the propeller. In the AES concept, prime movers are connected directly to generators, from which electrical power is sent to the propellers (via motor drives), and also to other ship systems.

The motivation behind this design shift is threefold. First, a future surface ship must be robust and reconfigurable; with an optimal power distribution grid, ship sectors damaged during combat can be isolated from the rest of the grid, minimizing damage and prolonging the ship’s operability. Second, with a standard modular power grid, maintenance and cost of repair to ship power systems would be more efficient and cost-effective. Finally, the Navy expects to integrate various high-energy weapons and radar systems in future warships. An optimized power distribution grid would allow the ship to employ large pulses of energy required to implement these advanced systems.

However, the introduction of advanced electronics and pulsed-energy systems on a surface ship is not without consequences. From a heat generation point of view, the AES will produce significant thermal side effects that have the potential to produce catastrophic failures at system and component levels. Thus, shipboard thermal management is considered an enabler for the innovative technologies likely to appear on an AES.

Currently, the Arleigh Burke DDG-51 class destroyer employs five 200-ton marine chiller units to handle active cooling of ship systems and components. Every shipboard component, from the smallest processor chip to the largest gas turbine, contributes dynamically to this thermal management challenge due to the generation of “waste heat”. It has been estimated [1] that, on average, approximately 681 tons of waste heat is rejected from an Arleigh Burke class warship. However, this average value does not capture the magnitude of peak waste heat during transient situations. On a highly dynamic, controls-oriented ship such as the notional AES, steady-state heat values provide little utility from a reconfiguration or system failure perspective. When a fully capable AES is deployed, shipboard cooling requirements are predicted to have increased by as much as 700% [1]. However, this steady-state value does not include the integrated effects of dynamic power buildup and adaptive grid response following the introduction of high-energy weapons and sensors. For the above reasons, it is the objective of the research reported in this paper to simulate shipboard thermal load management from a dynamic, controls-based, system-level perspective.

II. DYNAMIC THERMAL MODELING AND SIMULATION

The Navy has supported fundamental research directed at modeling of ship systems while emphasizing electrical, mechanical, and thermal components. The long-term goal has been development of robust, cost-practical, simulation software able to couple physics-based models in all three domains to simulate their dynamic interactions. Initially, a wide range of dynamic thermal models were developed in commercial software packages. For example, an industrial chiller was modeled in a steady-state fashion using CycleTempo, a research-based optimization tool for power
and refrigeration cycles, and dynamically using ProTRAX, a controls-based dynamic plant simulator. Recently, research has shifted from utilization of commercial software to development of a customizable in-house framework called the Dynamic Thermal Modeling and Simulation (DTMS) framework [2], [3]. Development of this framework is heavily represented in this paper. The goal for DTMS is that it be robust, scalable, and accessible to a larger community.

The modeling cornerstone behind DTMS is Bond graph theory written in a C++ environment. Bond graph theory, with its connection paradigms and data flow principles, allows complete modularity among all models. DTMS builds its basic component models from a flow-effort standpoint. These flow and effort models are represented by classes that generically determine mass flow rate (via flow models) and pressure (via node models). The object-oriented nature of C++ allows these flow and node models to communicate necessary information between connected components. The DTMS component models are then written to capture applicable physics while allowing the user total freedom of both configuration and complexity.

The two-phase, time dependent models reported here necessitate the availability of multiple fluids with frequent update of thermodynamic properties. Using standard properties for water [4] and REFPROP table values for refrigerant 134a [5], fluid subroutines were integrated into the DTMS framework to update relevant thermodynamic properties at every time step. From an object-oriented coding standpoint, this involved the use of containment and polymorphism. The development of fluid subroutines for DTMS is discussed in [2], [3]. These subroutines were validated for accuracy against benchmark data and compared with both data and a previous simulation of the starboard freshwater system for a DDG-51[6].

III. CHILLER COMPONENT MODELING AND VALIDATION

An initial research objective was simulation of the York marine, 200-ton vapor compression chiller in the DTMS framework. Models for various subcomponents of this chiller are discussed below.

A. Heat Exchangers: Evaporator and Condenser

Representation of two-phase heat exchanger physics is critical to a robust and detailed chiller simulation. Very few purely analytical, physics-based models capture the liquid-to-vapor separation using empirical relations during phase change. Here the two-phase heat exchanger state equations used in DTMS are provided and the importance of shell-side liquid level, specifically its effects on fluid flow and heat transfer, are discussed. In addition, two-phase heat transfer relationships implemented in DTMS for pressure drop and the heat transfer coefficients are provided.

In developing a new dynamic heat exchanger model, the usual conservation of mass and energy principles were used. However, since the outlet refrigerant quality (mass of vapor/total mass) differs greatly for a condenser and evaporator, two very different analytical approaches were involved. Proper selection of state variables requires consideration of the features of these heat exchanger models. The York 200-ton industrial chiller [7] contains a “flooded” evaporator, which implies that the refrigerant liquid level and corresponding void fraction inside the heat exchanger shell are of particular importance. In fact, the assumption of stratified flow inside the shell allows the compressor module to control the chilled medium using the refrigerant mass flow rate. Obviously, the thermal state of the chilled medium is the main focus of the chiller itself, so either temperature or enthalpy of tube fluid should be designated as a state variable as well. Since the fluid subroutines in DTMS often rely on enthalpy, the refrigerant liquid level and tube bundle enthalpy were selected as state variables.

A.1 Evaporator

Applying the conservation of mass principle to the shell-side fluid entering at an initial quality \(x_o\), a simple balance of inlet and outlet masses is obtained. After dividing the shell into sub-volumes corresponding to saturated vapor and liquid states, an expression relating the cross-sectional wetted tube surface area to known quantities is obtained:

\[
(\rho_f - \rho_g) \int L \frac{dA}{dt} = w_{in} - w_{out}
\]

It is then possible to solve for shell-side liquid level using a geometrical relationship involving the liquid level, the shell diameter, and a lumped tube diameter.

![Fig 1. Conservation of energy applied to evaporator shell.](image)

Applying the conservation of energy principle allows (1) to be written in terms of the heat transfer between the shell and tube fluids. The overall energy balance accounts for inlet and outlet enthalpies as well as heat transfer \(Q_{evap}\) from tube-side fluid to shell-side fluid. After separating shell volumes into saturated liquid and vapor, and assuming saturated vapor at the shell outlet, the result is:

\[
(\rho_f h_f - \rho_g h_g) L \frac{dA}{dt} = w_{in} \left[(1-x_o)h_f + x_o h_g\right] - w_{out} h_g + Q_{evap}
\]

Equations (1) and (2) can be manipulated to obtain the liquid level state equation as a function of the heat transfer. The result is:
The entire condenser shell and is considered too low to have
This volume corresponds to about one-sixth the volume of
prevent gas bubbles from entering the liquid outlet nozzle.
volume of liquid shell-side fluid substantial enough to
shell. Two-phase shell-and-tube condensers often contain a
constitutes the majority of heat transfer inside the condenser
heat transfer applies. While boiling heat transfer remains
largely empirical, correlations are available pertaining to
experiences heat transfer due to vapor cross flow over a tube
bundle, i.e., that not submerged in shell-side liquid,
the Reynolds and Prandtl numbers. The “dry” portion of
Nusselt number (dimensionless heat transfer coefficient) to
range. Specifically, correlations adopted from [9] relate the
boiling over tube bundles within an allowable heat flux
and can be found in any heat transfer textbook, e.g., [8].
A.4 Evaporator Phase Change Model
Assuming stratified flow, there are two distinct portions of
the shell that affect heat transfer. For portions of the tube
bundle submerged in a pool of refrigerant liquid, boiling
heat transfer applies. While boiling heat transfer remains
largely empirical, correlations are available pertaining to
boiling over tube bundles within an allowable heat flux
range. Specifically, correlations adopted from [9] relate the
Nusselt number (dimensionless heat transfer coefficient) to
the Reynolds and Prandtl numbers. The “dry” portion of the
tube bundle, i.e., that not submerged in shell-side liquid,
experiences heat transfer due to vapor cross flow over a tube
bundle. These relationships are lengthy yet self-explanatory
and can be found in any heat transfer textbook, e.g., [8].
A.5 Condenser Phase Change Model
Having assumed well-mixed flow, condensation
constitutes the majority of heat transfer inside the condenser
shell. Two-phase shell-and-tube condensers often contain a
volume of liquid shell-side fluid substantial enough to
prevent gas bubbles from entering the liquid outlet nozzle.
This volume corresponds to about one-sixth the volume of
the entire condenser shell and is considered too low to have
any effect on shell-side heat transfer. Thus, the outer heat
transfer coefficient for the condenser is given by a standard
relationship describing condensation over tube bundles [8]:

A.6 Shell-side Fluid Dynamics
In order to integrate flow models in the DTMS framework, it is necessary to have a relationship between
flow and pressure differential. The approach here invokes the
Lockhart-Martinelli relationship for pressure drop in
pipes with separated, two-phase flow as provided in [10]. In
this approach the liquid-only pressure drop is multiplied by
the two-phase coefficient to obtain the two-phase pressure
drop. The details of this approach are provided in [6].
B. Centrifugal Compressor and Expansion Valve
In modeling a dynamic chiller, it is recognized that initial
transients of the heat exchangers are often several orders of
magnitude slower than those of the compressor [11], [12].
For this reason, the constant-speed, variable-geometry
centrifugal compressor in the York 200-ton chiller [7] was
modeled as quasi-steady-state. Using dimensional analysis
based on pressure rise, mass flow rate, and suction area
controlled by variable inlet guide vanes (IGV) on the
compressor) as the pertinent variables, results in:

\[ W = \frac{\rho}{\rho_v} \left[ \frac{\dot{V}}{D^2} \left( \frac{\Delta P_{\text{max}}}{\Delta P} \right)^{1/2} \right] \left( \frac{\dot{m} \Delta P}{\rho_v D^2} \right) \]  

The tilde on the pressure values in (6) indicates dimensionless pressure based on values of normalized head. Design
values for mass flow rate, pressure, inlet area, and pressure rise are obtained from the compressor manufacturer. The
compression efficiency is then used to determine the power
required based on isentropic specific work:

\[ P = \frac{W \gamma}{n \gamma - 1} R T_m \left( \frac{P_{\text{out}}}{P_{\text{in}}} \right)^{\gamma-1} \]  

A generalized efficiency map is employed to determine
actual compressor power requirements.
For the thermostatic expansion valve, a similar
dimensional analysis was performed, with the result:

\[ W = \frac{Y \dot{m}_{\text{max}}}{\sqrt{\Delta P_{\text{max}}} \sqrt{P_{\text{in,D}}}} \sqrt{\Delta P} \]  

The expansion valve monitors the open valve fraction, \( \gamma \), to
control the shell-side liquid level of the condenser. Again,
the details of this approach are provided in [6].
C. Chiller Controls

The DTMS framework has the ability to assign PID feedback control to models, thus controlling output values of other devices. The user is able to characterize the control transfer function by specifying three constants in the source file. These constants represent the proportional gain $k_p$, the derivative time constant $\tau_d$, and the integrated time constant $\tau_i$, respectively. The controlled variable is bounded by upper and lower limits, and the metered variable is assigned a designated target value. After the user-defined parameters have been initialized, DTMS implicitly integrates the set of first-order, ordinary differential equations associated with the transfer function. From the previous time step value, a closed-loop output is determined after checking the user-defined upper and lower bounds.

The York 200-ton industrial chiller contains three basic controllers that have been replicated in the DTMS model. The chiller’s primary control monitors the tube-side exit enthalpy leaving the evaporator. This is accomplished via adjustment of the IGV position, which in turn affects the distribution of heat transfer in the evaporator shell. Second, the condenser’s “reservoir” of shell-side liquid must be monitored to ensure that the liquid level does not become too high or low. A high liquid level would diminish shell-to-tube heat transfer by affecting the fraction of condensation heat transfer, while a low liquid level could induce dry-out of shell-side fluid and subsequent failure. Finally, compressor mass flow rate is controlled by a hot-gas bypass (HGBP) valve ensuring that a suitably high flow rate is maintained to prevent compressor surge.

Over-adjustment of the refrigerant mass flow rate can induce flow instabilities. For this reason, correct user-defined control parameters are important to an accurate and stable flow network solution when control monitoring is performed. While a user could analytically determine the exact values of these coefficients via root locus or Bode plot techniques, trial-and-error parameter determination has been used for all DTMS controls. Regardless, some basic control theory allows the user to make sound assumptions when performing a trial-and-error study. For example, the proportional gain $k_p$ merely multiplies the closed-loop feedback by the gain value before supplying this product to the controller. A large gain corresponds to more drastic response. Thus, if the metered variable is oscillating around or slowly approaching the set point, an increase in proportional gain might be needed. Integral control adjusts the controlled value to approach the metered set point using current and past measured errors. This often can lead to oscillations as the coefficient $\tau_i$ increases. Also, since the derivative coefficient $\tau_d$ minimizes overshoot produced from integral control, it can be used to minimize oscillation but may produce instabilities if set at too high a value.

D. Chiller Configuration

With the models and controls representing the DTMS dynamic chiller now established, a detailed layout and description of the overall configuration can be provided. Any chiller simulation must feature three flow networks as shown in Figure 2: the refrigerant cycle, the freshwater loop, and the seawater loop. As depicted, the major components of the chiller must be connected via a series of nodes, pipes, and pumps to operate successfully. The independent node and separate flow model shown in the condenser shell represent the 1/6th “reservoir” of liquid present inside the condenser shell.

E. Chiller Results at Startup

To test chiller operation, startup conditions were simulated using initial conditions from [7] as shown in Table 1. The actual chiller employs R-236fa as its shell-side refrigerant. However, to establish a basis for comparison with previous results, these simulations were performed using R-134a.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Evaporator</th>
<th>Condenser</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shell Length</td>
<td>3.398 m</td>
<td>3.398 m</td>
</tr>
<tr>
<td>Tube Surface Area</td>
<td>149.85 m²</td>
<td>267.1 m²</td>
</tr>
<tr>
<td>Circulating Water</td>
<td>0.0454 m³/s</td>
<td>0.0416 m³/s</td>
</tr>
<tr>
<td>Tube-side Pressure</td>
<td>0.125 MPa</td>
<td>0.325 MPa</td>
</tr>
<tr>
<td>Inlet Water Temp.</td>
<td>283.54 K</td>
<td>304.26 K</td>
</tr>
<tr>
<td>Outlet Water Temp.</td>
<td>279.82 K</td>
<td>309.26 K</td>
</tr>
<tr>
<td>Saturation Pressure</td>
<td>0.323 MPa</td>
<td>1.021 MPa</td>
</tr>
<tr>
<td>Tube Number (Passes)</td>
<td>240 (2)</td>
<td>312 (2)</td>
</tr>
<tr>
<td>Outer Tube Diameter</td>
<td>0.01905 m</td>
<td>0.01905 m</td>
</tr>
</tbody>
</table>

Results for a standalone DTMS chiller at startup conditions dictated by [7] and Table I are presented below. These simulations were run for 500 seconds; however, transient characteristics have dissipated well within 200 seconds.

As shown in Figure 3, flow network solution instabilities are evident in the compressor power and flow. The HGBP valve closes during the first 15 seconds of the simulation, causing the condenser liquid level to rise. Once the liquid level set point is reached, the valve reopens and mass flow rises to near 6 kg/s. This response is due to a high $k_p$ value. Several other coefficient combinations were tried, but this particular $k_p$ value provided the best response. The refrigerant mass flow rate reaches steady-state prior to 60 seconds at a value of 5.016 kg/s. Compressor power varies as a function of mass flow rate, eventually reaching a steady-state value of 149 kW. Figure 4 illustrates the HGBP and thermostatic expansion valve response as condenser...
liquid level and evaporator tube-side enthalpy approach their set points.

![Graph](image1)

**Fig 3. Compressor power and refrigerant mass flow rate at startup.**

![Graph](image2)

**Fig 4. HGBP and expansion valve response at startup.**

To evaluate the success of the standalone DTMS chiller simulation, steady-state values were compared with known results from previous York 200-ton chiller simulations, i.e., steady-state values from ProTRAX simulations [13] and the results of steady-state calculations. These results are shown in Table II and indicate that the DTMS standalone chiller agrees well with both ProTRAX-generated and steady-state values. In fact, the DTMS simulation is markedly closer to steady-state results than is ProTRAX for all outputs except condensing pressure and discharge temperature. The chilled freshwater temperature lies remarkably close to the York-defined value.

**TABLE II**

**COMPARISON OF STANDALONE DTMS SIMULATION WITH ProTRAX AND STEADY-STATE VALUES FOR A YORK 200-TON ChILLER**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>DTMS</th>
<th>ProTRAX</th>
<th>S.S.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant Mass Flow</td>
<td>kg/s</td>
<td>5.016</td>
<td>4.663</td>
<td>4.897</td>
</tr>
<tr>
<td>Compressor Power</td>
<td>kW</td>
<td>149.02</td>
<td>145.29</td>
<td>155.92</td>
</tr>
<tr>
<td>Condensing Pressure</td>
<td>MPa</td>
<td>1.0495</td>
<td>1.0340</td>
<td>1.0210</td>
</tr>
<tr>
<td>Evaporating Pressure</td>
<td>MPa</td>
<td>0.3304</td>
<td>0.3502</td>
<td>0.3230</td>
</tr>
<tr>
<td>Discharge Temperature</td>
<td>K</td>
<td>333.25</td>
<td>325.61</td>
<td>324.64</td>
</tr>
<tr>
<td>Freshwater Outlet Temp</td>
<td>K</td>
<td>279.79</td>
<td>280.02</td>
<td>279.82</td>
</tr>
<tr>
<td>Seawater Outlet Temp</td>
<td>K</td>
<td>308.91</td>
<td>N/A</td>
<td>309.26</td>
</tr>
</tbody>
</table>

**IV. DYNAMIC THERMAL LOAD SIMULATIONS**

System-level behavior with chillers tied to a network of shipboard loads is demonstrated in this section. First, the York chiller model is connected to a series of thermal loads totaling 703.3 kW (200 tons) and simulated at startup conditions. The simulation is then modified to allow for specified variables to change in mid-simulation at a user-defined time. This is intended to simulate the onset of a part-load condition and might represent a number of transient scenarios (startup, full speed, combat, shutdown) anticipated for the HVAC system of a warship. Finally, an actual shipboard system, the starboard chilled water system of the DDG-51, is simulated at startup and part-load conditions, and results are compared with available performance data.

Consider first a chiller (Figure 5) connected to a set of seven notional thermal loads corresponding to 200 tons of cooling capacity. By simulating a startup scenario, controls-based response of the chiller to the introduction of multiple heat loads, of varying quantity at various distances, may be analyzed. The chilled water was assigned an enthalpy corresponding to the load set point temperature (283.5 K). The supply pump was assigned a design mass flow rate of 45.41 kg/s corresponding to data from [7]. Thermal loads, as shown in Table III, were introduced to analyze load handling and behavior with respect to chiller placement.

**TABLE III**

**THERMAL LOAD COOLING EXAMPLE: LOAD STATISTICS**

<table>
<thead>
<tr>
<th>Load</th>
<th>Magnitude (kW)</th>
<th>Distance (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>132.3</td>
<td>10</td>
</tr>
<tr>
<td>2</td>
<td>72</td>
<td>20</td>
</tr>
<tr>
<td>3</td>
<td>121</td>
<td>30</td>
</tr>
<tr>
<td>4</td>
<td>86</td>
<td>40</td>
</tr>
<tr>
<td>5</td>
<td>129</td>
<td>50</td>
</tr>
<tr>
<td>6</td>
<td>58</td>
<td>60</td>
</tr>
<tr>
<td>7</td>
<td>104</td>
<td>70</td>
</tr>
</tbody>
</table>

Results for load temperature and valve position versus time at full load are shown in Figures 6 and 7. It is evident that the highest loads (Loads 1, 3, 5, and 7) require the largest valve flow areas. A figure illustrating mass flow rate behavior versus time would be repetitive here, as load mass flows correspond directly to valve position via their effect on flow conductance. However, the behavior of high loads near the chiller (Loads 1 and 3) is unexpected. For example, the greatest load in the simulation, Load 1, is immediately cooled to a point below the set point temperature. The chiller effect on supply water is instantly experienced at Load 1, while the chilled water takes much longer (on the order of
200 seconds) to fully influence Load 7. The distance from the chiller to Load 7, approximately 70 m, leads to a peak load temperature of 286.1 K. Thus, peak load temperatures respond to two factors: absolute value of the thermal load and distance from the chiller.

The DTMS controls structure permits time-dependent analysis through the introduction of an “event.” An event implements the code’s control system to adjust the set point to a user-defined percentage of the original at a user-defined time in the simulation. Consider a 75% part-load (25% reduction in each load) condition introduced 500 seconds into a simulation as implemented in the notional seven-load configuration. Load temperature and valve position response are shown in Figures 8 and 9. The first 400 seconds, representing startup of the simulation, have been omitted from these figures.

Each of the seven loads responds immediately to partial load conditions initiated at 500 seconds. The redefined enthalpy set point corresponds to 281.9 K; all loads have attained this value after about 625 seconds of simulation time. The position of each load valve responds to partial load conditions as well. Each valve first falls below its previous steady-state setting before settling back to the previous value at approximately 900 seconds of simulation time. Since the mass flow rate was reduced in calculation of the updated enthalpy set point, relative valve positions were not expected to change from their previous steady-state value for this part-load condition. These part-load results correspond well with the ProTRAX results in [13].

On the DDG-51, a considerable portion of the ship’s overall cooling load is due to the starboard, freshwater, chilled loop. This loop involves 22 separate thermal loads representing thermally active compartments on the ship. The system configuration is shown in Figure 10 and the load distribution in Table IV. Each thermal compartment is represented by a specified heat input and employs a valve controller to monitor return freshwater enthalpy. Since the total heat load is 1324 kW split among 22 compartments, two 200-ton chillers are necessary for load management.

Figure 10. DDG-51 starboard freshwater loop configuration.
The thermal compartments in Figure 10 follow the William and Zebra load classifications set forth in [14]. Note that the York #1 chiller is closest to the largest thermal loads in the simulation, 3W and 4W. These loads can be attributed to the radar and defense electronics packages located in these compartments. To permit direct comparison with Navy data, each load, supply, and return is assigned pipe geometry according to the parameters set forth in [2].

<table>
<thead>
<tr>
<th>Zebra Compartment</th>
<th>Load Value [kW]</th>
<th>William Compartment</th>
<th>Load Value [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1Z</td>
<td>43.16</td>
<td>1W</td>
<td>71.33</td>
</tr>
<tr>
<td>2Z</td>
<td>57.86</td>
<td>2W</td>
<td>15.12</td>
</tr>
<tr>
<td>3Z</td>
<td>62.57</td>
<td>3W</td>
<td>256.30</td>
</tr>
<tr>
<td>4Z</td>
<td>38.27</td>
<td>4W</td>
<td>193.62</td>
</tr>
<tr>
<td>5Z</td>
<td>21.56</td>
<td>5W</td>
<td>81.35</td>
</tr>
<tr>
<td>6Z</td>
<td>76.50</td>
<td>6W</td>
<td>8.58</td>
</tr>
<tr>
<td>7Z</td>
<td>45.44</td>
<td>7W</td>
<td>68.73</td>
</tr>
<tr>
<td>8Z</td>
<td>55.54</td>
<td>8W</td>
<td>27.57</td>
</tr>
<tr>
<td>13Z</td>
<td>11.36</td>
<td>9W</td>
<td>90.22</td>
</tr>
<tr>
<td>A/C Waste Heat</td>
<td>10W</td>
<td>57.86</td>
<td>55.04</td>
</tr>
<tr>
<td>York #1</td>
<td>8.23</td>
<td>11W</td>
<td>27.43</td>
</tr>
<tr>
<td>York #4</td>
<td>8.23</td>
<td>Total</td>
<td>1324</td>
</tr>
</tbody>
</table>

In the results that follow, the metered freshwater return enthalpy is assigned a set point of 43.778 kJ/kg and the supply freshwater enthalpy set point remains at 28.146 kJ/kg, corresponding to the 279.8 K set point in [7]. Figure 11 shows the temperature response of each thermal load. As expected, both the load magnitude and its proximity to a chiller impact thermal response. Since the peak loads, 3W and 4W, are each only 10 m from A/C#1, they do not exhibit the highest peak temperatures in the simulation. Instead, Load 9W, the largest load located furthest from a chiller attains the peak temperature in this system. Instead, Load 9W is the last to reach the set point, at approximately 600 seconds. Small, proximate loads, such as chiller waste heat loads for A/C #1 and A/C #4 drop below the set point due to immediate cooling. In Figure 12, the valves corresponding to the 22 loads exhibit fairly smooth response before attaining steady-state near 1800 seconds. Obviously, the larger thermal loads result in greater supply flow. Thus, the valve corresponding to Load 4W exhibits a flow area near 80% of maximum.

To assess the effectiveness of the DTMS chiller model, the compartmental freshwater mass flow rates were compared with DTMS Version 1.0 [2] and Navy data [14]. As shown in Table V, the DDG-51 starboard loop simulation with dynamic DTMS chiller results in a 2.91% greater mass flow rate on average. This can be attributed to control-based variations in supply temperature driven by evaporator liquid level, whereas the DTMS Version 1.0 simulation consisted of chilled water supplied at constant pressure and enthalpy. The greatest percent difference, corresponding to the cumulative A/C exhaust heat load, can be attributed to uncertainty in load placement information from [14]. Otherwise, with percent variations on the order of 5% or smaller, successful incorporation of the DTMS chiller models for actual naval ship system simulation has been adequately demonstrated.

<table>
<thead>
<tr>
<th>Load</th>
<th>Data [kg/s]</th>
<th>Simulation with DTMS Chillers</th>
<th>% Error DTMS with Data</th>
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| Tot/Avg | 85.05 | 83.9 | 87.53 | 2.91% |
V. CONCLUSIONS

The Dynamic Thermal Modeling and Simulation (DTMS) framework, with its current control architecture, has been applied to the simulation of large dynamic thermal systems found aboard Navy surface ships. The ability to simulate dynamic, system-level events that might occur on a warship has been demonstrated. Specifically, a dynamic chiller model has been developed, tested, and demonstrated via connection to time-dependent thermal loads in a ship system-level simulation. As documented in Section IV, simulation of startup, full-load, and part-load operation produces outcomes within 7.6 % of both steady-state predicted and dynamic commercial software results obtained previously. Given the various load responses at startup and part-load, these simulations indicate the potential for use of DTMS as a shipboard, HVAC, optimization tool. In fact, two non-traditional shipboard cooling concepts have been implemented in DTMS and compared with current chilling systems. It is concluded that these advanced cooling strategies not only provide immediate power savings relative to baseline models, but also provide the potential for simulation of dynamic reconfiguration deemed necessary for future naval surface ships.

During this work, considerable difficulty was experienced obtaining stable, accurate DTMS simulation using the current control architecture. Without a more sophisticated controls approach, it is the responsibility of the user to find suitable control coefficient values for every controller in a simulation. When only one or two controllers are implemented, as in a standalone chiller simulation, controls tuning is straightforward. However, simulations featuring many controllers present real challenges and an extremely difficult "tuning" process. Further, the adjustment of other user-defined values (design mass flow rates, initial conditions, etc.) affects control relationships as well.

Before any attempt to simulate highly dynamic AES scenarios, it is recognized that a more robust control strategy must be developed and implemented. For example, allowing the user more freedom in controller type and open/closed loop setup would provide more flexibility in achieving stability and accuracy requirements. More sophisticated control analysis techniques, such as frequency response plotting, could provide more effective tuning methods. In addition, advanced control systems such as digital control could streamline the interactions of several controllers into a single response. These approaches, and others, will be investigated in future work.

Variables:

- \( A \) area \([m^2]\)
- \( D \) diameter \([m]\)
- \( g \) acceleration due to gravity \([m/s^2]\)
- \( h \) enthalpy \([kJ/kg]\)
- \( h \) heat transfer coefficient \([W/m^2\cdot K]\) (overbar for average)
- \( k \) thermal conductivity \([W/m\cdot K]\)
- \( L \) length \([m]\)
- \( t \) liquid level \([m]\)
- \( N \) number of tubes
- \( P \) power \([kW]\)
- \( p \) pressure \([N/m^2]\)

Subscripts:

- \( c \) compressor
- \( cond \) condenser
- \( D \) dependent, design
- \( evap \) evaporator
- \( f \) saturated liquid, fluid
- \( fg \) latent heat of vaporization
- \( g \) saturated vapor
- \( in \) inlet
- \( max \) maximum
- \( o \) initial, outer
- \( out \) outlet
- \( sat \) saturated
- \( w \) wetted, wall

References